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Synergistic integration of vibration absorption and damping into 3D-printed fixtures for thin-wall machining

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ABSTRACT

To reduce vibration during machining and to avoid re-manufacture or scrap parts, two main methods have been proposed, vibration damping and absorption. In this paper, a novel synergistic integration between both techniques using pillar elements known as flexures (acting as absorbers) and a flexible pneumatic expandable diaphragm (acting as a damper) is presented. Four setups have been compared: the proposed Hybrid Fixture, a traditional solid vice, a flexures fixture, and an expandable diaphragm-only clamp. The Hybrid Fixture was able to reduce the workpiece vibration by 58% and the surface waviness by 74.1% when compared with the traditional solid fixture.

1. Introduction

High-value manufacturing industries, such as aerospace and automotive, have a clear need for highly efficient lightweight components. This need is translated into workpieces with relatively wide surfaces and small thicknesses, resulting in low-stiffness structures (e.g., compressor blades and automotive casings) [1]. This specific characteristic presents a considerable challenge when machining processes are involved, as chatter occurring during this process is the main reason for worsening the finish of the part or even increasing the scrap ratios of the process [2–4]. In this respect, the approaches for reducing part vibration could be divided into two main groups. On one hand, vibration damping is the most common approach as it is widely used to reduce chatter, and on the other hand, there is vibration absorption.

The first route relates to reducing the amplitude of the vibration by dampening it and preventing the system resonance. This can be achieved by using damping elements in the system such as elastic components, magnets, and dampers [5,6]. This allows to overcome any uncertainties of the systems dynamic modelling which is ideal for simplified passive damping systems while enabling lower complexity than active damping systems. However, when using these solutions in environments with a continuous vibration source (e.g., machining), the dissipation that can be achieved by damping the system is limited.

On the other hand, vibration absorption systems rely on transmitting the vibrations to other elements that can dissipate it and create anti-resonances. This transfer is usually achieved using moving parts, like systems with springs and masses, to make these external parts resonate while absorbing the energy. The complexity of these systems is generally higher while their dynamic behaviours (e.g., natural frequencies, amplitudes) need to be very carefully detailed to obtain a fully functioning system.

The combined effect of these two approaches (i.e., vibration absorption and damping) is investigated in this paper by introducing a novel hybrid fixturing solution (Fig. 1). This innovative fixture uses pillar

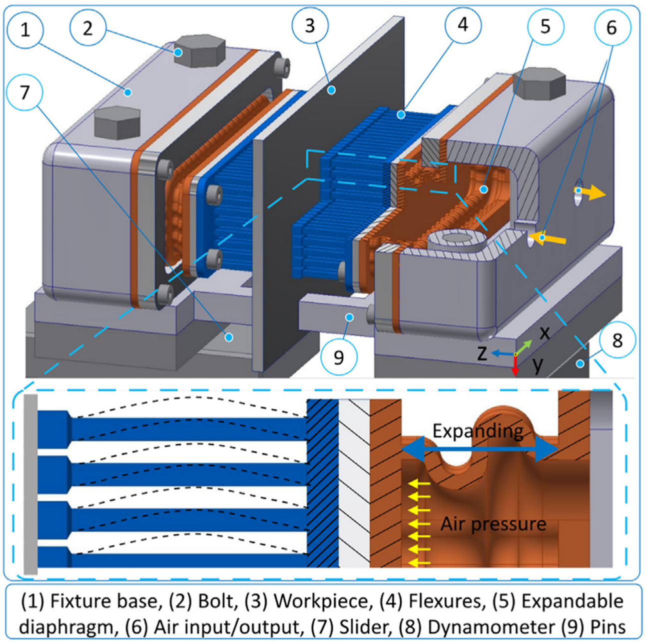

Fig. 1. Sectional view of the proposed hybrid fixturing system showing the combination and working principle of the diaphragm and flexures.

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elements known as flexures (i.e., vibration absorber) and a flexible pneumatic expandable diaphragm (i.e., vibration damper) to allow vibration absorption and damping simultaneously, while the inflation controls the clamping load generated on the flexures. It can be reconfigured and tuned to different frequencies by changing the shape of the pillars and adapting the pneumatic pressure. Furthermore, the actuation is separated into a rigid clamping force applied by the locating pins and an adaptable support load with the diaphragm. This novel system was evaluated in machining environments and the resultant surfaces from the machining tests were analysed.

2. Theoretical framework of the fixture dynamics

2.1. Mechanical design of the Hybrid Fixture

Additive manufacturing is primarily used for rapid tooling [7]. While the use of frequency-tuned flexures (④ Fig. 1), has been previously reported to reduce the vibration response of systems [8], they require very accurate control of the clamping load. The system proposed here addresses these drawbacks while making the solution closer to industrial use. This will be achieved by introducing an expandable diaphragm (⑤ Fig. 1) that will ensure that the clamping load and force distribution can be more accurately controlled by changing its internal pneumatic pressure. This system will be denominated Hybrid Fixture as it combines chatter suppression with conformability. Fig. 1 presents a 3D model (sectional view) of the most important parts of the proposed fixture. The flexures ④ are a matrix of pillars with the same height and cross section that together generate a support surface that goes in contact with the workpiece ③. Each flexure can only withstand a small load, but the addition of them together increases the total clamping load. However, for the machining trials, some rigid pins (⑨) had to be added to further increase the general clamping load and leave the flexures as a support element. When a workpiece is supported using flexures tuned at the problematic frequency for the system, they have been proven to be able to resonate and absorb the energy of the excitation and reduce the workpiece vibration. The drawback of the flexures is that it requires very accurate control of the clamping load as it needs to be high enough to avoid sliding with the workpiece but not too high as buckling might happen. To avoid this, the actuation of the flexures was enhanced by implementing an expandable diaphragm ⑤ which ensures a constant contact force. The expandable diaphragm is designed to be inflated in such a way to generate an adjustable clamping load axial to the direction of the flexures (z-axis) and having more distributed force between pins. The base ① is a rigid part that creates the foundation of the system.

2.2. The Hybrid Fixture model

To represent the dynamics of the system analytically, the proposed system is schematically represented in Fig. 2a. Two diagrams are created, Fig. 2b and d, as the effect of the flexures manifests in xy-plane and the effect of the diaphragm happens in z-axis. Three masses need to be considered to represent the full dynamics of the system m_{wp}, m_{flex} and m_{dia}, for the workpiece, the flexures, and the expandable diaphragm, respectively. By connecting them with springs (k) and structural dampers (β), the dynamics of the system can be represented. Since the flexures have a square shape, Fig. 2b shows a 3DOF schematic of how the flexures can resonate and absorb vibrations in the x and y axes (i.e., parallel to the workpiece), against the forces generated in these axes by the tool. The equation of motion for the three elements composing the system, is presented in Eq. (1):

$$M\begin{Bmatrix} \ddot{x}_{wp} \\ \ddot{x}_{flex} \\ \ddot{x}_{dia} \end{Bmatrix} + \begin{bmatrix} \beta_3 & -\beta_3 & 0 \\ -\beta_3 & \beta_2+\beta_3 & -\beta_2 \\ 0 & -\beta_2 & \beta_1+\beta_2 \end{bmatrix} \begin{Bmatrix} x_{wp} \\ x_{flex} \\ x_{dia} \end{Bmatrix} + \begin{bmatrix} k_3 & -k_3 & 0 \\ -k_3 & k_2+k_3 & -k_2 \\ 0 & -k_2 & k_1+k_2 \end{bmatrix} \begin{Bmatrix} x_{wp} \\ x_{flex} \\ x_{dia} \end{Bmatrix} = \begin{Bmatrix} F_{xy}(t) \\ 0 \\ 0 \end{Bmatrix} \quad (1)$$

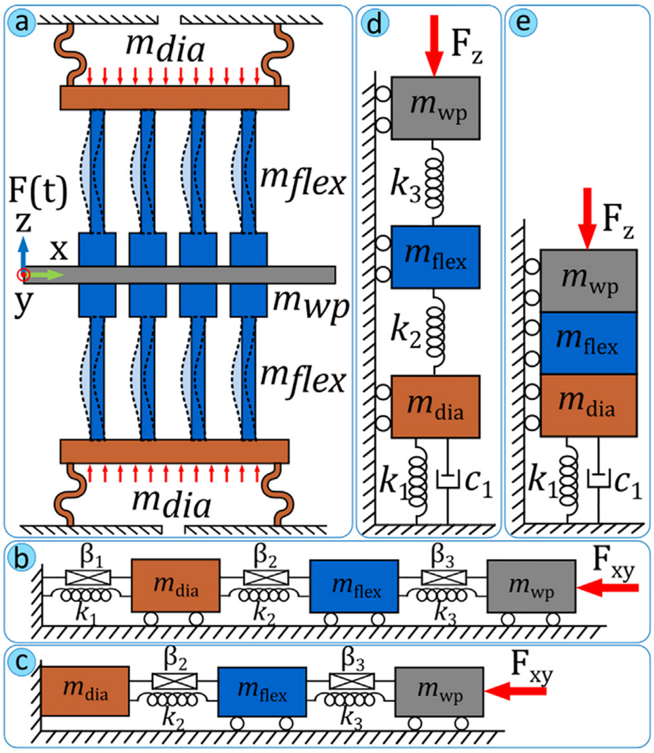

Fig. 2. Hybrid fixture free body diagrams: a) whole system diagram, b) 3DOF dynamics in xy-plane, c) simplified 2DOF dynamics in xy-plane, d) 3DOF Dynamics in z-axis, and e) simplified 1DOF dynamics in z-axis.

where, M is a diagonal matrix of (m_{wp}, m_{flex}, m_{dia}). The main characteristic of this model is that even if three stiffnesses are used to represent the system (i.e., k_1, k_2, and k_3), the stiffness of the diaphragm (k_1) is so high in these directions. Hence, it could be relatively considered rigid, so the diaphragm line of the matrix could be dismissed, and the fixture would become a 2DOF system as in Fig. 2c. This is due to the unique design of the fixture containing the expandable diaphragm with the double curvature (as shown in Fig. 1), which makes it more stable in the horizontal plane (xy-plane) while still allowing for the expansion of the structure (z-axis). On the other hand, the resonance and vibration dissipation capability of the flexures is still present with springs k_2 and k_3, which are calculated from the geometry of the flexures, so when these are tuned to match the frequency of the excitation force (F_{xy}), the mass of the flexures (m_{flex}) will resonate thus absorbing the vibration and reducing the movement of the workpiece (m_{wp}) [8].

On the other hand, Fig. 2d represents how the expandable diaphragm dissipates the vibration in the z-axis (i.e., perpendicular to the workpiece and axial to the flexures). When considering the forces in the z-axis, being axial to the pillars, the flexures might not be able to enter their resonance mode so they will not absorb the vibrations. However, the vibration dissipation of the diaphragm will do the viscous damping (c_1) as due to its expansion in this direction (z) and the internal pneumatic pressure that acts as a dissipation system. The equation of motion is as follows:

$$M\begin{Bmatrix} \ddot{x}_{wp} \\ \ddot{x}_{flex} \\ \ddot{x}_{dia} \end{Bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & c_1 \end{bmatrix} \begin{Bmatrix} \dot{x}_{wp} \\ \dot{x}_{flex} \\ \dot{x}_{dia} \end{Bmatrix} + \begin{bmatrix} k_3 & -k_3 & 0 \\ -k_3 & k_2+k_3 & -k_2 \\ 0 & -k_2 & k_1+k_2 \end{bmatrix} \begin{Bmatrix} x_{wp} \\ x_{flex} \\ x_{dia} \end{Bmatrix} = \begin{Bmatrix} F_{xy}(t) \\ 0 \\ 0 \end{Bmatrix} \quad (2)$$

In this case that the flexures are axially excited and cannot work as vibration absorbers, they could be considered rigid (i.e., having infinite stiffness), so with this assumption, the model for the z-axis dynamics can be simplified and represented as a single-degree-of-freedom system as in Fig. 2e. Even so, the diaphragm will not reduce the vibrations as much as the absorption of the flexures does, but it should still dampen the forces happening and not let the workpiece resonate. Following the above hypothesis about flexures' stiffness, Eq. (3) represents the dynamics of the workpiece:

$$\left[m_{wp+flex+dia}\right]\{\ddot{x}_{dia}\} + [c_1]\{\dot{x}_{dia}\} + [k_1]\{x_{dia}\} = \{F_z\} \tag{3}$$

When combining both dissipation methods in a single fixture, the absorption of the flexures will act in two directions of the machining forces (i.e., x and y), while the damping will affect the third axis (i.e., z, axial to the flexures). This will ensure that the flexures minimize the vibration of the workpiece at the resonance frequency, while the viscous damping (c_1) of the diaphragm will dissipate the forces in the direction that the flexures cannot.

3. Experimental setup

Fig. 3 shows examples of the proposed configurations to compare the vibration absorption and damping capabilities of the novel Hybrid Fixture and thus, show its advantages.

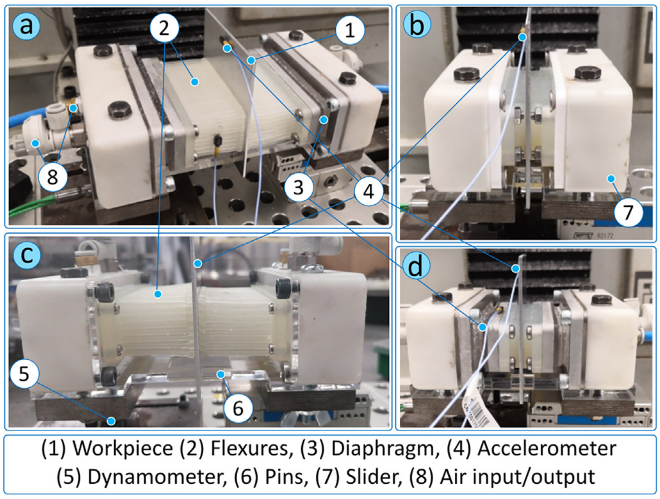

Fig. 3. Experimental setups and elements: a) Hybrid Fixture, b) solid fixture, c) expandable diaphragm fixture only.

Each setup had its dynamic behaviour and was used to analyse different aspects of the Hybrid Fixture separately.

(a) **Proposed Hybrid Fixture**: This combines the vibration absorption by the flexures and the vibration-damping capabilities of the expandable diaphragm (Fig. 3a).
(b) **Solid fixture**: This setup represents the traditional fixtures used for machining in which a rigid vice clamps a workpiece as strongly as possible (Fig. 3b).
(c) **Flexures fixture**: This is only flexures and rigid pins (Fig. 3c).
(d) **Diaphragm fixture**: This includes only the expandable diaphragm (Fig. 3d).

To prove the ability of this Hybrid Fixture to minimise vibration, experimental trials have been carried out by milling of a thin-wall component (that is prone to chatter).

3.1. Workpiece properties and flexure design

The workpiece used for these experiments is a flat sheet of aluminium alloy 1050 with a size of 82 × 82 mm. The thickness of the workpiece is 3 mm and side milling was performed using a 4 flute 10 mm diameter cutter. To analyse the dynamics of the workpiece on the setups, hammer tests have been done to obtain the Frequency Response Functions (FRF) using accelerometers and a hammer system (Prosig 8006). By doing this, the resonances and anti-resonances of the workpiece and flexures could be analysed, together with the damping effect induced by the diaphragm. The resultant FRFs obtained can be seen in Fig. 4.

When the workpiece is held rigidly, the natural frequency is around 500 Hz, and so, if the machining is done close to this frequency, the workpiece might enter its resonance mode. On the other hand, the diaphragm fixture shows a wider and lower peak at a similar frequency, which represents a damped resonance, confirming the damping effect of the inflatable diaphragm.

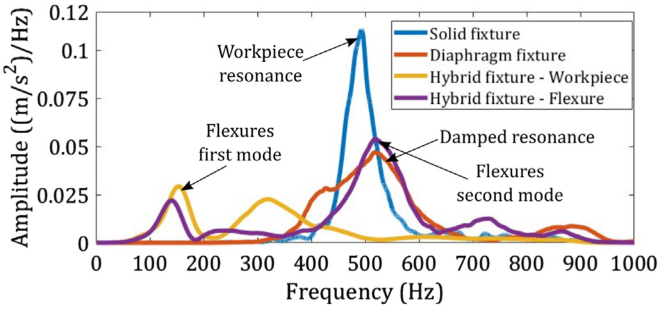

Fig. 4. FRF comparison between the dynamic behaviour of the workpiece when being clamped with the setups.

Considering this, the flexures needed to be tuned to resonate at that frequency by selecting the key dimension. Based on the models described previously [8] it was calculated that the flexures needed to be 30 mm in height and have a 1.5 mm square cross-section. These dimensions are defined considering the properties of the selected 3D printing Durable resin from Formlabs.

By doing the hammer tests on the Hybrid Fixture, the dynamics of the workpiece and flexures is analysed. Both show a natural frequency of around 150 Hz, which is related to the first vibration mode of the flexures. However, at around 500 Hz, the flexures show a resonance, due to the second mode of vibration, while the workpiece does not, which might be due to the anti-resonance generated by the flexures. In conclusion, according to the analytic models and the experimental results, the designed flexures should be capable of absorbing the vibration generated in the workpiece when they are excited by the machining process at around 500 Hz.

3.2. Machining experiments

To validate the proposed Hybrid Fixture, thin-wall workpieces were side (climb) milled as this approach generated cutting forces and subsequent vibrations in the direction of the flexure's absorption capability. Table 1 presents the characteristics of the machine and the tool, and the process parameters.

Table 1
Parameters of the machine and the tool.

Machine (Tormach 770)		Tool (HSS End mill)	
Spindle RPM	6k, 7.5k, 9k	Diameter (mm)	10
Depth of cut (mm)	1.0 axial 0.25 radial	Tooth passing frequencies (Hz)	400 500 600
Clamping load (N)	200 positioning 50 supporting	Cutting edges	4
Feed rate (mm/min)	280	Tool overhang (mm)	40

The data was acquired using a Kistler 9317C dynamometer to measure the clamping loads generated by the fixtures and the flexures, a Prosig P8004 data acquisition system, and a single-axis PCB® piezotronics 352C23 accelerometer (0.5 mV/(m/s^2)) to record the vibration response of the workpiece. The sampling rate selected to acquire the acceleration data was 100,000 Hz.

4. Experimental results and analysis

The four setups were tested at three tool spindle speeds (6000 rpm - 400 Hz, 7500 rpm - 500 Hz, and 9000 rpm - 600 Hz). The vibration response of the workpiece, the waviness of the resultant surface, and the depth of cut (DoC) obtained have been analysed. Identical machining condition were applied for each setup to ensure a fair comparison with the parameters shown in Table 1.

4.1. Workpiece vibration

The workpiece vibration presented in Fig. 5a are the values obtained during the cutting process from the attached

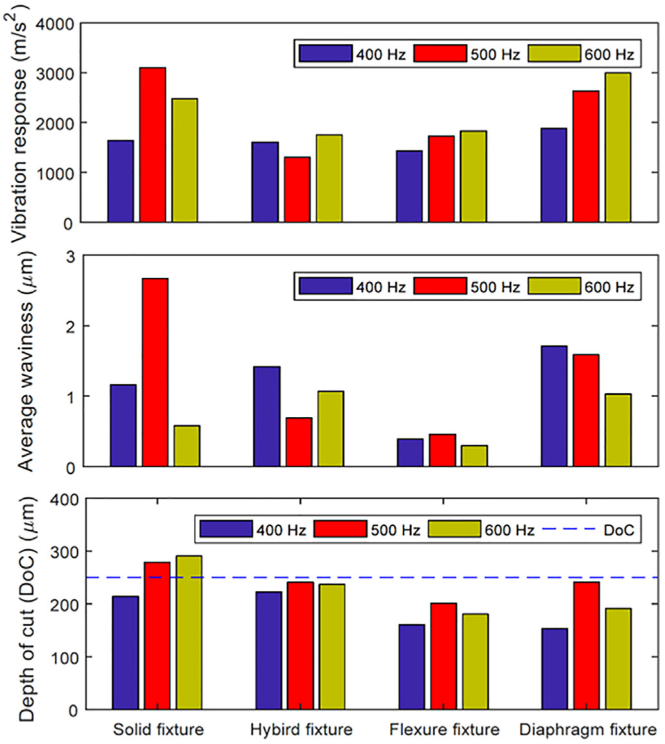

Fig. 5. Comparison between the machining experiments result of the four setups: a) vibration response of the workpiece, b) average waviness, c) depth of cut (DoC) measurements.

accelerometers on the workpiece. A general trend can be seen in the solid and expandable diaphragm (Fig. 5a) setups where at 400 Hz the vibration response is the lowest and it increases when the spindle speed is raised. For the proposed Hybrid Fixture (Fig. 5a), at 500 Hz, which is the resonance frequency of the flexures, the vibration of the workpiece is counteracted compared to all other three setups (1308 m/s^2) as expected. The value of this is even bigger when comparing it with the solid Fixture result as it achieves a vibration response 58% lower when the workpiece should be in resonance. The expandable diaphragm fixture does not achieve such a good vibration dissipation (2633 m/s^2) as it does not have vibration absorption capabilities. On the other hand, it proves that just dampening can reduce the vibration by 15% as the workpiece resonance does not happen at 500 Hz.

4.2. Surface morphology analysis

To evaluate the effect of the vibration absorption on the milled thin-wall parts, their surface morphology was evaluated in an Alicona G4 3D metrology system. The waviness of each sample is calculated in MountainsMap (Digitalsurf) and compared as shown in Fig. 5b. When increasing the tool rotational speed, the waviness of the surface decreased (i.e., better surface quality). However, for the problematic frequency (500 Hz), there is a very outstanding peak in the case of the solid fixture. Considering that the vibration response tests already showed that the workpiece had a high acceleration when excited at 500 Hz, it can be stated that the elevated average waviness (2.67 μm) is a direct result of the chatter happening in the process. The waviness outcome from the expandable diaphragm fixture tests at 500 Hz (1.59 μm) matches with the results from the vibration analysis. Also, the surface waviness obtained with the Hybrid Fixture at the same frequency is 74.1% lower (0.69 μm), proving that the vibration absorption of the flexures also affects the surface waviness. Although the flexures fixture reduces the surface waviness, it is a result of the low depth of cut achieved by this fixture as discussed in the following.

4.3. Geometrical accuracy

The last analysis to indicate the accuracy of the proposed Hybrid Fixture is to measure the real radial depth (a_e) of the cut achieved. Considering that the machining was done in one pass and with a nominal a_e = 0.25 mm, these measurements will give an insight into the part deflection. As such, Fig. 5c shows that the Hybrid Fixture yields the closest a_e values to the nominal (93.5%) one. On the other hand, the solid fixture results are further from the nominal (113.9%) value due to the resonance of the workpiece. The flexure fixture achieved a significantly lower depth of cut due to uncontrolled force on the flexure. Finally, the expandable diaphragm fixture shows the worst results in terms of depth of cut accuracy (78.1%).

5. Conclusions

In this paper, a novel Hybrid Fixture system has been designed that combines the action of a vibration absorber with load control and damping in a synergistic manner. The hypotheses are: 1) the flexures act as vibration absorbers when tuned at the right frequency to minimize the vibration of the workpiece, and 2) the viscous damping of the diaphragm dissipates the forces in the direction that the flexures cannot. These hypotheses are presented in the analytic model and have been validated by the results of the machining tests. Four setups are compared, the Hybrid Fixture gives the best results in both workpiece vibration and accuracy of the machined surfaces when the cutting forces are at a problematic frequency of the traditional fixture (i.e., 500 Hz) which is the resonance frequency of the flexures. The vibration of the workpiece is counteracted compared to all other three setups (1308 m/s^2). The tests with the solid fixture also showed how a workpiece enters its resonant mode (500 Hz) when the cutting process operated at this frequency, leading to a deterioration of the workpiece surface quality. As such, the analysis of the surface morphology showed that at the resonant frequency, the waviness obtained with the Hybrid Fixture is 74.1% lower (0.69 μm) when compared to the rigid vice (2.67 μm). In conclusion, the proposed Hybrid Fixture containing frequency-tuned flexures and an expandable diaphragm showed advantages in reducing workpiece vibration and hence it ensured improved machining performance and surface quality compared to the traditional solid fixtures.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. Dragos Axinte reports financial support was provided by The Manufacturing Technology Centre Limited.

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