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TOPOLOGICAL OPTIMIZATION AND VIBRATION ANALYSIS OF UPPER CONTROL ARM

A. Boualaoui [1] **D. Sarsri** [1] **M. Lamrhari** [2]

1. Electrical and Industrial Engineering Department, National School of Applied Sciences, Abdelmalek Essaadi University, Tangier, Morocco, abderrazak.boualaoui@etu.uae.ac.ma, dsarsri@uae.ac.ma
2. National School of Arts and Crafts, Moulay Ismail University, Meknes, Morocco, lamrha@yahoo.fr

Abstract- Designing suspension systems for cars has always been a challenge due to the loads and requirements for durability and reliability. However, with simulation tools like topological optimization, it is possible to design improved products. This article presents a study on designing the upper triangle of a car using a topological optimization approach. The primary objectives are to minimize volume and compliance while maximizing the fundamental natural eigenvalue of the mechanical part during free vibrations. Achieving these goals involves integrating optimization modules, material interpolation models, filters, sensitivities, and a solver. The optimization solutions are then analyzed. For the upper control arm of the suspension, the aim is to maximize the first natural frequency as much as possible to avoid vibrations while ensuring a robust part. The use of new materials allows for adapting additive manufacturing as a production process to fully benefit from topological optimization. The MMA (Method of Moving Asymptotes) is used as topological optimization for eigenvalues. Depending on the parameters used, various optimization solutions are explored and analyzed.

Keywords: Topological Optimization, Method of Moving Asymptotes (MMA), Natural Frequency, Fatigue.

1. INTRODUCTION

Simulation is an essential tool for better understanding the outcomes of design decisions, significantly reducing design process failures [1]. Topological optimization plays a crucial role in creating sturdy and lightweight parts that can be produced through additive manufacturing despite their complexity. Moreover, the introduction of topological optimization expands the field for 3D printing to compete with conventional processes. Solving passive design issues like vibrations and noise is vital in engineering. A common goal is to shift a structure's natural frequencies away from external excitation to avoid resonance [2]. Topology optimization based on eigenvalues is a critical approach aimed at adjusting natural frequencies to eliminate any resonance during design [3-4]. It's noted that a higher first natural frequency results in a stiffer structure [5].

This optimization has various applications, including designing resonators for vibration absorption [6-7-8] and comparing different vibration control strategies as developed in previous works [9]. Tsanev used modelling to determine the optimal natural frequency [10]. On the other hand, Rostami, et al. discussed the maximization of natural frequencies under various constraints using derivative-free optimization of continuous structures [11]. By employing an optimization model, Wang, et al. developed maximum eigenvalues of the first order with a constraint on the total volume [12]. Furthermore, the maximization of the fundamental frequency through topological optimization of the coupled-constraint continuum has been explored in the literature [13]. Topology optimization has also been studied to maximize the first eigenfrequency using the topology optimization formulation by constraining the couple of continuous structures [14]. Most studies have used minimizing compliance under static loads as a criterion. However, structures are generally under dynamic loads, making natural frequency a critical design criterion not to be overlooked. Furthermore, in the literature, there has not been enough emphasis on the design domain (optimization boundaries) and their effect, especially on natural frequencies.

In this article, we present the topological optimization of an upper control arm to maximize the first natural frequency. To combine usefulness with pleasure, we have chosen new materials suitable for additive manufacturing and taken advantage of topological optimization. The focus is on stainless steel 316L and aluminum AlSi10Mg to replace the material of the initial design (AISI 1045). The challenge of optimizing the topology to maximize eigenvalues is approached using the MMA (Method of Moving Asymptotes), known as the fastest solver. Achieving an optimal solution at a lower cost involves several parameters, requiring a more rigorous study [15] or the use of artificial intelligence [16].

2. MATERIALS AND METHODS

2.1. General Optimization Model with Maximization of the Fundamental Natural Frequency

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Finite element analysis allows us to represent the dynamic behavior of the continuous structure by the general problem of eigenvalues:

$$([K]-\omega_i^2[M])\{u_i\}=\{0\} \tag{1}$$

where, u_i is the eigenvector and ω_i is the ith corresponding natural frequency. Therefore, the optimization of the eigenvalues can be can be written as a max-min. The approach is shown in Figure 1.

$$\max\left\{\min i_{i=1\ldots N}\left\{\omega_i^2\right\}\right\} \tag{2}$$

$$st[K]\{u_i\}=\omega_i^2[M]\{u_i\}, i=1\ldots N \tag{2a}$$

$$\{u_i\}^T[M]\{u_j\}=\delta_{ij}, i\geq j, ij=1\ldots N \tag{2b}$$

$$\sum_{e=1}^{n}x_e-fV_0\leq 0 \tag{2c}$$

$$0<X_{\min}\leq x\leq 1 \tag{2d}$$

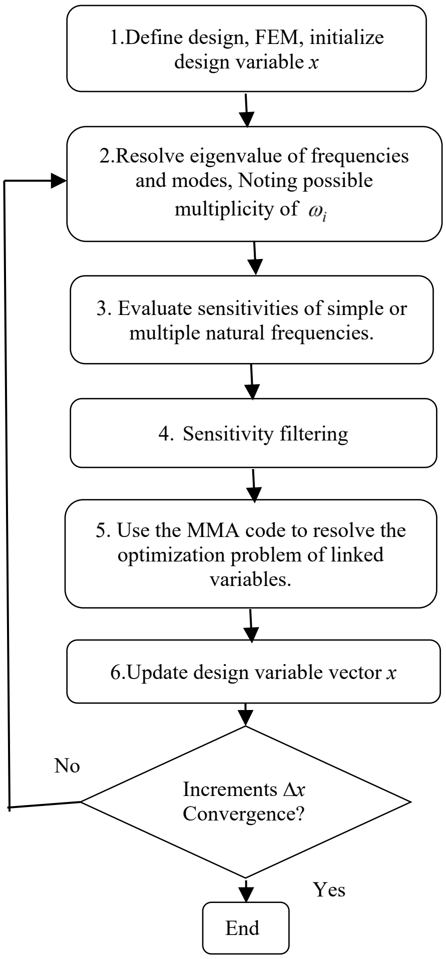


Figure 1. Problem methodology flowchart

2.2. Material Interpolation Models

The pseudo density ρ of element e represents the presence or absence of solid material (0 or 1) in topological optimization. However, with the two interpolation models SIMP and RAMP, ρ can take values from 0 to 1. These two models are introduced in this work to observe the effect on maximizing the natural frequencies in MMA optimization.

2.2.1. SIMP Model

The interpolation schemes are shown in Figure 2 [17].

$$E=\rho^p E_0 \tag{3}$$

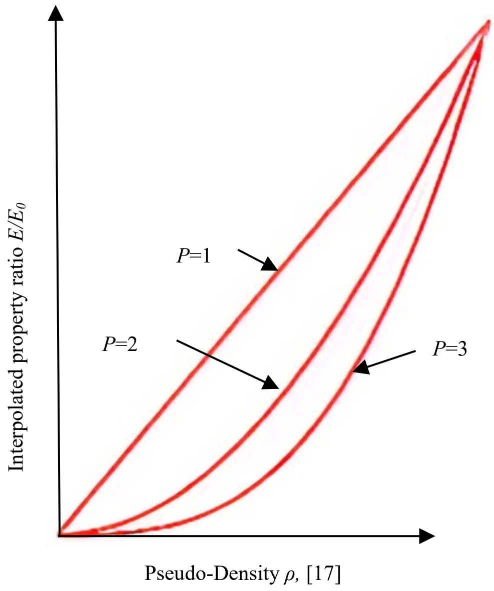


Figure 2. SIMP Model for different penalization factors p [17]

2.2.2. RAMP Model

The interpolation schemes are shown in Figure 3, [17].

$$E=\frac{\rho}{1+p(1-\rho)}E_0 \tag{4}$$

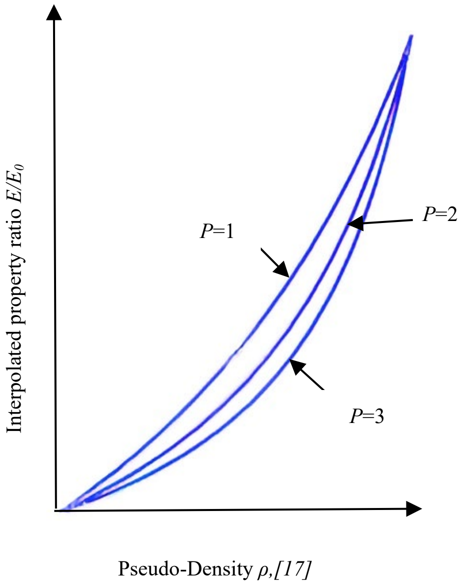


Figure 3. RAMP Model for different penalization factors p [17]

2.3. Case studies: The Upper Control Arm of Cars

The upper control arm is located on the suspension system as shown in Figure 4 [18]. This system absorbs enormous vibrations. It ensures good stability of the car, passenger comfort and good steering maneuverability.

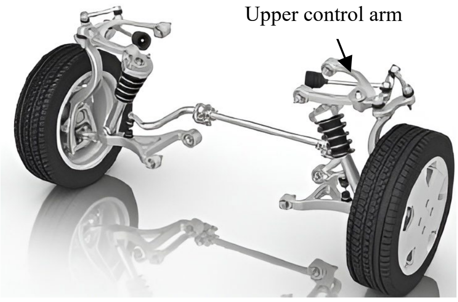


Figure 4. Suspension system [18]

Control arms resist a multitude of loading forces, such as acceleration or braking, cornering while turning, and the hanging weight of the vehicle body. They also have the added function of maintaining dynamic wheel alignment. This helps reduce transmitted noise, shock and road vibration while providing resistance to unwanted suspension movement. The literature [19-20] is demonstrated that combustion in diesel engines can generate vibrations varying from 0.3 to 1.5 kHz. The Figure 5 shows the initial upper arm made of AISI 1045 whose first natural frequency is 532.79Hz and the mass is 1.2 Kg.

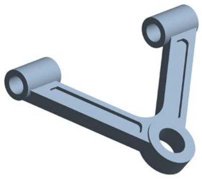


Figure 5. Initial design made of AISI 1045

The maximization of the natural frequencies was computed in Comsol Multiphysics software. To adapt our product for additive manufacturing and enhance the solutions, the choice has been made for the following two materials: Stainless Steel 316L and aluminum AIsi10Mg. Table 1 indicates the properties of the two materials.

Table 1. Material properties

Material	Stainless steel 316L	Aisi10Mg
Young's Modulus (GPa)	180	70
Density (Kg/m^3)	8000	2650
Poisson's Ratio	0.27	0.33

The initial design is potentially exposed to resonance frequencies. So, the aim of optimization is to increase the first natural frequency as much as possible through a new design and new materials suitable for metal 3D printing.

2.3.1. Non-Optimized Design Domain and Zones for Case A and Case B

The choice of admissible design domain is focused on two cases of non-optimized zones as indicated in Figures 6 and 7.

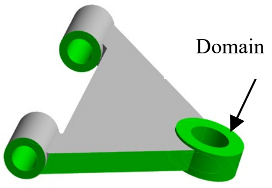


Figure 6. Case A: Domain and boundaries not optimized

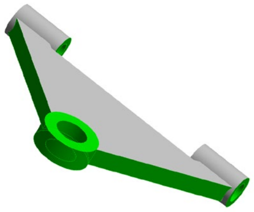


Figure 7. Case B: Non-optimized boundaries

2.3.2. Boundary Conditions and Loading

Figure 8 shows the fixed support at the level of the two coaxial holes. The force of 5500N is equivalent to a quarter of the weight of the car.

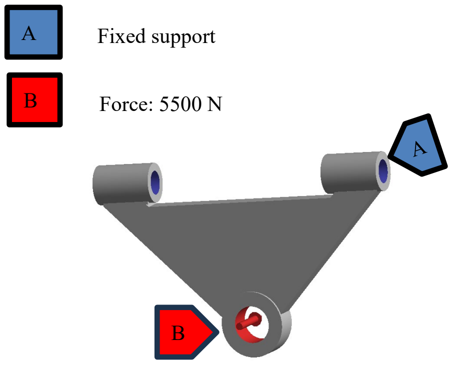


Figure 8. Boundary conditions

2.3.3. Selection of *p* for SIMP

Following Figure 2 and Equation (3), if $p = 1$, there will be no penalizing effect on the material (linear curve). In the case where $1<p<2$ the penalizing effect is insufficient. So, we take $p>2$. Furthermore, the RAMP curve (Figure 3, Equation (4)) is similar to that of SIMP, but the RAMP model provides more stability if p is high and RAMP model. As a result, p=3 is chosen for SIMP and p=4 for RAMP.

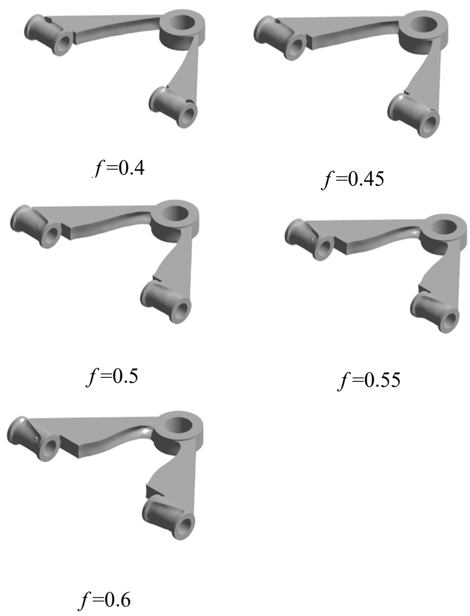
3. OPTIMIZATION RESULTS

3.1. Case A

3.1.1. 316L Stainless Steel (SIMP Interpolation, *p*=3)

In Figures 9, we represent the different optimization results depending on the volume fraction *f*.

Table 2 shows the first natural frequency for each volume fraction. The first maximum natural frequency is 780.06Hz for a volume fraction of 0.6.



f=0.4 f=0.45

f=0.5 f=0.55

f=0.6

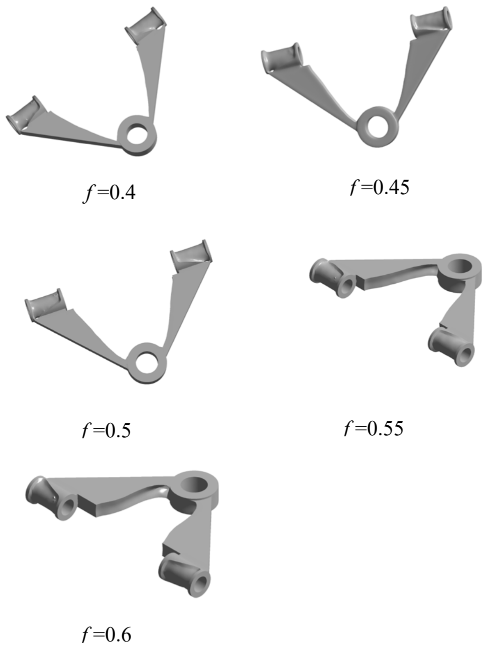
Figure 9. Optimizations based on volume fractions (316L, pSIMP=3)

Table 2: The first natural frequency F_1 and the volume fraction f

f	0.4	0.45	0.5	0.55	0.6
F_1 (Hz)	650.26	706.77	744.95	768.26	780.06

3.1.2. ALsi10Mg (SIMP Interpolation, *p*=3)

In Figures 10, we represent the different optimization results depending on the volume fraction *f*.



f=0.4 f=0.45

f=0.5 f=0.55

f=0.6

Figure 10. Optimizations based on volume fractions (Alsi10Mg, pSIMP=3)

According to Table 3, the first maximum natural frequency is 846.75Hz for a volume fraction of 0.6.

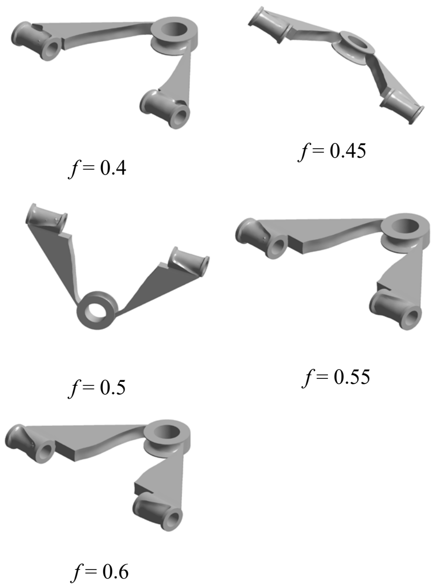
Table 3. The first natural frequency F_1 and the volume fraction f

f	0.4	0.45	0.5	0.55	0.6
F_1 (Hz)	704.94	766.47	808.21	833.77	846.75

3.2. Case B

3.2.1. Alsi10Mg (RAMP Interpolation, *p*=4)

In Figures 11, we represent the different optimization results depending on the volume fraction *f*.



f= 0.4 f= 0.45

f= 0.5 f= 0.55

f= 0.6

Figure 11. Optimizations based on volume fractions (Alsi10Mg, pRAMP=4)

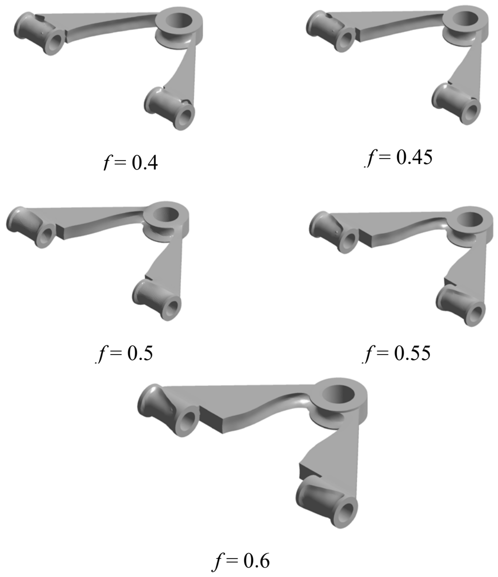
Table 4 shows the first natural frequency for each volume fraction. The first maximum natural frequency is 928.78Hz for a volume fraction of 0.6.

Table 4. The first natural frequency F_1 and the volume fraction f

f	0.4	0.45	0.5	0.55	0.6
F_1 (Hz)	746.61	828.39	883.22	914.72	928.78

3.2.2. 316L Stainless Steel (SIMP Interpolation, *p*=3)

In Figures 12, we represent the different optimization results depending on the volume fraction *f*.



f= 0.4 f= 0.45

f= 0.5 f= 0.55

f= 0.6

Figure 12. Optimizations based on volume fractions (316L, pSIMP=3)

According to Table 5, the first maximum natural frequency is 850.57Hz for a volume fraction of 0.6.

Table 5. The first natural frequency F_1 and the volume fraction f

f	0.4	0.45	0.5	0.55	0.6
F_1 (Hz)	682.36	758.86	809.5	838.08	850.57

4. DISCUSSION

Liu, et al. [21] discussed the impact of parameters such as mass fraction, penalization factor and beta projection on the structure through MMA topological optimization. In this work, the design domain (design boundaries) was added to observe its effect on maximizing the first natural frequency. Figure 13 summarizes the results obtained for both cases A and B.

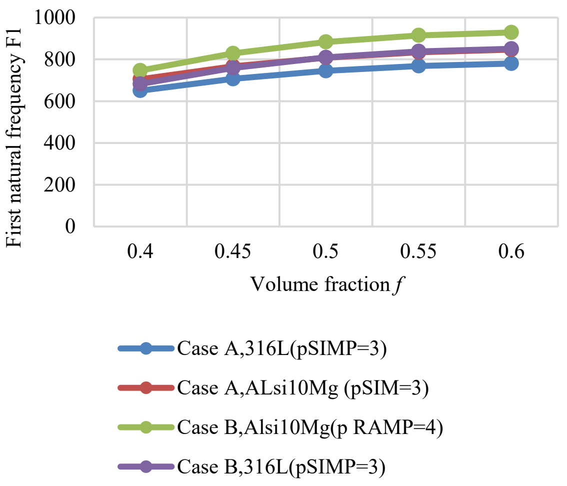


Figure 13. Summary curves

See the different solutions, the choice focuses on case A (ALsi10Mg with pSIMP=3 and volume fraction of 0.45) and case B (316L stainless steel with pSIMP=3, volume fraction of 0.5) which respectively give a first natural frequency of 766.46Hz and 809.5Hz with maximum material removed. Without forgetting that to maximize the frequencies we increase the rigidity of the part.

Note that by changing the non-optimized domains, we change the topology of the upper arm. Furthermore, RAMP interpolation with p=4 gives almost the same topology but with a slight increase in natural frequencies. To decide between the two solutions, we used a static study on Ansys. Then we obtain the following results:

4.1. Case A (ALsi10Mg with pSIMP=3 and Volume Fraction 0.45)

According to Figures 14, 15, 16 and 17, we see that the stress, the total displacement and the life are acceptable. The final mass is 0.33 Kg.

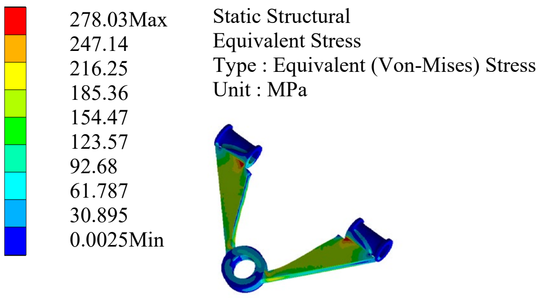


Figure 14. Equivalent Stress (Case A, ALsi10Mg; pSIMP=3)

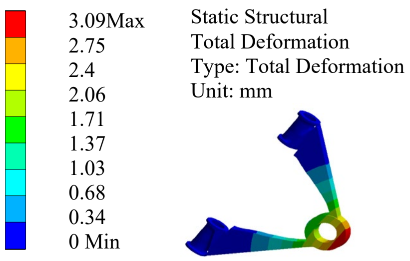


Figure 15. Total Deformation (Case A, ALsi10Mg; pSIMP=3)

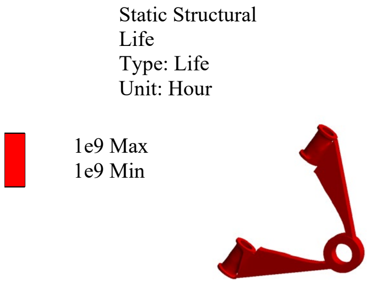


Figure 16. Life (Case A, ALsi10Mg; pSIMP=3)

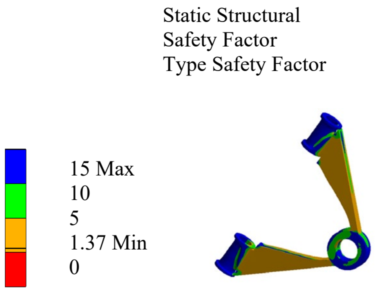


Figure 17. Safety Factor (Case A, ALsi10Mg; pSIMP=3)

4.2. Case B (316L Stainless STEEL with pSIMP=3, volume Fraction 0.5)

According to Figures 18 to 21, we see that the stress, the total displacement and the life are acceptable. The final mass is 1.022Kg.

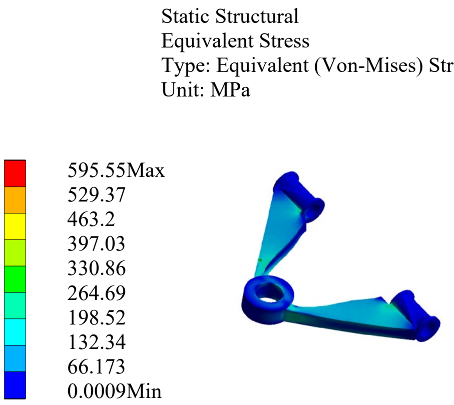


Figure 18. Equivalent Stress (Case B, 316L, pSIMP=3)

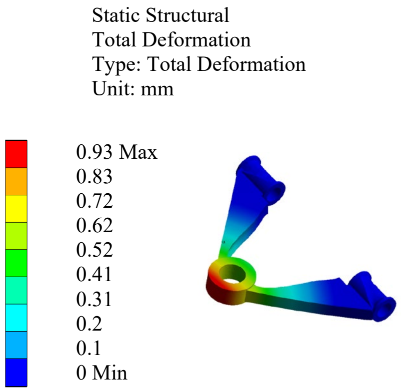


Figure 19. Total Deformation (Case B, 316L, pSIMP=3)

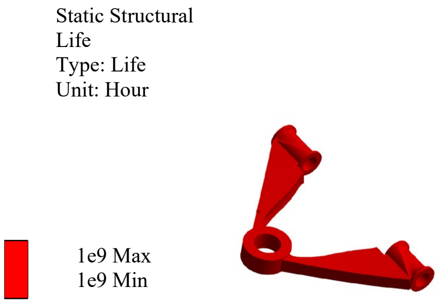


Figure 20. Life (Case B, 316L, pSIMP=3)

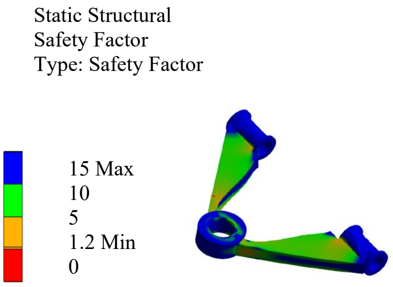


Figure 21. Safety Factor (Case B, 316L, pSIMP=3)

So, with this solution we can replace the initial triangle while adapting it to additive manufacturing. Thus, to have an optimal solution at a lower cost, several parameters come into play. Hence the need for a more rigorous study.

5. CONCLUSIONS

Topological optimization is a crucial tool in selecting the optimal solution for structural designs. Specifically, optimizing natural frequencies allows us to address resonance issues. In this work, the MMA method is used to move the fundamental frequency away from the vibration zone. To maximize the first natural frequency, we have chosen the case study of the upper control arm of a car suspension. This component indeed experiences vibrations from the engine block. To replace the initial forged solution, we opted for new materials.

The selection of 316L stainless steel and ALsi10Mg aluminum ensures that additive manufacturing is suitable as an alternative to conventional processes (forging). Several solutions are analyzed based on volume fractions, penalty factors (SIMP and RAMP models), and optimization boundaries. The results demonstrate that this optimization technique effectively separates the fundamental frequency from the resonance zone, thereby preventing high vibration levels. Therefore, topology optimization provides engineers with the opportunity to design robust components while avoiding risky frequency ranges. This approach can be applied to similar parts, particularly aimed at maximizing natural frequencies and expanding the scope of additive manufacturing.

NOMENCLATURES

1. Acronyms

MMA	Method of Moving Asymptotes
SIMP	Solid Isotropic Material with Penalization
RAMP	Rational Approximation of Material Properties
FEM	Finite element method

2. Symbols / Parameters

K : global stiffness matrix

ω_i : ith natural frequency

$[M]$: global mass matrix

$\{u_i\}$: ith eigenvector

$\{u_j\}$: jth eigenvector

N : Degrees of freedom in the admissible design domain

f : volume fraction

V_0 : Volume of the design domain

x : the vector of elementary densities of the material

$X_{\min}$: a limit vector lower for x

ρ : Pseudo density

E : Young's modulus, $[Pa]$

E_0 : Element Young's modulus, base material, $[Pa]$

p : Penalization factor ($p \geq 3$)

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BIOGRAPHIES

Name: **Abderrazak**
Surname: **Boualaoui**
Birthday: 26.05.1973
Birthplace: K. Tadla, Morocco
Bachelor: Mechanical Engineering, ENSAM, Mohammed V University, Rabat, Morocco, 1998
Master: Mechanical Engineering, ENSAM, Mohammed V University, Rabat, Morocco, 2020
Doctorate: Student, Topological Optimization, Electrical and Industrial Engineering Department, Sciences and Technologies Faculty, Abdelmalek Essaadi University, Tangier, Morocco, Since 2021
The Last Scientific Position: Teacher, Industrial Engineering Sciences, CPGE Mohamed VI, Kenitra, Morocco, Since 2014
Research Interests: Topology Optimization, Additive Manufacturing, Vibration of Structures
Scientific Publications: 1 Paper

Name: **Driss**
Surname: **Sarsri**
Birthday: 01.01.1971
Birthplace: Tetouan, Morocco
Bachelor: Mechanical Engineering, ENSAM, Mohammed V University, Rabat, Morocco, 1994
Master: Thermal-Energetic-Mechanical, Faculty of Sciences, Sidi Mohamed Ben Abdellah University, Tetouan, Morocco, 2001
Doctorate: Mechanic Stochastic Analysis of Structures Using Modal Synthesis Methods, Mechanical Engineering Department, Sciences and Technologies Faculty, Abdelmalek Essaadi University, Tangier, Morocco, 2004
The Last Scientific Position: Prof., Electrical and Industrial Engineering Department, Sciences and Technologies Faculty, Abdelmalek Essaadi University, Tangier, Morocco, Since 2010
Research Interests: Mechanical Engineering, Structural Engineering, Automotive Engineering
Scientific Publications: 35 Papers

Name: **Mohammed**
Surname: **Lamrhari**
Birthday: 03.07.1971
Birthplace: Sefrou, Morocco
Bachelor: Mechanical Engineering, ENSAM, Mohammed V University, Rabat, Morocco, 1993
Master: Mechanical Engineering and Industrial Maintenance, Mohammedia School of Engineers (EMI), Mohammed V University, Rabat, Morocco, 2002
Doctorate: Ph.D., Physics, Mechanical Engineering and Structure "Vibrations and Reliability of Nonlinear Structures Assembled and with Random Parameters", Physics Department, Faculty of Sciences, Moulay Ismail University, Meknes, Morocco, 2019
The Last Scientific Position: Prof., Mechanical and Structural Engineering Department, ENSAM, Moulay Ismail University, Meknes, Morocco, 2019
Research Interests: Vibration, Additive Manufacturing, Wind Energy
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