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Stress relaxation insensitive designs for metal compliant mechanism threshold accelerometers

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A R T I C L E I N F O

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A B S T R A C T

We present two designs for metal compliant mechanisms for use as threshold accelerometers which require zero external power. Both designs rely on long, thin flexures positioned orthogonally to a flat body. The first design involves cutting or stamping a thin spring-steel sheet and then bending elements to form the necessary thin flexors. The second design uses precut spring-steel flexure elements mounted into a mold which is then filled with molten tin to form a bimetallic device. Accelerations necessary to switch the devices between bistable states were measured using a centrifuge. Both designs showed very little variation in threshold acceleration due to stress relaxation over a period of several weeks. Relatively large variations in threshold acceleration were observed for devices of the same design, most likely due to variations in the angle of the flexor elements relative to the main body of the devices.

1. Introduction

Previous work has demonstrated the idea of using a fully-compliant bistable mechanism as a threshold accelerometer [1,2]. Such a device is especially interesting as a sensor requiring zero power which can be left in place over long periods. Potential applications would include package shipping in which these types of sensors could indicate harmful shocks or drops. Sensors could also be placed on vehicles, buildings, or bridges [3–5] to monitor impacts or seismic activity. Reports have been made on a variety of low power or zero power accelerometer designs, often involving MEMS structures and monitoring circuitry built on VLSI chips [6–9]. While a macroscopic bistable design cannot be readily fabricated on a silicon substrate, it can be integrated with RFID sensors for remote, zero power sensing of acceleration events [10]. While a number of zero power mechanical structures exist and are commercially available [11, 12] there remains a desire to pursue lower cost alternatives which can be easily adapted to automated readout systems.

The previous and present bistable mechanism designs use four compliant flexible members with two stable positions. The mechanism's central shuttle is meant to attach to an object and measure its acceleration. In these types of mechanisms, as the flexible members are acted upon by an external force, elastic energy can be stored and then released as kinetic motion. The device's outer frame serves as a proof mass which causes a force on the compliant members when under acceleration. If acceleration goes beyond a threshold, the proof mass moves between two possible stable positions, in effect recording a "threshold" event. The basic design and a force–displacement diagram are shown in Fig. 1.

Compliant mechanism threshold accelerometers have been made from sheets of Delrin and ABS plastic and formed through laser cutting [1]. While plastic offers low manufacturing costs and these devices did display bistable switching behavior, they were susceptible to changes in switching threshold over time due to stress relaxation of their flexor elements [13]. In fact, an average increase of 54% in the threshold acceleration was measured after leaving the devices in the stressed state for 72 h [13]. This kind of drift is very undesirable for a zero power sensor meant to be used over long periods.

To address the problem of plastic relaxation, metal threshold accelerometers have been made from sheets of spring steel with designs cut using wire Electrical Discharge Machining (EDM) [14]. Flexor elements were formed by bending thin strips of the spring steel perpendicular to the outer frame. These devices exhibited long term stability with regard to threshold acceleration, but were susceptible to "out of plane" movement during switching. Bracketing elements had to be added to the frames, which induced friction and made thresholds unpredictable from device to device.

This paper introduces two new metal threshold accelerometer designs, both of which have long term threshold stability and do not

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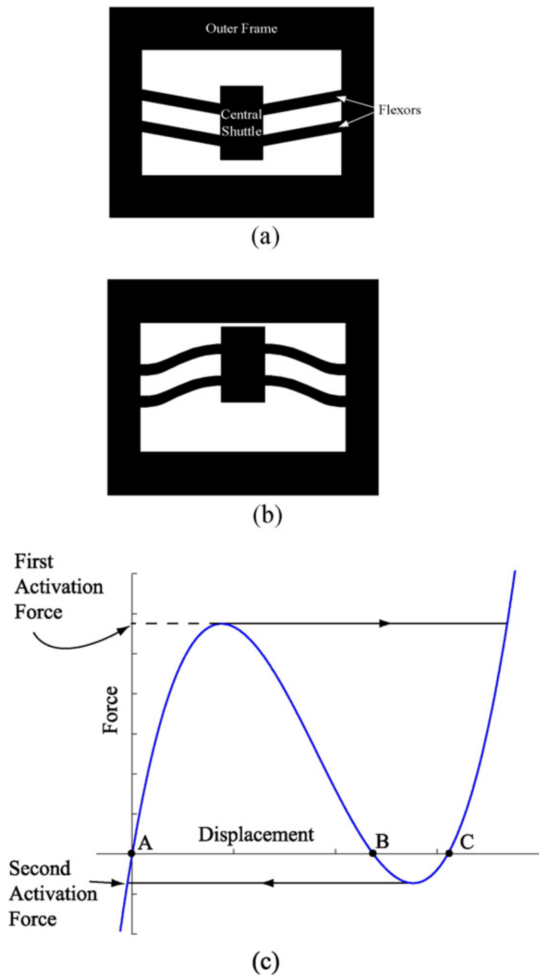

Fig. 1. (a) Compliant bistable mechanism design with four flexor elements in the first stable position, corresponding to point A on the force vs. displacement diagram in (c). (b) Compliant bistable mechanism in the second stable position, corresponding to point C on the diagram in (c). Point B is the unstable position.

require bracketing elements to keep switch motion in a single plane. The first design builds from the earlier work involving cutting spring steel and then bending to form flexor elements. The second design uses metal molding to form a switch with spring steel flexors and a tin frame. Fabrication details for both designs are provided along with a description of how acceleration thresholds were tested. Data demonstrating low variation to stress relaxation for both designs is shown.

2. Fabrication

2.1. Cut and bend design

Our compliant mechanism designs require flexors with small in-plane widths compared to their out-of-plane thicknesses. To achieve this in our "cut and bend design" a sheet of 0.004 in. (~100 μm) spring steel was cut using wire EDM into the shape shown in Fig. 2a. The cut piece was then placed in a specially designed fixture (Fig. 2b) which allowed for the thin flexor elements to be bent down perpendicular to the plane of the main body of the device. A small-diameter drill bit (#60–1.02 mm diameter) was placed in the fixture to guide the bend, avoiding excessive stress during bending. The design is amenable to fabrication by stamping. This design differs from the earlier version of the "cut and bend" devices in that previous versions included a thin

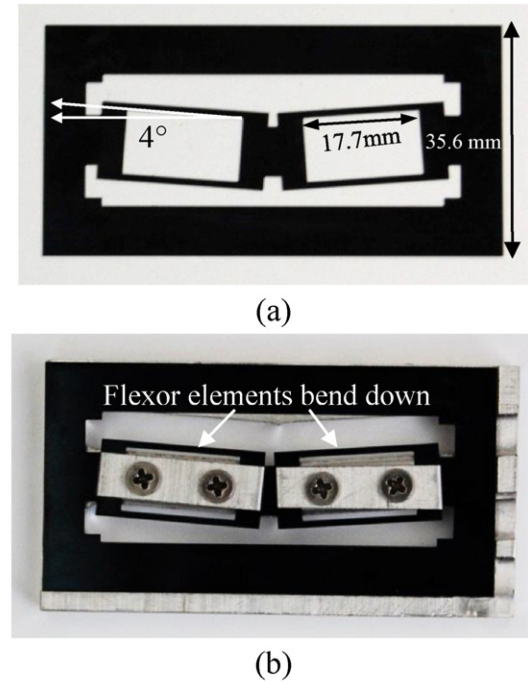

Fig. 2. (a) Top view photograph of the spring steel "cut and bend design" accelerometer after being cut using wire EDM. (b) The cut spring steel positioned in custom fixture used for bending flexor elements perpendicular to the outside frame.

segment of the flexor that was intended to be plastically deformed in torsion. The thin torsion segments resulted in very small out-of-plane stiffness, requiring the placement of guides to prevent out-of-plane motion [15]. The guides induced friction during switch motion. In contrast, the "cut and bend" designs presented here have no rubbing surfaces during motion.

After bending the flexor elements, 2.5 mm thick plastic frames were glued to the outer frame of the devices to add support and weight. Fig. 3 shows photos of the completed accelerometers in both stable positions. The added plastic frames are below the spring steel frames and not visible in these pictures.

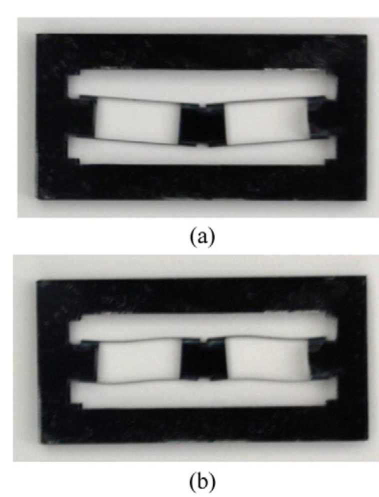

Fig. 3. (a) "Cut and bend" design in the first stable position. (b) The same device in the second stable position with the flexor elements deformed. A plastic frame has been attached to the bottom of the device's outer frame to provide stability and extra weight.

2.2. Molded design

The molded version of the accelerometer sensor starts with a plastic prototype made by laser cutting a 2.9 mm thick acrylic shuttle and frame elements and then gluing wire EDM cut flexors in between these elements. A silicone mold is then made using this prototype as shown in Fig. 4a.

Before casting, additional spring steel flexors, cut using wire EDM, are placed inside the silicone mold as shown in Fig. 4b. The legs were designed 2.5 mm longer on each end than the intended distance between the shuttle and frame to allow molten tin to wrap around them and keep them anchored in place when the tin solidifies. In the extra length on the ends of the legs, a notch and a tab were added and the tabs bent. The notch and tab were intended to add anchoring support between the flexors and tin elements. A close up photo of one of the flexor elements is shown in Fig. 4c.

The final fabrication step used a small stove to melt pure tin. Once the tin liquefied, it was poured into the closed mold (Fig. 4d) and then allowed to cool. The cast switch was then removed from the mold (Fig. 4e) with flexor and tin elements fused together. Fig. 3 shows photos of completed molded switches in both stable positions.

3. Testing

The acceleration threshold for switching between two stable states was measured using a centrifuge and a tachometer. Between tests, the switches were left in their stable, stressed-state position. The testing procedure consisted of placing switches in the centrifuge and increasing its rotation speed until an audible snap was heard — indicating a switch had changed bistable positions. The tachometer was used to measure rotations per minute (RPM) at the switching threshold. Seven trials were made for each switch during each day of testing. Switch velocity was calculated using the equation

$$v = R(2\pi\omega/60), \tag{1}$$

where v is the velocity (in m/s), R is the distance from the center of rotation (in meters) to the center of the shuttle, and ω was the measured RPM for the trial. The radial distance of the sensor from the centrifuge's center was R = 0.276 m. Switching acceleration was determined using the equation

$$a = v^2/R, \tag{2}$$

and average acceleration for each set of tests is reported in Figs. 6 and 7.

4. Results and discussion

The switching acceleration for each of several switches is shown in Fig. 6 for the “cut and bend” design, and in Fig. 7 for the molded design. Each switch was left in the stressed stable state (shown in Figs. 3b and 5b) for a period of 2–3 weeks, and each was tested several times over that period to determine its switching acceleration on that day. The data shows very low variation, particularly when compared with previous tests using plastic switches, which showed an average increase in threshold acceleration of 54% after only 72 h in the stressed state [13]. By comparison, the largest variation between data points in the “cut and bend” design is 2.8% (the diamonds in Fig. 6), and all the variations in Fig. 6, from top to bottom, are 1.6%, 0.8%, and 2.8%. The largest variation in the molded design is 14.9% (the diamonds in Fig. 7). Note that switches represented by the other symbols in Fig. 7 are much smaller, with variations of 5.7%, 8.2%, 14.9%, and 3.2% from top to bottom. Also note that there is no clear trend toward increasing accelerations over time for any of the switches, unlike the previously tested plastic switches which indicated continued plastic deformation. Measured variations for these metal switches was likely due to unintended bending of

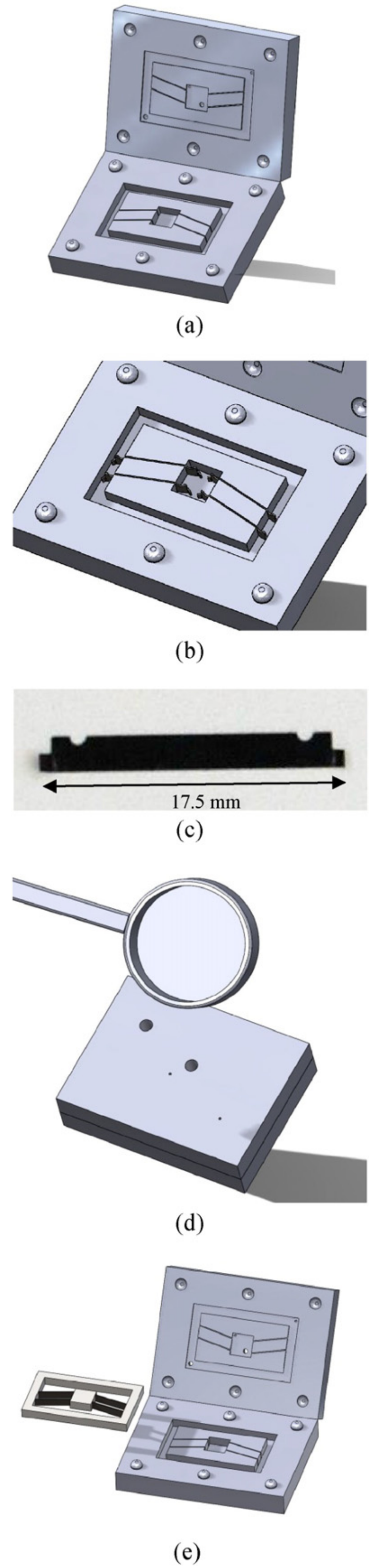

Fig. 4. Fabrication process for molded device. (a) Silicone mold is formed using a plastic prototype of the accelerometer. (b) Four flexor elements are placed in slots inside the mold. (c) Photograph of a flexor element containing notches and tabs on its edged, made by wire EDM of spring steel. (d) Molten tin is poured into the closed mold. (e) The fused device is removed from the mold after the tin has cooled.

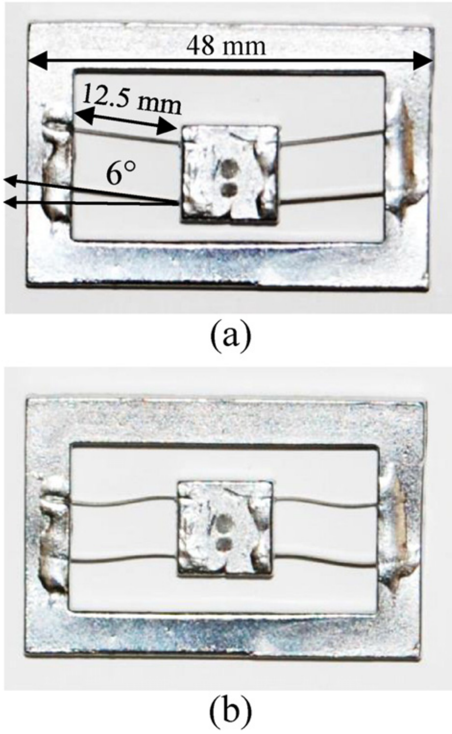

Fig. 5. (a) Molded design in the first stable position. (b) The same device in the second stable position with the flexor elements deformed.

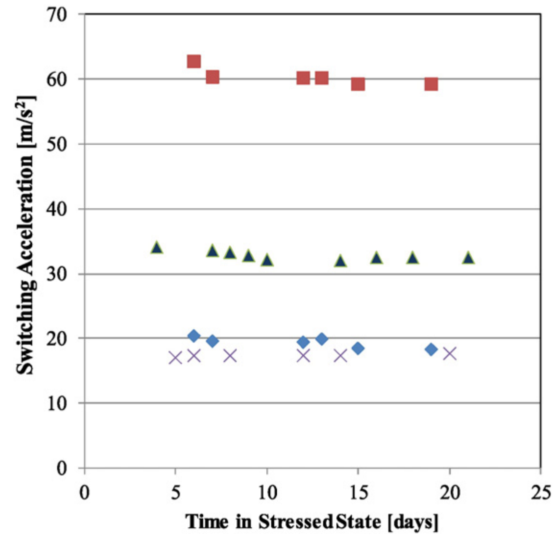

Fig. 7. Average switching (threshold) acceleration for four different molded accelerometers measured over an extended period. Each of the symbols represents a different device and they were left in the stressed bistable position (Fig. 5b) in the time between tests.

the flexor elements as the switches were loaded and unloaded from the testing centrifuge.

Each individual switch tested showed very low shift in the acceleration threshold over time; however, Figs. 6 and 7 both show considerable variation in switching acceleration from switch to switch. For the cut and bend switches, the maximum variation was about 23% compared to the average switching acceleration of about 17 m/s^2. For the molded switches, the maximum variation was about 86% compared to the average of about 32 m/s^2. To better understand the source of this variation, we modeled how the switches would respond to variations in the flexor angle in the switch. Induced strain due to cooling of the molten tin may also be a source of variation for the molded switches. We modeled both of these effects using the model previously described [15]. Briefly, this model solves for the bending of the thin beams using an elliptic integral approach to solve the governing differential equation. The resulting equations are solved numerically by applying a given displacement to the beams, and calculating the forces that result. Note that more complex models have also been developed for similar bistable mechanisms [16,17] but we have found the more simple model is sufficient for understanding most of the performance variations we measured.

Based on the model in [15], Fig. 8 shows the percent change in switching angle for a single flexor in the "cut and bend" design if that flexor is at a different angle than the designed angle of 4°. The data was generated using a model of flexor motion. The results show that, for this design, the force can vary by as much as about 23% for a deviation in flexor angle of only 0.75°. Analysis of the leg angles shown in Fig. 3a suggests that, in this image, the flexors vary by as much as about 0.6° compared to the designed value. Hence, variation in flexor angle likely accounts for most of the variability seen in Fig. 6.

Similarly, Fig. 9 shows the percent change in switching angle for a single flexor in the molded design at a different angle compared to the average leg angle of about 6°. For this design, the force can vary by about 38% for a deviation in flexor angle of about 2°. Analysis of the

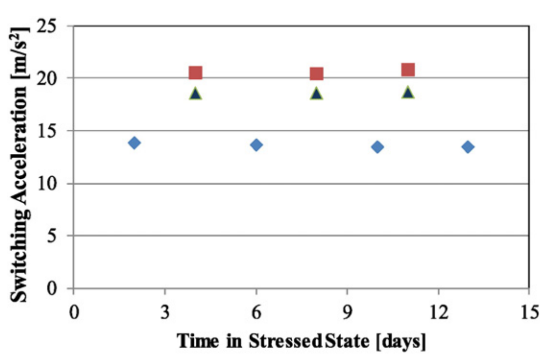

Fig. 6. Average switching (threshold) acceleration for three different "cut and bend" accelerometers measured over an extended period. Each of the symbols represents a different device and they were left in the stressed bistable position (Fig. 3b) in the time between tests.

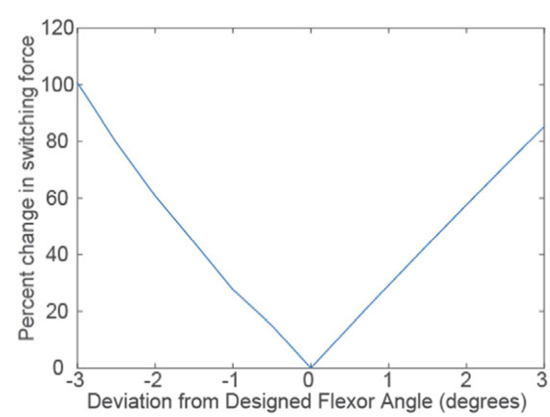

Fig. 8. The theoretical percent change in switching force for a single flexor at a different angle compared to the designed angle for the "cut and bend" design.

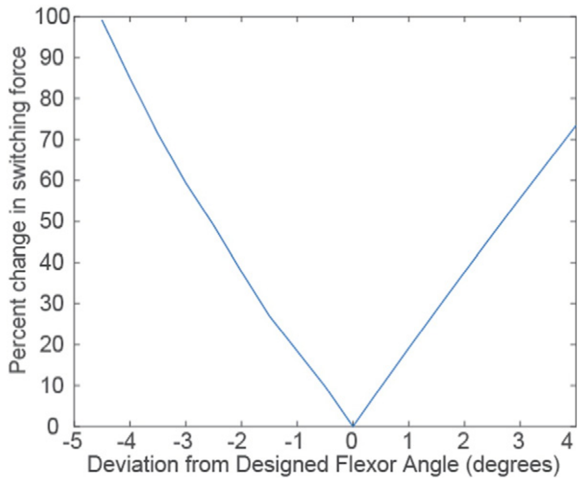

Fig. 9. The theoretical percent change in switching force for a single flexor at a different angle compared to the designed angle for the molded design.

leg angles on fabricated switches (such as the switch shown in Fig. 5a) shows that this amount of variation is typical for this design. Hence, flexor angle variation can account for much of the variation shown in Fig. 7, though not all. Other sources of variation are possible, however. For example, the switches represented by the triangle and the X's in Fig. 7 were fabricated by using a steel flux clean on their ends before pouring the molted tin, while the other two switches received no flux. The switches fabricated using flux seem to exhibit better bonding between the steel and tin, which may account for their lower variation with respect to each other.

The analysis shown in Figs. 8 and 9 suggests that the influence of small shifts in the flexor angle may be mitigated through design. Specifically, switches designed with a larger flexor angle will show smaller variation with small changes in leg angle. Therefore, we recommend that future device designs consider minimizing force variation with small changes in flexor angle.

The cooling of the tin during molding is an important source of induced stress in the molded design. As the tin cools, the outside frame of the system shrinks, inducing compressive strain in the thin beams. To model this effect, an additional horizontal strain was added to the model, and the force required for switching was calculated for the

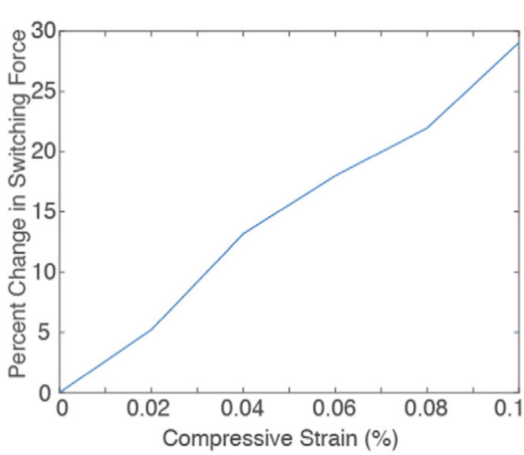

Fig. 10. The theoretical percent change in switching force for a single flexor under induced thermal strain for the molded design.

geometry of the molded design. The strain may be estimated by finding the thermal strain due to cooling of the tin. With a tin linear thermal expansion coefficient of $22 \times 10^{-6}\ K^{-1}$, and cooling of 200 K from the melting point, the estimated compressive strain is 0.044%. The results are shown in Fig. 10. Note that noise in the data is due to the numerical solution used.

Fig. 10 shows that at the estimated induced strain, the percent change in switching force relative to a switch with no induced strain is almost 15%. Hence, while this effect is smaller than the variation due to small changes in the beam angle, it may also be a significant contributor to the variation seen in Fig. 7. To minimize these effects, selecting flexors and bodies made from either the same metal or from metals with similar expansion coefficients would be preferable.

5. Conclusions

This paper presents the results of design, fabrication, and testing for two acceleration switch designs both based on the deformation of steel flexors. The use of steel minimizes the influence of stress relaxation, resulting in acceleration switches with very small drift over time. One design, which is amenable to stamping from stainless steel sheets, showed a threshold variation of 2.8%. The other design, based on tin molding around steel flexors, showed a variation of 14.9%. In both cases, switches were tested over a period of 2–3 weeks. By comparison, previous plastic switches have exhibited average drift of 54% over a period of only 72 h [13]. The acceleration switches demonstrated here could be used in a variety of applications requiring power-free measurement of threshold acceleration, including shipping or infrastructure monitoring. The switches are also easily adapted to use wireless readout using RFID technology [10].

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