Iterative experimental/numerical procedure for improvement of dynamic experimental facilities

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Abstract

Onera performs dynamic tests for material and assembly characterization using a high velocity hydraulic jack. When the impact velocity increases, the device is disturbed by a natural frequency (around 5kHz). As a consequence, oscillations appear on experimental responses (mainly for the load), measurements cannot be used directly, and the operating range is limited to [0, 100s⁻¹] for strain rates. Today, Onera uses filtering techniques to improve the operating range and measure mechanical properties over a maximum of strain rate range. Nevertheless, the tools may be ineffective for high velocity impact tests. The paper deals with an original method to throw natural frequencies out of the experimental device used for dynamic characterization. It is based on an experimental and numerical modal analysis of each part of the device. A shock hammer and Nastran FE code are used for the experimental and numerical stages respectively. A numerical sensitivity study is carried out regarding the material properties and the geometry of each part of the dynamic facility. The aim is to identify the most sensitive parameters in order to increase the natural frequencies of the device. Mesh size and type of finite elements are also considered. Finally, a satisfactory agreement is found between numerical and experimental results and a numerical optimization exercise is performed.
1 Introduction

In the field of dynamic characterization of materials, Onera-lille has developed a special device for its dynamic hydraulic jack facility. The experimental device is designed to perform tension and compression tests at several imposed velocities. The expected range was \([0, 0.5]\) for plastic strains and \([0, 500s^{-1}]\) for the strain rates. This device has the advantage of characterizing the material behaviors at moderate strain rates, which are not accessible with quasi-static and Split Hopkinson Bars facilities. However, the measurements (e.g. load, strain or displacement) are disturbed by a natural frequency of the device (5kHz approximately). Today Onera-Lille uses numerical filtering to improve the operating range, but these filtering tools do not guarantee the quality of the material data obtained for high velocities impact tests.

The paper presents a method for identifying the natural pumping frequencies of the dynamic device. The method is based on coupling of experimental and numerical approaches at a preliminary stage for numerical validation. The parameters of interests for improvement of the experimental device are analyzed in terms of natural frequency, and a sensitivity study is conducted to discriminate the parameters. Improvement of the device is archived by a numerical optimization procedure.

The new experimental device is expected to perform correct dynamic tests for all types of materials. It was developed to perform uniaxial tension and compression dynamic tests. The operating range of the new device is \([0, 0.5]\) for plastic strains and \([0, 250s^{-1}]\) for strain rates. This new setup will have the advantage of characterizing the strain rate behavior of materials and the dynamic strength of assemblies used for automotive, railway or aircraft crashworthiness FE simulations.

2 Description of the problem

The dynamic facility and the specific device used for strain rate material characterization are shown on Figure 1.

![Figure 1: Experimental facility and device for moderate velocities.](image)

The hydraulic jack is mounted on a rigid frame with 3 degrees of freedom (2 for translation and 1 for rotation) to load structures with various impact angles.
The jack capacities are: ±50kN for the load and 10m.s⁻¹ for the maximum velocity.

The results obtained by Fabis [1, 2] and Langrand [3] for several materials highlight the disturbing phenomenon due to the frequency response of the device (Figure 2). This phenomenon appears for 30s⁻¹ and increases for higher strain rates. When the number of periods is sufficient, numerical techniques can correctly filter the measurements. However, when the velocity increases or ultimate strain is low, the number of periods decreases. The numerical techniques have difficulty in successfully filtering the measurements which are then unreliable.

![Comparison of 2024 T3 aluminum tensile specimen tests at different strain rates](image1)

**a- 2024 T3 aluminum.**

![Comparison of XES Steel tensile specimen tests at different strain rates](image2)

**b- XES steel.**

Figure 2: Load vs. displacement diagrams observed at different strain rates.

### 3 Design parameters and identification method

Many authors are confronted with similar problems on experimental facilities. Some use numerical filtering tools ([4], [5]), and others have developed calibrated samples fitted with large strain gauges [6]. The frequency response of the device can be filtered naturally using a contact with damping [7]. The experimental device can also be based on the principles of Split Hopkinson Bars [8] to avoid vibratory effects and to access higher strain rates.

Delsart [9] and Fabis [2] have identified design parameters to increase the natural frequency observed for pumping modes (i.e. density, stiffness of some parts of the experimental device). Moreover, parameters such as specimen stiffness or strain rate do not significantly disturb the pumping mode frequency but its amplitude. To quantify and improve the results from the analytical model, the frequency response of a simplified experimental device (see Figure 3) was studied and an experimental parameters sensitivity analysis was conducted.

![Added mass M](image3)

**Figure 3: Simplified experimental setup.**

![Load cell 9031A (K₉ = 6.10⁵ N/m)](image4)

The parameters were the added mass $M$, the stiffness of pre-load screw $K$, and the pre-load value $F$. The results are given in Figure 4. Using this simplified
device, the analysis showed that the added mass, $M$, was the most influential parameter for the first pumping natural frequency. The sensitivity diagram observed for the added mass can be fitted by an hyperbolic mathematical model (see Figure 4). The other parameters appeared to have less influence. The sensitivity of the pre-load screw stiffness $K$ is linear and that of the pre-load value (torque) is asymptotic. However, a complex testing device may be difficult to optimized based on these results alone, especially since the parameters may be coupled.

![Figure 4: Results of sensitivity study (mass, pre-load, stiffness).](image)

To improve the device in terms of its natural frequency, optimization of the experimental setup using an inverse numerical method is proposed. The advantage of FE simulation is to quantify the sensitivity of all parameters (whether or not a they are accessible to the experiment). The drawback of this method is that the optimal solution may be not physically realistic (e.g. material may be fictitious or unrealistic). However, the method may be used to draw the main design features of the new device and validate it before manufacture.

### 4 Validation of the method and optimization of the device

#### 4.1 Validation of the method

The validation step is performed using parts of the existing experimental device shown in Figure 1. For the experimental work, a shock hammer (bandwidth 25kHz) was used to impact each part and a Nicolet MultiPro data acquisition unit was used to measure the frequency response. The deformed shapes were visualized using a B&K modal analyzer. Modal analysis was conducted for free-free and realistic boundary conditions (material testing conditions). Figure 5 shows the experimental configuration obtained for free-free and realistic boundary conditions. Results are given in table 1 for each experimental condition in terms of pumping modes (the most relevant for design of the device).
A numerical approach was made using the Nastran FE code. The Lanczos solving method was chosen because the results remain correct when a large number of modes have to be extracted from calculation [10]. For the FE model of the load cell (Kistler 9077A), an equivalent elastic modulus was determined according to the mechanical characteristics given by Kistler eqn (1). The density $\rho$ was calculated with respect to the load cell mass.

$$E_{\text{equ}} = K \cdot \frac{L}{S}$$  \hspace{1cm} (1)

Table 1.: Experimental results.

<table>
<thead>
<tr>
<th>pumping modes</th>
<th>modal analyzer</th>
<th>shock hammer</th>
<th>average values</th>
</tr>
</thead>
<tbody>
<tr>
<td>free-free</td>
<td>realistic</td>
<td>free-free</td>
<td>realistic</td>
</tr>
<tr>
<td>1\textsuperscript{st}</td>
<td>5200</td>
<td>5500</td>
<td>5350</td>
</tr>
<tr>
<td>2\textsuperscript{nd}</td>
<td>7334</td>
<td>12600</td>
<td>9967</td>
</tr>
</tbody>
</table>

where, $K$ is the geometric stiffness, $L$ the length of the load cell and $S$ the cross section of the load cell.

The size $a$ of the finite elements was determined as a function of the upper limit of the frequency domain eqn (2). The mesh size is about 12mm considering $N=10$ and $f=30\text{kHz}$.

$$\lambda = a \times N = \frac{C}{f} = \frac{\sqrt{E/\rho}}{f}, \text{ where } 8 \leq N \leq 10$$  \hspace{1cm} (2)

where $C$ is the elastic wave velocity, $E$ is the Young's modulus, $\rho$ the density, $f$ the upper limit of the frequency domain and $N$ the number of element per wavelength $\lambda$.

Linear hexahedral and quadratic tetrahedral elements were used to study the sensitivity of the numerical results to the type of elements (Figure 6). The use of a linear hexahedral mesh (10,600 elements) allowed the boundary conditions to be described correctly and to the material properties in the real area to be defined.

However, these types of elements lead to geometric approximations. On the contrary, quadratic tetrahedral elements (9,800 elements) correctly meshed the complex parts of the device, but the boundary conditions and material properties
were sometimes difficult to apply properly. Nonetheless, the standard deviation observed was less than 8% for the two meshes (hexahedral and tetrahedral meshing) and the boundary conditions studied here. The numerical results are in good agreement with experimental data in terms of deformed shapes and natural frequencies concerning pumping modes (table 2). The difference between experimental and numerical results was also about 8%. A numerical analysis was performed for the main parts of the experimental device.

The numerical prediction of modal responses was also correct for pumping modes. The prediction capability of FE models was thus demonstrated and the optimization of the experimental device could begin.

Table 2.: Comparison between experimental and FE results – Average values.

<table>
<thead>
<tr>
<th>pumping modes</th>
<th>free-free boundary conditions</th>
<th>realistic boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>EXP</td>
<td>FE</td>
</tr>
<tr>
<td>1\textsuperscript{st}</td>
<td>5350</td>
<td>5475</td>
</tr>
<tr>
<td>2\textsuperscript{nd}</td>
<td>9967</td>
<td>9747</td>
</tr>
</tbody>
</table>

(a- 1\textsuperscript{st} pumping mode)  (b- 1\textsuperscript{st} pumping mode)

Figure 6: FE deformed shape for 1\textsuperscript{st} pumping mode.

4.2 Optimization of the device

Optimization of the experimental device was carried out using a purely numerical modal analysis. The identification procedure was conducted in two steps: first, the material properties, then the geometrical parameters. The optimization criterion was expressed as a function of the pumping frequencies. Service constraints related to the vibratory response of the device as well as design constraints added for the measurement requirements (optical displacement and load cell measurements) led to the FE hexahedral model shown in Figure 7.

Optimization of the material properties was carried out with boundary constraints. For the rigid frame (Figure 3), the values of Young’s modulus and density were very close to those for steel ($E=2.1\times10^5\text{MPa}$ and $\rho=7700\text{kg/m}^3$). For the added mass on the load cell, the values found were similar to those for titanium ($E=1.1\times10^5\text{MPa}$ and $\rho=5000\text{kg/m}^3$). Global equivalent mechanical properties identified from the load cell technical data (Kistler 9071A) were: $E=1.48\times10^5\text{MPa}$ and $\rho=6050\text{kg/m}^3$ according to eqn (1).
With these properties the standard deviation in terms of load cell mass was small (about 2%). At this stage of the optimization process, the first pumping frequency is increased to 8.8kHz.

Deformed shape obtained for the first pumping mode is shown in Figure 8.

![Deformed shape](image1)

Figure 7: New device shape. Figure 8: Deformed shape - 1st optimization.

The choice of a material such as beryllium (density 80% lower and elasticity modulus 50% upper than steel) for the rigid frame would numerically increase the first natural pumping frequency to 9.5kHz. However, this material is very expensive with respect to the gain obtained (about 8%).

Another solution consists in optimizing the bending stiffness of the rigid frame as shown on Figure 9 by varying:

1. the depth of the boring,
2. the diameter of the rigid frame part below the load cell,
3. the thickness of the rigid frame part below the load cell.

The bending stiffness of the rigid frame effectively appears to be a sensitive parameter regarding the pumping modes of the device. When optimization of the material properties was completed, the identification procedure was continued by varying the geometrical parameters of the rigid frame.

![Rigid frame geometric description](image2)

Figure 9: Rigid frame geometric description.

Figure 10 shows the gain for the first pumping frequency (output parameter) versus the chosen part size (input parameter) and the process used to converge to the optimal solution in terms of rigid frame geometry. With this new geometry (Figure 11), the first pumping frequency was increased to 10.5kHz from 8.8kHz. A FE model was constructed with 15,000 linear hexahedral elements and the deformed shape obtained for this new device is shown in Figure 12.
At the end of the optimization steps (material and geometry), the main modifications were as follows:

- choice of the load cell ($K_{opt} = 3 \times K_0$ and $M_{opt} = M_0/3$)
- the new material of the added mass on the load cell was titanium
- the geometric shape of the rigid frame is optimized as regards the bending stiffness.

The first pumping frequency was increased to approximately 10.5kHz. Testing should be conducted using this new device material without hard filtering techniques up to 250s\(^{-1}\). The direct modal response of the existing and new devices showed that the first natural pumping frequency was effectively increased to 10.5kHz. The signal amplitude was also significantly decreased (Figure 13).
The optimized device for dynamic testing is presented in Figure 14 for tension and compression loads. Other modifications were made:

- the slideway guide system for the jack piston was reversed to limit spurious frequencies
- the tension bar was stiffened to limit generation of low frequencies on impact (tension bar/slider contact).

Figure 14: Device optimized for dynamic material testing (tension and compression loads).

5 Conclusion

This paper describes a design procedure for improving the dynamic testing capabilities of the device used to characterize material behavior at moderate strain rates. The aim was to increase the first natural pumping frequency of the experimental device, which biased measurements of the material behavior at higher strain rates.

The method is based on numerically modeling the frequency response of the existing device to validate the numerical approach, and on optimizing the design parameters of such a device (stiffness, density, pre-load, geometry, etc.).

The first part of the paper discusses validation of the method. Frequency analysis of the existing device parts was conducted experimentally (using a shock hammer/Nicolet MultiPro unit and a modal analyzer) and numerically (using the Nastran FE code). The numerical results are in good agreement with experimental data on the pumping modes.

The new device was designed numerically by varying the design parameters to increase the pumping mode frequency. The improvements mainly concerned the choice of the load cell (mass and stiffness), optimization of the added mass on the load cell and the rigid frame bending stiffness. Using these new material and geometrical properties, the first pumping natural frequency is increased to 10.5kHz, up from 5kHz for the initial device. Using this new device, the upper limit of the strain rate domain should be 250s⁻¹ instead of approximately 100s⁻¹.
The new device for strain rate material characterization is being manufactured. Experiments using metallic materials (strain-rate-dependent such as a mild-steel and -independent such as aluminum alloys) will be carried out to completely validate the optimization principles and results. The new upper bound in terms of pumping frequency and strain rate domain will be evaluated for final validation the numerical approach.

References