# EFFECT OF THE DESIGN OF THE MECHANICAL AMPLIFIER IN THE BEHAVIOUR OF A HIGH POWER ULTRASONIC TRANSDUCER

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**Resumen**

Los procesos asistidos por ultrasonidos de potencia se han convertido en una solución atractiva en el ámbito industrial debido a su sostenibilidad (bajo consumo energético, procesos no contaminantes). Con el fin de obtener los efectos deseados en estos procesos, los transductores ultrasónicos de potencia tienen que trabajar a la frecuencia deseada y consiguiendo valores altos de desplazamiento con bajos valores de tensión mecánica en cada una de sus partes constitutivas (transductor tipo Langevin, amplificador mecánico o horn, y radiador de placa).

El horn es donde se produce la amplificación del desplazamiento, y el lugar en el que se dan los mayores valores para la tensión mecánica, principalmente en las zonas de contacto entre el horn y la placa y la zona en la que se produce el cambio de sección en el propio amplificador mecánico.

El objetivo de este trabajo es mostrar el comportamiento, obtenido mediante métodos numéricos, de un transductor de placa rectangular para amplificadores mecánicos con diferentes secciones.

**Palabras-clave:** transductores, ultrasonidos de potencia, amplificador mecánico, método de los elementos finitos.

**Abstract**

Industrial processing assisted by high-power ultrasound has become an attractive field for industries due to its sustainability (low energy consumption, non-pollutant processes). In order to obtain the desired effects, the high power ultrasonic transducers have to work properly, vibrating at the desired frequency and achieving high values of displacement with a minimum mechanical stress in its different parts (Langevin type vibrator, mechanical amplifier or horn, and plate radiator).

The horn is the part of the transducer where the highest displacement amplification takes part, and also the place where the highest mechanical stresses occur, mainly in the connection point with the plate radiator and in the junction of the two different sections.

The objective of this work is to show the behaviour, obtained by numerical methods, of a rectangular plate transducer when different shapes of mechanical amplifiers are designed.

**Keywords:** transducers, high-power ultrasound, mechanical amplifier, finite element method.

# 1 Introduction

Industrial processing assisted by high-power ultrasound (HPU) has become a new, green and efficient technology with a high potential in its implementation. Previous researches [1, 2] show that these HPU technologies provide a good performance in processes such as particle agglomeration, ultrasonic cleaning or defoaming, among others. In the particular case of food dehydration, it has been proved by [3] that HPU provides a faster and more economic performance, also improving the quality of the final product [4].

In order to produce changes in the internal structure of the food samples, it is necessary to drive an ultrasonic transducer to the desired frequency mode and to a high-power level. This may lead the transducer to work in a nonlinear regime, appearing effects like hysteretic response, frequency shifts and drops, or multimodal responses among others [5, 6].

The aim target in the design of this kind of ultrasonic transducers is to minimize those nonlinear effects by applying some physical changes in the flange [7] or other parts of the transducer.

The goal of this work is to compare the performance of a high-power ultrasonic transducer for different designs of mechanical amplifier. All the analyses have been done by applying the finite element method.

# 2 Transducer design

In order to study and determine the influence of the design of the mechanical amplifier, a high-power ultrasonic transducer has been design, taking into account the work of [8] for the Langevin type transducer and [9, 10] for the rectangular plate radiator.

# 3 Mechanical amplifiers

There have been five different designs of mechanical amplifiers, besides the one taken as reference, which is the one with circular section. All the considered sections are shown in figure 1:
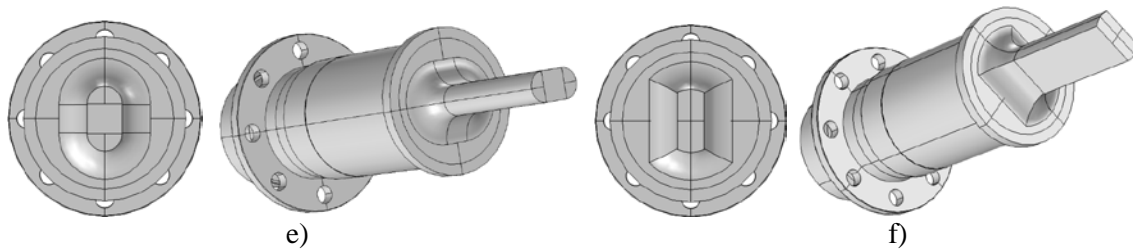


a)

b)

c)

d)
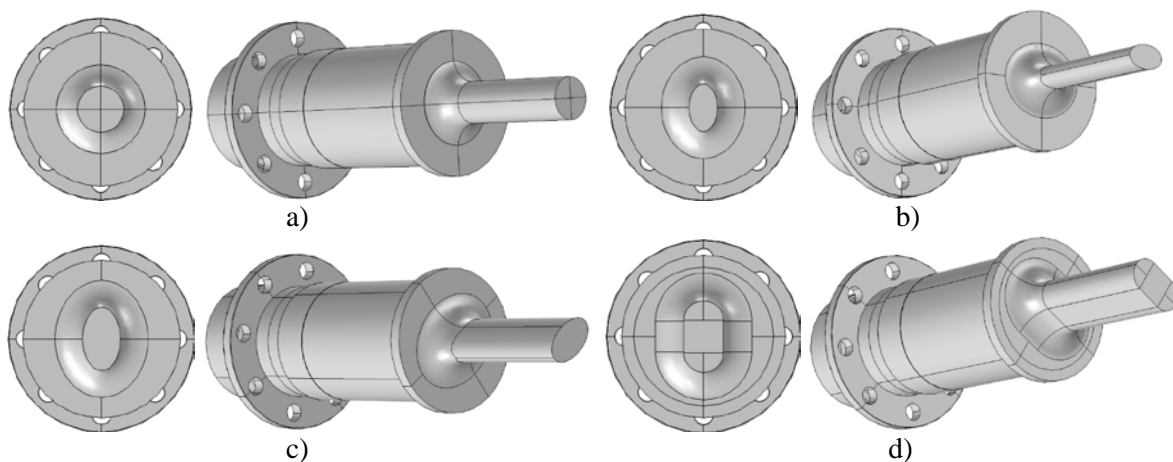
Figure 1– Different sections of the mechanical amplifier. a) Reference circular section. b) Section 1.
c) Section 2. d) Section 3. e) Section 4. f) Section 5.

The analysis comprises five different sections of mechanical amplifiers and different situations for each one. Among these sections there are two elliptical, two rounded and a quasi-rectangular section with edges. All these sections with their characteristics are shown in Table 1:

Table 1 – Different sections of the mechanical amplifier.

| Section | Shape | Area (mm$^2$) |
|---------|-------|---------------|
| Reference | Circular | 138.5 |
| Section 1 | Elliptic | 82.81 |
| Section 2 | Elliptic | 147.2 |
| Section 3 | Rounded | 183.4 |
| Section 4 | Rounded | 148.6 |
| Section 5 | Quasi-rectangular | 122.7 |

The behaviour analysis of the behaviour of the five sections considered has been done in two different ways: considering a system without plate radiator and considering an ultrasonic system with a plate radiator.

## 4   Systems without plate radiator

In the first case, unloaded ultrasonic transducers have been considered. According to [11], the radiation impedance of an ultrasonic transducer is directly related to the radiation surface, therefore the section of the mechanical amplifier. So, the displacement amplification should also be related to the area of the mechanical amplifier. The aim of the first case is to know how the horns behave in comparison with the case referred and considering the area of its ending.

In order to obtain those comparisons, a fixed displacement of 1 µm is applied in the axial direction to the joint between the front mass and the mechanical amplifier, as shown in figure 2. The final results will be the displacement amplification and the maximum mechanical stress for each mechanical amplifier vibrating at its resonance frequency. In the Table 2, the final results obtained after these simulations are indicated.
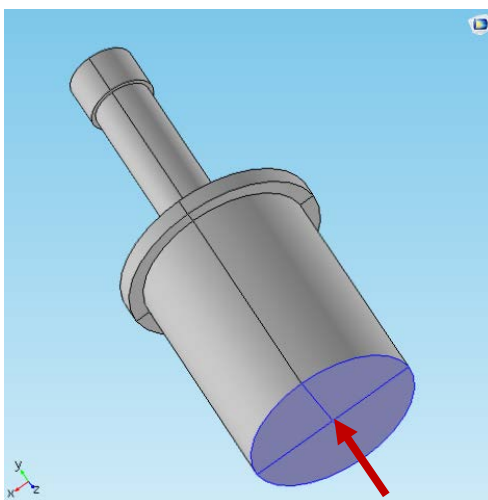
Figure 2– Placement and direction of the displacement applied in the rear face of the mechanical amplifier.

Table 2 – Results obtained for unloaded mechanical amplifiers.

| Section | Area (mm²) | Frequency (Hz) | Amplification | Maximum stress (MPa) |
|---------|-----------|----------------|---------------|----------------------|
| Reference | 138.5 | 23304 | 13 | 53 |
| Section 1 | 82.81 | 23247 | 22 | 100 |
| Section 2 | 147.2 | 23337 | 12 | 60 |
| Section 3 | 183.4 | 23465 | 10 | 44 |
| Section 4 | 148.6 | 23382 | 12 | 53 |
| Section 5 | 122.7 | 23412 | 15 | 69 |

As shown in Table 2, the mechanical amplifier with section 1, elliptical, is the one with a higher amplification ratio, but also the one with higher mechanical stress, produced in the join between the two sections of the horn. In figure 3, the behaviour of this mechanical amplifier is shown:



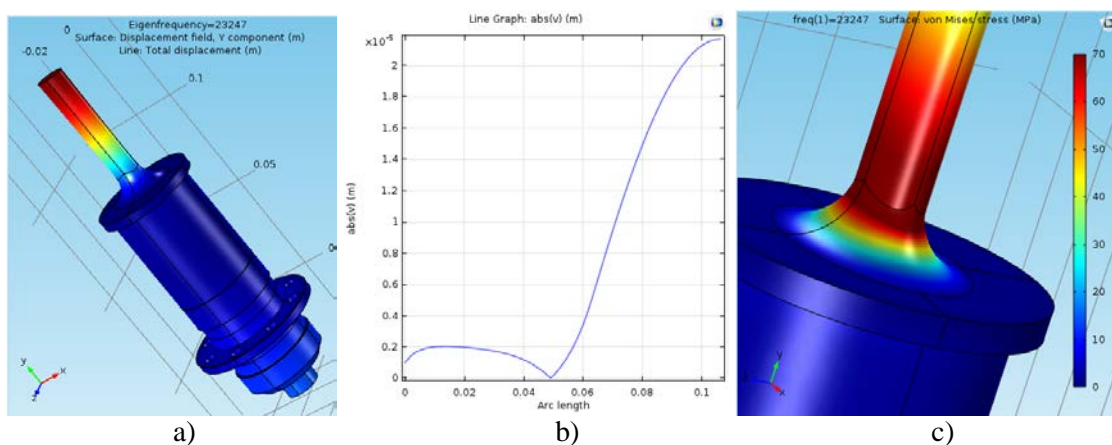a)                              b)                              c)

Figure 3– Behaviour of the mechanical amplifier with section 2, elliptic shape. a) Axial mode of the transducer. b) Displacement amplification in the axis of the horn. c) Maximum mechanical stress.

The results shown in Table 2 may confirm the dependence of the amplifier's performance with the area of its section. The mechanical amplifier with section 1 provides the highest displacement ratio and mechanical stress with the smallest area. On the other side, the horn with section 3 (rounded shape) has the biggest area and provides the lowest displacement and stress. In order to minimize the influence of the area, a normalized comparison has been considered, as shown in Table 3:

Table 3 – Normalized results obtained for unloaded mechanical amplifiers.

| Section | Area (mm$^2$) | Frequency (Hz) | Amplification | Maximum stress (MPa) |
|---------|---------------|----------------|---------------|----------------------|
| Reference | 138.5 | 23304 | 13.0 | 53,0 |
| Section 1 | 82.81 | 23247 | 13.2 | 59.8 |
| Section 2 | 147.2 | 23337 | 12.8 | 63.8 |
| Section 3 | 183.4 | 23465 | 13.2 | 58.3 |
| Section 4 | 148.6 | 23382 | 12.9 | 56.9 |
| Section 5 | 122.7 | 23412 | 13.3 | 61.1 |

All sections provide similar displacement amplification values, with ratios between 12.8 and 13.3 times the displacement in the mass of the transducer. On the other hand, the reference mechanical amplifier is the one that experiments the lowest values for the stress. However, here it can be stated that mainly is the area of the mechanical amplifier's tip the parameter that determine the behaviour of the unloaded horn.

# 5    Systems with plate radiator

When a rectangular plate radiator is added to the transducer, the radiation impedance depends only on the characteristics of the plate and not on the mechanical amplifier. In this case, the objective of this second part is to determine how the shape of the horn affects to the vibration of the rectangular plate. As well as in the previous situation, the simulation consists on a fixed 1µm axial displacement in the joint between the front mass and the horn at the desired frequency, as shown in figure 2, in order to get an axial mode of the vibrator and flexural mode of the rectangular plate with 12 nodal lines (12NL) parallels to the shorter side of the plate, as shown in figure 4:
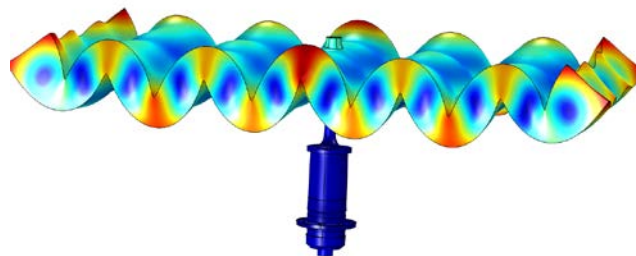


Figure 4– Desired vibration mode of the transducer for the mechanical amplifier with circular section.

## 5.1    Configurations and results

For each of the sections studied, several conditions have been taken into account, on one side, that the rectangular plate radiator and the mechanical amplifier are joined by one or two bolts, and on the other side, the relative position between the long side of the plate and the long diameter of the horn. Figure 5 shows these options for the Section 5 transducer:
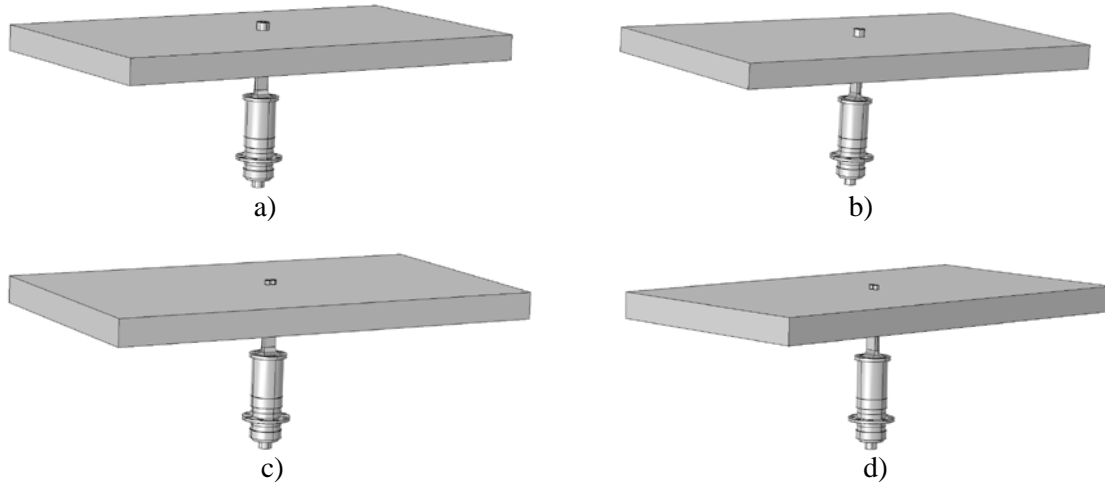


a)    b)

c)    d)

Figure 5– a) One bolt, and a parallel relative position between the long side of the plate and the long diameter of the horn. b) One bolt, perpendicular relative position. c) Two bolts, parallel relative position. b) Two bolts, perpendicular relative position.

The results shown in Table 4 indicate the maximum displacement obtained in the rectangular radiator and the maximum stress in the mechanical amplifier. Here the radiation impedance is similar in all cases as the radiator is the same.

Table 4 – Results obtained for transducers with rectangular plate radiator.

| Section | Configuration | | Frequency (Hz) | Maximum plate displacement (µm) | Maximum stress (MPa) | | |
|---|---|---|---|---|---|---|---|
| | | | | | Plate | Amplifier | Bolt |
| Reference | | | 21152 | 23.4 | 57.5 | 62.3 | 291.1 |
| Section 1 | One bolt | Parallel | 21149 | 34.9 | 84.8 | 79.3 | 466.5 |
| | | Perpendicular | 21146 | 33.7 | 81.6 | 80.0 | 496.7 |
| | Two bolts | Parallel | 21162 | 42.8 | 106.2 | 90.6 | 682.2 |
| | | Perpendicular | 21161 | 40.5 | 100.8 | 79.3 | 707.0 |
| Section 2 | One bolt | Parallel | 21155 | 24.1 | 58.9 | 48.9 | 270.9 |
| | | Perpendicular | 21152 | 22.0 | 53.2 | 48.4 | 255.1 |
| | Two bolts | Parallel | 21167 | 27.4 | 68.2 | 59.3 | 411.3 |
| | | Perpendicular | 21165 | 25.1 | 62.5 | 49.3 | 424.02 |

| Section | Configuration | | Frequency (Hz) | Maximum plate displacement (µm) | Maximum stress (MPa) | | |
|---|---|---|---|---|---|---|---|
| | | | | | Plate | Amplifier | Bolt |
| Section 3 | One bolt | Parallel | 21158 | 20.7 | 51.0 | 36.4 | 226.6 |
| | | Perpendicular | 21159 | 18.8 | 45.6 | 36.5 | 215.8 |
| | Two bolts | Parallel | 21169 | 23.4 | 58.5 | 43.3 | 339.8 |
| | | Perpendicular | 21167 | 21.4 | 53.3 | 38.2 | 335.3 |
| Section 4 | One bolt | Parallel | 21155 | 24.1 | 58.8 | 44.5 | 270.3 |
| | | Perpendicular | 21151 | 21.9 | 53.4 | 43.5 | 267.9 |
| | Two bolts | Parallel | 21167 | 27.3 | 67.8 | 45.3 | 392.2 |
| | | Perpendicular | 21164 | 25.0 | 62.2 | 47.6 | 419.2 |
| Section 5 | One bolt | Parallel | 21153 | 28.1 | 68.6 | 56.6 | 300.6 |
| | | Perpendicular | 21149 | 25.5 | 62.5 | 56.2 | 360.8 |
| | Two bolts | Parallel | 21165 | 32.0 | 79.9 | 56.2 | 494.8 |
| | | Perpendicular | 21163 | 28.8 | 71.7 | 55.9 | 473.1 |

Several conclusions can be obtained from Table 4. Firstly, the use of two bolts to hold the rectangular plate and the mechanical amplifier provides a higher resonance frequency and displacement of the plate, but also a much higher stress in the bolts (about 50% higher in most of the cases), as can be seen in figure 6:
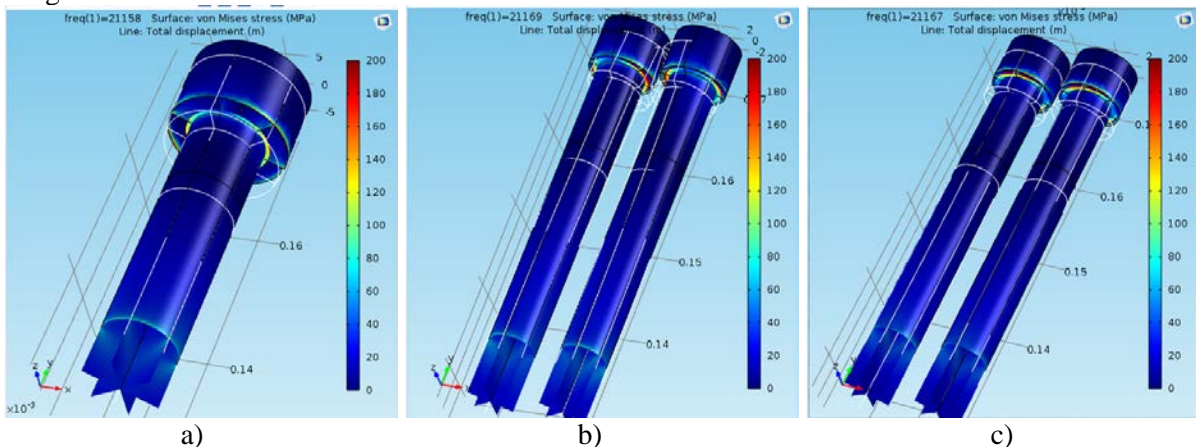


Figure 6– Stress in bolts for Section 3, parallel. a) One bolt. b) Two bolts. c) Two bolts, perpendicular.

No noticiable differences have been observed regarding the resonance frequency in the five sections considered, there are minor variations of the different mechanical amplifiers.

On the other hand, regarding the maximum displacement and stress obtained in the plate and in the mechanical amplifier, the results obtained show a similar behaviour as the model without plate radiator. The influence of the chosen mechanical amplifier is high and the simulated performance of each transducer depends on the section considered.

Further quantitative analysis of the obtained results may be done by normalizing the values of stress and plate displacement. In figure 7, all normalized results can be observed, as well as the maximum displacement in the plate:
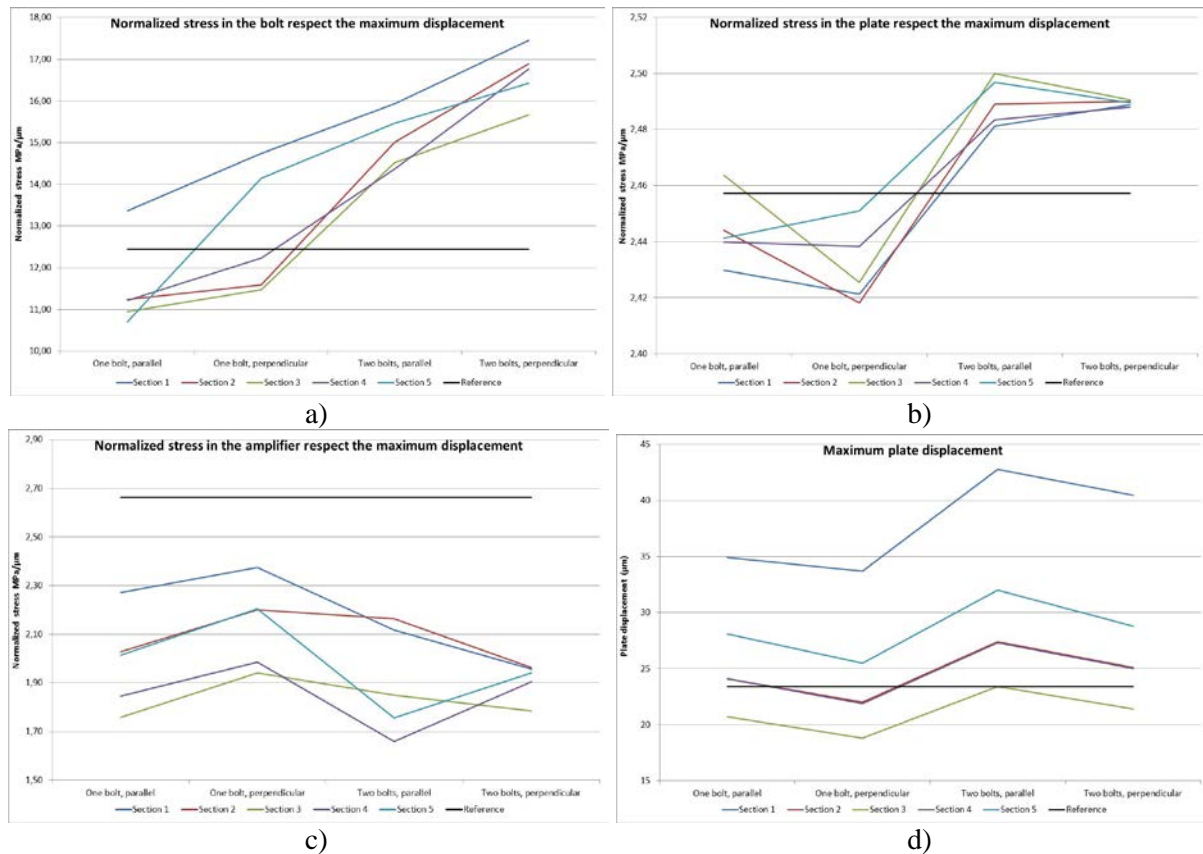


Figure 7– Quantitative analysis of the behaviour of different transducers with different mechanical amplifiers. a) Normalized stress in the bolts respect the maximum displacement in the plate. b) Normalized stress in the plate respect the maximum displacement. c) Normalized stress in the mechanical amplifier respect the maximum displacement in the plate. d) Maximum displacement in the plate

In figure 7 it can be observed that, although Section 3 provides the lower maximum displacement of the plate – but near the values obtained by the reference mechanical amplifier-, it also shows the lowest relation between stress and displacement, meaning that each µm displacement requires less MPa in the bolts and in the horn.

Finally, the numerical analysis doesn't indicate substantial differences between a parallel or perpendicular position between the horn and the plate. The charts in figure 7 may point to the fact that parallel direction provides a slightly higher displacement with a lower stress in the bolts and in the horn, but higher stress in the plate.

In figure 8 is represented the displacement of the rectangular radiator along one of the long edges for each section, considering one bolt and a parallel direction between the mechanical amplifier and the plate.
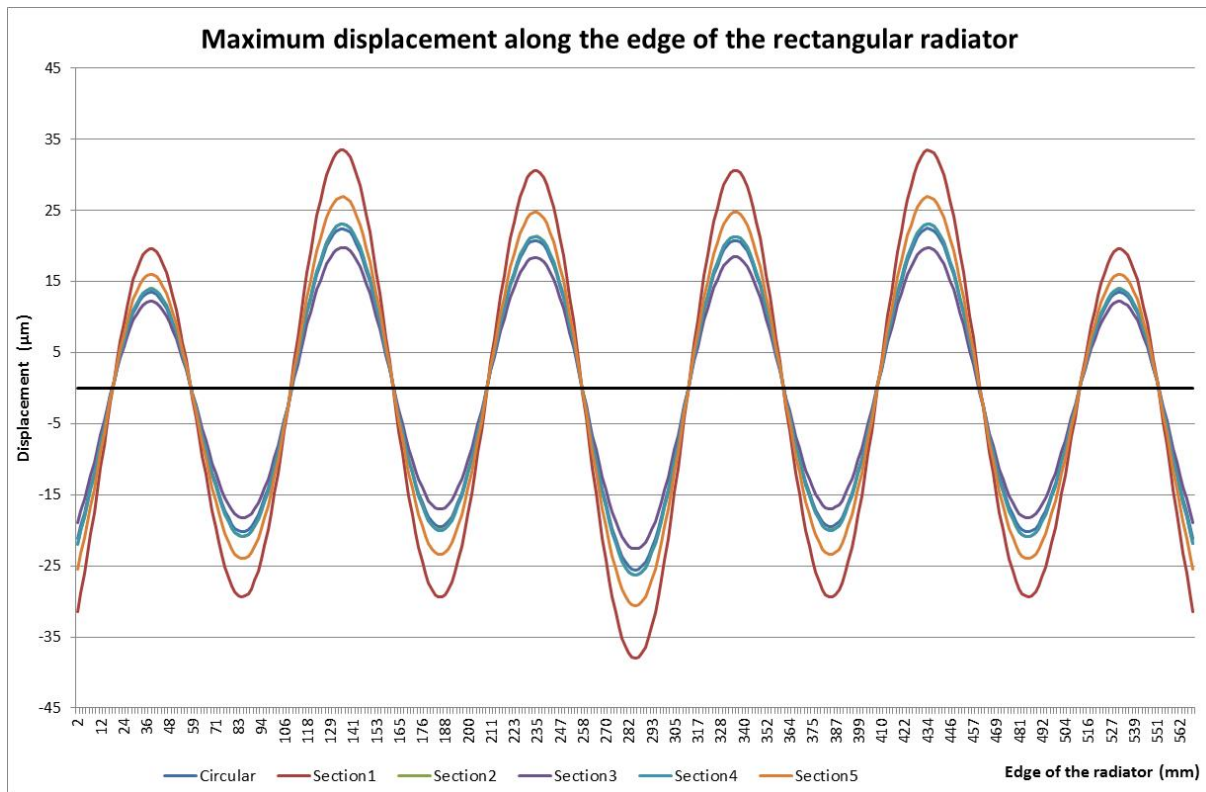
Figure 8– Displacement of the long edge of the rectangular radiator for each section, considering just one bolt and a relative parallel position between the long side of the plate and the long diameter of the mechanical amplifier.

The twelve nodal lines of all the configurations can be easily observed in figure 8. The choice of mechanical amplifier affects on the displacement obtained but not in the vibration shape of the radiator (shown in figure 3).

# 6  Conclusion and future research lines

The main aim of this work is to determine by numerical methods, the general behaviour of different designs of mechanical amplifiers, trying to avoid the existence of longitudinal edges (except for the Section 5) and to have similar areas in the horn's tips.

According to the obtained results, all the mechanical amplifiers show similar performances although the circular horn seems to provide the lowest mechanical stress for a similar displacement amplification when working without load.

When the mechanical amplifiers are loaded with a rectangular plate radiator, the behaviour of the whole transducers also depend on the section of the horns, providing a higher displacement when the area is smaller but also suffering from higher stress.

This work has a clear continuation by the experimental validation of the different mechanical amplifiers analysed, whose principal goals should be reducing mechanical stresses in contact zones between the horn and the plate radiator, keeping a high displacement amplification in the tip.

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