Displacement amplifier mechanism for piezoelectric actuators design using SIMP topology optimization approach

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***Abstract*— Due to their inherent crystalline properties piezoelectric actuators have a limited deformation. This intrinsic drawback deprives to exploit the potential of these actuators such as, high bandwidth and high resolution in applications that require large displacement range. To overcome this limitation, classical as well as systematic approaches were proposed to design amplification mechanisms. The classical approach leads to empirical mechanisms which are not trivial and needs much experience and intuition. In contrast, systematic approach uses topology optimization method which permits to automatically derive optimal designs that can satisfy specified performances and imposed constraints simultaneously, this with a reasonable time and cost.**

This paper proposes the design of a mechanism devoted to amplify the displacement of a piezoelectric actuators (PEA). Based on the SIMP topology optimization method, the approach permits to derive a design with a displacement amplification ratio of 4.5**, which is higher than with the existing method of Rhombus mechanism. Both finite element (FE) simulation and experimental results confirm and demonstrate the efficiency of the approach.**

***Index Terms*— piezoelectric actuators, optimal design, compliant structure, SIMP topology optimization**

I. INTRODUCTION

Nowadays, the interest of piezoelectric actuators is considerable and well established [1]. As they provide high displacement resolution, large output force, high dynamic response and high energy density, they have been used in various applications where high precision is required, for instance: precise alignment, scanning force microscopy, micro/nano assembly and micro/nano manipulation and interferometry. Nevertheless, one of their main drawback is the low relative deformation restricting their range of motion. In fact, larger strain would increase the performances and range of applications of piezoelectric actuators and is therefore desirable. To deal with this limitation, several solutions have been proposed to increase the stroke of these actuators. As reviewed in [2] and [3] classical as well as systematic approaches were investigated. The trivial approach that operates on the shape, the polarity and the geometric arrangement of the actuator leads to piezostack, bimorph and rainbow configurations. In the first one, several piezoelectric layers are stacked together in serial. In this case, the output displacement is the sum of the displacement of each actuation layer. However to produce reasonable output displacement, this configuration leads to long and non-compact actuators. The second configuration combines two piezoelectric layers. It provides a larger displacement than piezostack however the provided force is low. Another classical approach employs external mechanism to magnify the deformation of the piezoelectric actuator. Known as amplifying mechanism, several structures have been proposed in the literature. The widespread one is the flexural lever structure [4]. Flextensional mechanism such as Moonie [5] [6], Cymbal [7] and nested rhombus structure [3] are also investigated. However, the design of such mechanisms is not trivial because much experience and intuition are required.

In contrast to the above mentioned approach, systematic approaches including topology optimization [8], interval [9]–[11] and blocs methods [12] can be used. Topology optimization [13] [14] [15] [8] is particularly suitable to systematize the design of such mechanisms. It aims to propose a systematic methodology for optimal designs and it can find its application in a wide variety of field, from designing a lightweight city bus, bone remodelling for prosthesis design or frame optimization design for crashworthiness of cars as presented in [8]. Considering the amplification mechanisms as monolithic compliant structures, this method can be applied directly to master the trade-off between the mass, the volume and the stiffness of the structure. The literature review shows that various studies attempted to utilize this approach to design mechanical amplifier. In 1997, Sigmund was the first to apply topology optimization to design an inverting displacement amplifier based on maximum mechanical advantage [14]. Subsequently, Canfield et al. proposed a design where the overall stroke amplification and mechanical efficiency are considered as objective functions [16]. Silva et al. applied the homogenization formulation of topology optimization to design flextensional actuator and piezoelectric transducer [17] [18]. Unlike these studies, another formulation is proposed in [2] [19] to design an amplification mechanism under dynamic motion. Other designs have been also presented in [20] [21]. Although the mentioned studies describe interesting solutions, their interpretation remains difficult and thus their application to new physical systems design is not straightforward. This is mainly due to the formulation of the topology optimization based on homogenization method and the variable thickness method which are purely mathematical and which are far to physical signification. Furthermore, the problem resolution leads often to designs that require a post-processing step to extract a realistic structure. Finally there is no study focusing on how the supports and the volume of the amplification mechanisms influence the optimal solution.

To address the first matter, we propose here to use SIMP

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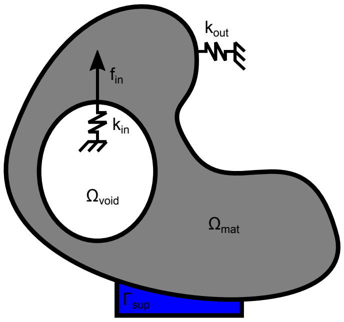

Fig. 1. Mechanical modeling of a topology optimization problem.

(Solid Isotropic Material Penalization) formulation of topology optimization to design a novel amplification mechanism. This formulation uses penalization power law to make material intermediate density unattractive and therefore avoids the 0-1 problem. Otherwise, it is mathematically well-posed and easy to implement. Indeed, a Matlab implementation including 99 code lines was proposed by Sugmund in [22]. To address the second matter we propose to substitute the OC (Optimality Criteria) method originally implemented in [22] by MMA method (Method Moving Asymptotes) in order to take into account more than one constraint. Finally once the optimization is realized under Matlab software, we propose an automatic procedure to extract the resulted structures. This step is important to convert the structures to a format compatible with CAO software's. To resume, this paper deals with the optimal design of an amplification mechanism for piezoelectric actuator. The contributions of the paper are:

- an improvement of the 99 code of topology optimization presented in [22] by using MMA method,
- an analysis of the influence of the supports and the volume as design variables on the amplification mechanism,
- an automatic extraction of the designed structure under Matlab, realization and experimental validation of the optimal amplification prototype.

The remainder of the paper is organized as follows. First, section-II provides the SIMP topology optimization method adapted to a mechanism design. In section-III, the specification of the amplification mechanism are formulated in order to further apply SIMP method. The influence of the supports and the volume on the optimal designs is analyzed in the same section. Finite element simulation is also carried-out on the optimal design. In Section-IV the realization of the amplification mechanism and its experimental characterization are presented. Discussions and comparison with the simulation results are also given. Finally section-V gives the conclusion and some perspectives.

II. TOPOLOGY OPTIMIZATION

This section provides a reminder about topology optimization. First the mechanical formulation used by this method is presented. Then a brief explanation of SIMP method and the formulation of support optimization are given. Finally, The

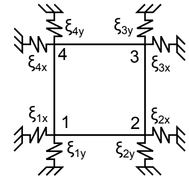

Fig. 2. Support modeling where each node have springs attached in X and Y direction.

optimization algorithm and the computation of sensitivities are presented. For more details readers can refer to [8].

A. Modeling

Consider a mechanical element occupying a domain Ω_{mat} as illustrated in Fig. 1 included in a larger reference domain Ω in R^2. This domain is modeled as a plane-stress solid. There is also a fixed domain $\Omega_{void} \subset \Omega_{mat}$ without material. Furthermore a number of supports are to be set on a fixed boundary Γ_{sup}: two fixed input ports lie on the Γ_{void} boundary and one fixed output port lies on $\Gamma_{mat} \backslash \Gamma_{sup}$ boundary. The piezoelectric actuator which is a stack (piezostack) placed at the input port is assumed to have a linear behavior. It has a lamped model characterized by a spring of stiffness k_{in}. The driving voltage will be represented by an input force f_{in}. From these parameters, the free (un-loaded) actuator displacement is given by: $u_{in} = f_{in}/k_{in}$, as demonstrated in [8] for further details.

B. Method formulation

In this paper we propose to use the SIMP method (eq. 2c-d) [8] to perform the optimization under finite element (FE) linear elasticity theory with Hooke's law (eq. 2b). The classical FE approach is used with 4-node bilinear quadrilateral elements mapping the whole design domain. The objective function is set as the maximum output displacement u_{out}, reformulated as a minimization (eq. 2a), with a constraint on the maximum volume (eq. 2-f). In these equations, ρ, K and K_e are the density, the global and the element wise stiffness matrix respectively; U and F are the displacement and the force vector respectively; N is the total number of element of the mesh, ρ_e is the element wise density, p is the penalization factor, and finally ρ_{min} stands for the minimum bound for the density and V_{frac} the normalized maximum allowed volume between 0% and 100%. Regarding the optimization of supports, the formulation of [8] is used.

To summarize, first linear springs are added on both x and y direction at each node of the elements in Ω_{sup} (Fig. 2). The springs stiffnesses are set to be larger than the diagonal elements of the K_e matrix, typically ten times. So we introduce a diagonal element support stiffness matrix $K_{e,sup}$ as follows:

$$K_{e,sup} = 10 \ \text{diag}(K_e) \tag{1}$$

The same penalization rule is applied to the springs with variables ξ_e (eq. 2c-e). This means that near $\xi = 0$ the node behaves as a free node, and near $\xi = 1$ it behaves as a support. In order to limit the amount of supports, equation 2g is added as an extra constraint. Consequently, the optimization problem for all these constraints and objectives can be written as follows:

Problem-1

$$\underset{\rho,\xi}{\text{minimize}} \quad J = -u_{out} \tag{2a}$$

$$\text{subject to} \quad KU = F, \tag{2b}$$

$$K = \sum_{e=1}^{N} \rho_e^p K_e + \sum_{e=1}^{N} \xi_e^q K_{e,sup}, \tag{2c}$$

$$\rho_e \in [\rho_{min}, 1], \tag{2d}$$

$$\xi_e \in [\xi_{min}, 1], \tag{2e}$$

$$\sum_{e=1}^{N} \rho_e \leq V_{frac} \sum_{e=1}^{N} 1, \tag{2f}$$

$$\sum_{e=1}^{N} \xi_e \leq S_{frac} \sum_{e=1}^{N} 1 \tag{2g}$$

C. Resolution algorithm and sensitivities

When working with the SIMP approach, several algorithms can be used to solve the topology optimization problem. The historical one is the Optimal Criterion (OC) algorithm, widely used for topology optimization, especially for its speed and ease of implementation. However it can only support one constraint, i.e. it cannot be used to solve problem-1 since this latter contains an optimization of the supports which introduces an additionnal constraint beside the volume constraint. In [22], Sigmund presented an interesting algorithm of SIMP method coded under Matlab, but it uses the OC algorithm. To overcome this limitation and consider multiple constraints, we propose to implement the MMA method from K. Svanberg [23]. By doing so, the algorithm allows to solve problem-1.

In order to apply the MMA algorithm we need the sensitivities of the objective function (output displacement) and constraints (eq. 2a-f-g) of the optimization problem-1. To derive the sensitivities, the classical adjoint method is used. To compute the sensitivity of the objective, we wrote the objective function according to the displacement vector U as $J = -u_{out} = -L^T U$. The vector L has value 1 at the degree of freedom corresponding to the output point and with zeros otherwise. Then the adjoint method is used by rewriting again the objective as $J = -L^T U = -L^T U + \Lambda^T(KU - F)$ with Λ an arbitrary but fixed real vector that has the same dimension than U. This is justified as $KU - F = \mathbf{0}$ according to (2b). This reformulation allows to compute the sensitivity of the objective with respect to any variable $x = \{\rho_e, \xi_e\}$ as follows:

$$\frac{\partial J}{\partial x} = \frac{\partial}{\partial x}(-L^T U) + \frac{\partial}{\partial x}(\Lambda^T(KU - F)) \tag{3}$$

$$= -L^T \frac{\partial U}{\partial x} + \Lambda^T\left(\frac{\partial K}{\partial x}U + K\frac{\partial U}{\partial x}\right) \tag{4}$$

$$= \Lambda^T \frac{\partial K}{\partial x}U + (\Lambda^T K - L^T)\frac{\partial U}{\partial x} \tag{5}$$

As Λ is an arbitrary vector, it can be selected in order to cancel the term before the unknown sensitivity $\frac{\partial U}{\partial x}$. This leads to $\Lambda^T K - L^T = \mathbf{0}$. For symmetric K, we can reformulate the former equation to finally obtain $K\Lambda = L$. This equation can be interpreted as solving another mechanical problem where Λ and L are respectively displacement and force vectors. The difference with the initial (2b) is the force vector being an unity value and applied to the output port in the opposite direction as the displacement objective. The partial derivative of the stiffness matrix from $\{\rho_e, \xi_e\}$ variables can be easily computed using equation (2c) as follows:

$$\frac{\partial K}{\partial \rho_e} = p\rho_e^{p-1} K_e \tag{6}$$

$$\frac{\partial K}{\partial \xi_e} = q\xi_e^{q-1} K_{sup} \tag{7}$$

The second order sensitivities are also computed and utilized in the MMA algorithm to improve its convergence.

$$\frac{\partial^2 K}{\partial \rho_e^2} = p(p-1)\rho_e^{p-2} K_e \tag{8}$$

$$\frac{\partial^2 K}{\partial \xi_e^2} = q(q-1)\xi_e^{q-2} K_{sup} \tag{9}$$

The mechanical modeling coupled with the SIMP optimization is a powerful tool for designing mechanical systems. We shall now provide specification for the design of a mechanism dedicated to amplify the displacement of a piezostack actuator.

III. Design and Analysis of the Mechanical Amplifier

The described topology optimization approach is used to derive specification for a displacement amplifier mechanism using a piezostack. We first optimize the design with a lower resolution to study the influence of supports and volume. A design is then selected and used as an initial guess for a refined optimization. A described post processing is applied allowing to obtain a CAD sketch to perform a FE simulation.

A. Specification

Figure 3 represents the complete design specification. We would like to design a mechanical amplifier that has horizontal displacement input in both directions and a displacement output in the upwards direction, according to the mechanical model given in Fig 1. The design domain Ω is set as a rectangle centered on the piezostack, modeled as a rectangle with dimensions 6x18 mm corresponding to the void domain Ω_{void}. Around the piezostack a 6 mm zone is set as the Ω_{mat} domain resulting in a Ω domain as a 18x30 mm rectangle. The supports are optimized over a fixed domain Γ_{sup} set as the left side of the Ω domain. A linear spring is attached to the output port with a fixed stiffness k_{out}. This allows to force the design of a displacement (respectively force) amplifier by imposing a low (respectively high) value of the ratio k_{out}/k_{in}, k_{in} being given by the actuator performances

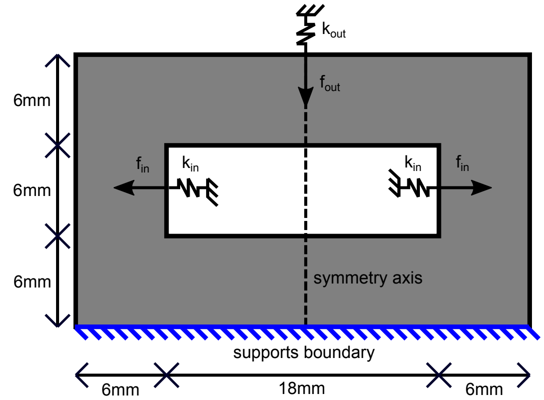

Fig. 3. Design specification.

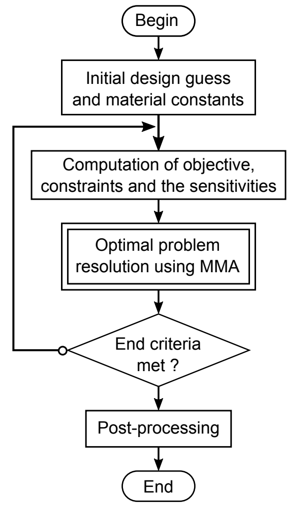

Fig. 4. Flowchart of optimization program.

(see III-D). We choose here a ratio of the output and input spring stiffness of 10^{-3} to force the design as a displacement amplifier. The design domain being symmetric with respect to the x axis, only the upper half is considered by the optimization algorithm. This allows to save memory and reduce computational time.

B. Topological optimal design

The base code is taken from the renowned article [22], using the compliance objective presented plotted in [8]. However several changes, apart from the design specification, have been made for this paper. First of all, the optimization of supports have been added by implementing the formulation of the optimization problem-1. Next, the MMA optimization algorithm was implemented, the one originally present being the OC algorithm. Finally a post processing was added and is discussed in the next section.

The algorithm is used to produce designs with different volume and supports fraction, seen as parameters. Each one varies from 10% to 90% resulting in a total of 81 designs. The step size is set as 200 µm, accordingly every design is made of 90x75 elements and the void area of 30x45 elements (those values are calculated after the y axis symmetry). Following this, we have 6750 density variables and 152 supports variables to optimize for each design. A flowchart of the optimization design process is summarized in Fig. 4.

The amplification factor of every design was computed allowing to obtain the curve in Fig. 5. Two main results can be observed. First, from the support fraction axis, we see that the supports do not have an important influence on the amplification factor for a given volume fraction. To stress this claim, the relative standard deviation of the amplification factors was computed for every volume fraction which is summarized in table I. The table shows that the relative standard deviation is low. This means that the variation of supports creates low dispersion, and consequently the supports has very weak influences on the amplification factor. Furthermore this also indicates that the optimal design does not need much more than 10% of supports. In consequence we will indifferently retain only the 9 designs with 50% supports. The second result is observed along the volume

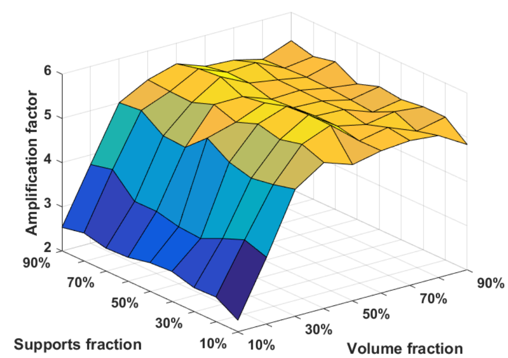

Fig. 5. Estimated amplification factor of designs. Volume and supports fraction ranged from 10% to 90%.

fraction axis. There is a rapid increase at first. Then an optimal solution is met around 50% of volume and finally a stabilization is found above 60%. However this trend does not reflect the reality of the designs as there is still intermediate elements and one node hinges as explained in the next paragraph.

TABLE I
RELATIVE STANDARD DEVIATION σ OF AMPLIFICATION FACTOR FOR EACH VOLUME FRACTION V

V (%)	10	20	30	40	50	60	70	80	90
$\sigma/\lvert\mu\rvert$ (%)	4.7	6.3	3.5	1.6	3.8	1.2	1.3	1.4	3.0

C. Structure extraction

One main drawback of the SIMP approach is the need of post processing. Indeed, the relaxation of the variables allows to simplify the optimization process. However most of the time the obtained optimal design still have intermediate elements which are difficult to interpret [8]. Furthermore

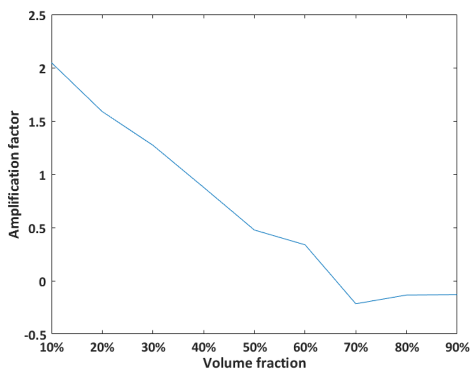

Fig. 6. Estimated amplification factor of designs. Volume ranged from 10% to 90% with 50% supports.

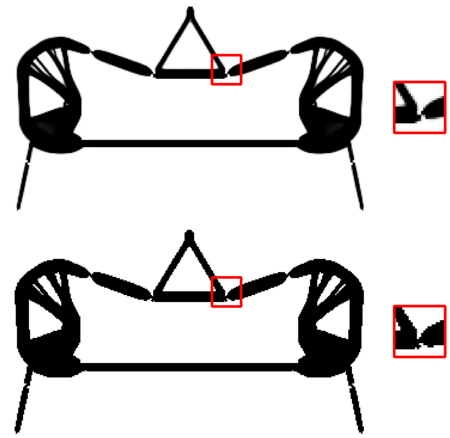

Fig. 7. Selected structure before (up) and after (down) post processing. On the right a zoom on one node hinge before and after post processing.

numerical artifacts appear in the form of one node hinges as in Fig. 7. Those are the result of the FE modeling, the algorithm taking advantage of artificially stiff corner to corner connection of two elements. This increase the flexibility of those hinges, hence improving the objective function. However this results in sharp hinges where the stress, being badly modelled, would approach infinity causing a structural failure [8]. Several approaches can be used to counter the apparition of such hinges during the optimization process such as the MOLE (MOnotonicity based minimum LEngth scale) method [24]. Other approaches can be found in [8].

In this paper we use a classical post processing approach to tackle together the need for a discrete design and the removal of such nodes. It is composed of two steps. First a Gaussian filter is applied to each pixel with a kernel size of 3x3 and deviation of 0.5. A lower deviation value does not create enough matter especially at the one-node hinges and gives raise to unfeasible design. Then a thresholding of 5% is applied to allow the obtention of a discrete 0-1 design. The one-node hinges are effectively removed as the Gaussian filter acts as a blurring filter, smoothing the image gradient especially near the hinges. This operation allows, after the thresholding to add matter around the hinges, resulting in a more realist hinges design. This post processing is resumed on Fig. 7 with a zoom on the effect of a one-node hinge.

For the selected 9 designs with 50% of supports, the previously described post processing was applied and the amplification factor was again computed. Figure 6 represents the evolution of the gain according to the volume fraction. The amplification factor now decreases with the volume. This change is explained as high volume designs were mostly matter and the post processing made them too bulky. Even worse, over 40% volume, we obtain force amplification mechanism with an amplification factor lower than 1; and over 70% of volume, we even have an inversion of the output direction (negative amplification factor).

Given this new conclusion on the obtained optimal designs, we choose to select the design having 20% of volume. This design provides a good amplification/feasibility compromise since the 10% volume design is too fragile to be prototyped. To improve the selected solution, another run with 100μm precision is made. To reduce the convergence time we used the selected 20% volume and 50% supports design before post processing as the initial guess for another optimization. This new design is then post processed with a deviation of 0.1 as shown on Fig. 7. We use a lower deviation value than the previous 200μm precision designs. Indeed this new value can now produces feasible designs and a lower deviation allows for more flexible designs as it adds less matter.

D. Structure simulation

We use the same FE solver from the optimization process to simulate the structure performances. The material properties were set according to the proprieties of the VisiJet Crystal as follows: a Young's modulus of E = 1.463 GPa and a Poisson coefficient of $\nu = 0.35$. The piezostack free displacement is given as $u_{in} = 19.6$ µm and the blocking force $f_{in} = 1000$ N, resulting in an input spring stiffness of $k_{in} = f_{in}/u_{in} \approx 51 \times 10^6$ N/m. The simulation used the design before and after the post processed design and we obtained an amplification factor of 4.47 and 4.05 respectively.

IV. Experimental validation

A. Experimental bench

To validate the design an experimental characterization was carried-out. It consists to asses the structure amplification ratio. First of all, we started by extracting a CAD compatible format of the structure design from the image given by Matlab. A two steps process was developed that allows for an automatic creation of a sketch. To begin, a void layer of pixels is added at the external contours to make their detection easy. Following, a Matlab routine is used to extract the these contours as an ensemble of points. Then, all the extracted points are loaded in a CAD software (Inventor) where an approximation with splines is made to create a sketch. In order to simplify the bench design, small features were added before prototyping. First to ensure

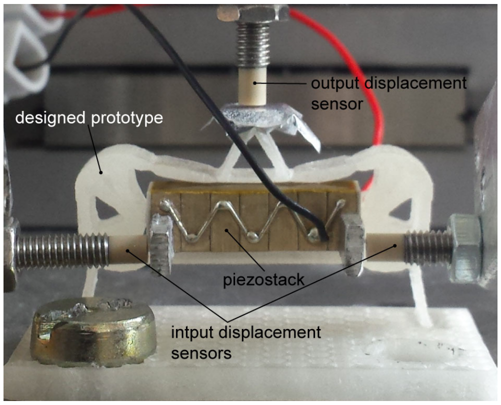

Fig. 8. Experimental bench picture.

the clamping and punctual transmission of force from the piezostack small pieces of matter were added at the input nodes. Second, to facilitate the measurement of the output displacement a square plate is added at the output tip. Finally, a rectangle base allowing to screw the prototype is designed using honeycomb structure, allowing a flatter surface. The complete design was 3D printed using the Pro jet SD 3500 machine with the VisiJet Crystal material.

To assess to the amplification ratio, three inductive sensors were used. The sensor placed on the top measures the output displacement of the mechanism whilst the other sensors measure the input displacement provided by the piezostack actuators (see fig 8). This kind of sensor require a reflecting surfaces such as metal. To do so, two small Al (Aluminium) pieces were glued to the ends of the piezostack and a small sheet of Al was glued at the output plate, as illustrated in Fig. 8. The piezostack is then inserted in the structure as described in the specification. The complete structure is then screwed to the table with the sensors surrounding it as shown on Fig. 8. The piezostack and sensors are controlled using a dSPACE card with a x20 voltage amplifier to obtain the full displacement range. A MATLAB program is used to save, process and visualize the experimental data.

B. Experimental characterization and results

The piezostack is excited from 0V to its maximal range (75 V) with a step of 10V. The inductive sensors (ECL202 from IBS company) are calibrated to have a resolution of 40 nm . Two inductive sensors are placed at the two extremities of the piezostack following the scheme in figure 8 in order to measure its displacement. A third inductive sensor is used to measure the (amplified) output displacement. For each step the measure of the two input sensors are summed to estimate the piezostack displacement. Figure 9 presents the mechanism output displacement versus the piezostack displacement. A linear interpolation of the data allows to estimate the amplification factor as the slope of a regression line, resulting in a value of 4.05. With a coefficient of determination R^2 of 0.9996, this experimental result indicates a very good agreement with the post-processing.

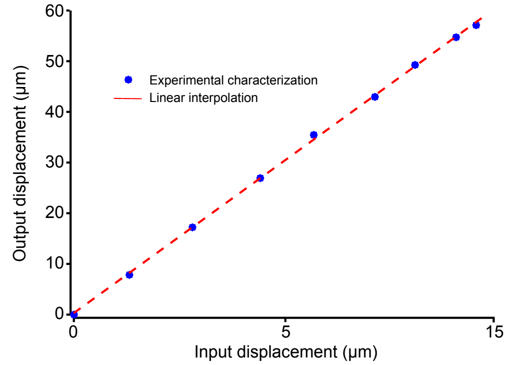

Fig. 9. Experimental characterization of the prototyped mechanism.

TABLE II

SUMMARY OF AMPLIFICATION FACTOR AT DIFFERENT STAGES AT DIFFERENT PHASES OF DESIGN

Phase	Theoretical	Post Processed	Experimental	External work [25]
Amplification factor	4.47	4.05	4.05	3.33

C. Discussion

Table II resumes the obtained amplification factors at the different phases: theoretical design, post-processed design and prototyped design. As can be seen, a small reduction of about 10% of the ratio from 4.47 to 4.05 is observed between the theoretical and the other designs. This variation is expected as SIMP method relaxes the binary densities (0-1 problem) to a continuous formulation. In this configuration, the final design includes an intermediate densities that are difficult to interpret and not easy to manufacture [8]. Those elements have lower stiffness than full elements according to the interpolation law presented in eq. (2c). This artificial low stiffness contributes to produce higher deformation and improves by the way the objective especially around the hinges. However, the post-processing described in section III-C transforms some of those elements to full elements through a Gaussian filter. As a consequence, the stiffness of the hinges increases and therefore reduces the amplification factor. In contrast, the experiment result shows a good agreement with the post-processing design. Almost the same amplification factor is obtained for both design. Otherwise, prototyping with plastic material allows to obtain higher compliance, but at the expense of the mechanical efficiency than metallic material. Finally a comparison with one classic rhombus mechanism as in [25] allows us to obtain up to 20% higher amplification factor, asserting the validity of the approach.

V. CONCLUSION

The design and the fabrication of a mechanism devoted to magnify the deformation of a piezoelectric actuator is presented. Commonly known as ”amplification mechanism” this design is obtained by utilizing the SIMP formulation of topology optimization method. This formulation has been chosen for many reasons: (i) unlike classical approaches, it leads to a systematic design (ii) it allows to avoid the 0-1 material density problem (iii) it is mathematically well-posed and easy to implement. To take advantage of this method, we started by improving the SIMP code presented by Sigmund in [22]. Basically, we substituted the OC method reported in the original code by the MMA method [23] in order to extend the algorithm to problems with multiple constraints. Then, optimal designs were derived according to the amplifying specification. An analysis was carried out on the influence of the supports and volume using a post processing to select an optimal design. We observed that supports have a weak influence as above 10% designs don't take advantage of having more available supports. Moreover lower volume give better designs as high volume create bulky designs resulting in low performances. A finite element simulation and a experiment with a 3D printing realization were proposed to validate the performances of the selected optimal design. We obtained a theoretical amplification factor of 4.47 and a lower simulation and experimental result of 4.05. This difference is explained from the post processed.

Further work will be focused on the optimization of supports over an area instead of a boundary. This approach can improve the objective for compliant mechanism as presented in [8]. A future work will be to not use deported actuators but design directly a piezoelectric material.

VI. Acknowledgment

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