Design of a smart bidirectional actuator for space operation

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Abstract

A common need for space borne instruments, satellites and planetary exploration payloads is the usage of compact, light and low power actuators. In the recent years, this need has been partially solved by the development of customized solutions with an increasing usage of smart materials. A linear bidirectional actuator based on shape memory alloy technology is presented in this work. The device has been conceived to lock the double-pendulum scanning mechanism of a miniaturized Fourier transform spectrometer for planetary observation. The mechanism class is that of pin pullers, with the pin locking the movable components of the spectrometer during launch and landing phases. The proposed mechanism, differently from available off-the-shelf devices, allows multiple actuations without the need of manual resetting. Moreover, the device requires to be powered only to change its status. An appealing feature of the adopted concept is that the actuation is intrinsically shock-less, a key requirement for deployment of devices sensitive to mechanical vibration and shocks. All these characteristics, in addition to the design flexibility of the proposed concept in terms of achievable forces and strokes, make the designed actuator promising for many different applications, from space to ground. The designed bidirectional actuator provides 0.6 mm stroke and a 50 N preload but it represents just an example of implementation for the proposed concept. Structural design of the functional elastic components
and SMA alloy characterization have guided the actuator development. A mockup of the actuator has been manufactured and the predicted performances preliminary validated.

**Keywords**

SMA, bidirectional actuator, compliant structure, resettable, pin-puller, holding mechanism, space application, Mars, FTS.

### 1 Introduction

It is well recognized that SMAs provide some advantages in mechanisms and actuators design, i.e. simplicity of related mechanisms, low driving voltage and sensing capability [1]. These characteristics have been recently exploited in different fields, i.e. robotics [2], industry [3] and aeronautics [4] to develop low power linear and rotational actuators. SMAs are very attractive for space applications as well, where SMA technology usage is quite recent [5, 6] and mainly focused on the actuation of deployable systems or damping system for spacecraft antennas [7]. In fact, SMAs allow control of the deployment process and thanks to the shock-less actuation have been used for developing low-shock release devices (LSRD). A clear advantage is that no additional damping is required to suppress shocks and vibrations due to the deployment, unlike the systems based on elastic energy storage. Example of space designed LSRD can be found in [8] where SMA and steel springs work in contrast to achieve automatic resetting [9]. Some additional examples of LSRDs can be found in [10-12], where SMA wires are used as triggers for the actuation. Moreover, thanks to the advantages of the SMA technology, some recent customized spaceborne applications can be found in gas release mechanism [13] or rock splitters [14].

The advantages of the SMA materials have been also exploited to develop low power linear and rotational actuators [15]. Available spaceborne devices provide mass ranging between 4 and 40 g and output force between 10 and 50 N. These actuators exploit SMA wire working against bias springs, added to recover the initial configuration once SMA is not heated. This is a relevant limitation for a
general space application and in particular for instruments mounted on planetary rovers, given that
the deployed configuration requires a continuous power consumption.

Paraffin based actuators have been used in space as well, thanks to their reliability, large strokes (up
to 13 mm) and high forces (about 100 N). Unfortunately, the actuator mass is not negligible, no less
than 80 g. Piezoelectric actuation has been evaluated as well among the existing solutions, thanks to
the accuracy of the output motion, the large stuck forces and the compatibility with low temperature
and vacuum environment. Anyway, the provided output displacement is generally limited to few
microns or, to achieve larger strokes, the actuator mass becomes quite large.

Thus, SMA technology was identified as the most promising for the development of an innovative
locking mechanism for the scanning pendulum of a miniaturized FT spectrometer 140x140x120 mm³
in size, 1 kg mass, designed to be mounted on a rover for Mars [16]. Holding was needed to keep the
pendulum in a safe position during launch and rover landing phases. This mechanism has become a
key component in any proposal for miniaturized FTS that has been conceived since then [17, 18]. As
evidenced by market and literature review, no multiple actuation devices compatible with our design
constraints are available, because they are either too massive or exceeding the size and power budgets.
Moreover, most of them are qualified down to -40°C, an operational limit not compatible with
environmental requirements for planetary surface operation. Exomars mission mechanical
environment had also challenging requirements due to the expected acceleration peak at landing on
Mars.

The developed actuator has many positive features: it provides two stable positions, requires power
only to change its status, uses only solid lubricants, can withstand at least one thousand working
cycles without relevant performances degradation and is capable of working at low temperatures. The
actuator concept is provided in the following whereas thermo-mechanical design is presented in
Section 2. SMA wire characterization and actuator preliminary testing are provided in Section 3 and
Section 4 eventually completes the paper.
1.1 Actuator concept

In order to understand the working principle of the proposed actuator two key points have to be highlighted:

- Actuating force and displacement are provided by SMA wire contraction as consequence of the material heating; and

- The actuator stroke is achieved by means of a displacement amplifier based on selectively compliant element that amplifies the SMA wire contraction without frictional elements.

A sketch of the actuation concept is shown in Figure 1. SMA wire is wrapped around an elastic support of diameter D and since the wire has been previously deformed, thanks to the material memory effect [19], it recovers its original shape once heated. The wire contraction reduces the support diameter and a new configuration (with diameter D’) is achieved, resulting in the output stroke provided by the actuator (as consequence, the elastic support extends from L to L’). The concept is simple and reversible, since a pushing force applied to the elastic support stretches the wire to its original configuration once the yield stress is exceeded.

The elastic support design has a key role in the actuator performance since the output displacement and force can be amplified by proper geometry selection. In SMA actuators design, displacement amplification is a recurrent problem, developed mechanisms can be found in various references [20, 21]. Authors in [21] design a passive elastic system to increase the output stroke by about a factor 2.5 and achieve constant force during actuation. Mechanism proposed in literature cannot be implemented in our case, because of our strict requirements of lightness, compactness and avoidance of lubricants. Thus, the wire contraction amplification has been achieved with selectively compliant element described in the next section. A preview of the final geometry is shown in Figure 2. The elastic supports are derived from a thin disk slotted in eight sectors. Two disks are kept together by an additional elastic structure that allows the initial positioning and avoid unwanted motions of the disk sectors during the actuation. The proposed system has an intrinsic higher stability than planar
devices and allows the implementation of friction less joints, a mandatory requirement for space and vacuum applications. Thermally and electrically insulating sectors are glued over the cylindrical structure between the wire and the disks. These are cylindrical as well and guide the wires avoiding any axial sliding during the actuation. Finally, two crimping elements are used to transmit the wire load during the actuation.

![Figure 1 Actuation concept: wire heating varies elastic support geometry providing required output force and stroke.](image)

The system described in Figure 2 could be used alone as single shot actuator since it provides a stable open position once the SMA has been heated. However, no autonomous multiple actuations would be possible with this configuration because restoring of the initial position requires an external work. In resettable systems, this is achieved manually or in multiple actuations devices thanks to a bias spring that sets back the system when the SMA element cool down [15]. The latter configuration is simple and reliable but as previously mentioned, the need of power to keep the actuator in open configuration is often not acceptable, surely it would not be in our case where actuation would require
almost the whole instrument allocated power. Thus, in our design two single shot actuators are mounted in opposition. By facing two components of the type of that in Figure 1, and playing with the alternate powering of the two actuators, the final configuration allows either amplification of the output stroke and realization of the bidirectional actuator.

1.2 Bidirectional actuator

A section view of the designed bidirectional actuator is shown in Figure 3. Different elements are present:

- a hold-down pin locking the spectrometer pendulum; and
- a bias spring, providing the required preload on the pendulum; it has to be noticed that the spring force is used only to provide the locking action and once the system is in open position, it gives a static load to be overcome by the actuator 1; and
- two single-shot actuators facing each other; and
- SMA wire wrapped around the insulating supports; number of wires turns was determined to be 2 for the bottom and 1 for the top actuators, respectively; and
- an aluminum frame, to complete the assembly.

Figure 3 Section view of the bidirectional SMA actuator.
The two single-shot actuators (hereafter named actuator 1 and 2) are mounted in opposition to realize the bidirectional configuration. Schematic of the intended opening/closing phases is shown in Figure 4.

Figure 4 (a) Schematic of the actuation to achieve open stable position (b) backward actuation.

When actuator 1 is heated, the hold-down pin moves upwards. A complete release is possible only if actuator 1 overcomes the spring bias force and stretches actuator 2 wire. Once heating of the actuator 1 is switched off, the pin remains in the stable open position since the force provided by the bias spring is lower than the one required to deform the wire 1. This is achieved by sizing the cross section and number of wire turns of actuator 1.

Figure 4b shows the locking procedure. The system starts in the stable open position as result of the previous actuation. Wire 2 is heated, recovering its shape and providing the force required to move back the system. Wire 2 heating is switched off when actuator 1 is completely stretched back and the
hold-pin is locking the scanning mechanism. The bidirectional actuator is reset and ready for a new cycle. In the following section detailed design of the single-shot actuator is provided. In particular, disks geometry will be defined accounting for the motion phases and design requirements and the single-shot actuator design will be optimized. To achieve required movement, actuator 1 and 2 comprise 2 and 1 turns of SMA wire, respectively.

2 Thermo-mechanical design

2.1 Design requirements

The actuator was originally developed for the Mars Infrared Mapper (MIMA)[22], a miniaturized infrared spectrometer payload of the 2007 configuration of the ExoMars high-mobility rover devoted to Mars surface observation. Unfortunately, a mission redesign aimed to mass and cost reductions stopped MIMA development since the instrument mineralogical and atmospheric science was regarded of minor importance with respect to the main exo-biologic mission’s objectives. Despite that, MIMA was a lucky pick-up for the mechanism requirements definition; the extreme temperature range of the Martian environment, strict mass and power limitations of a rover mounted instrument, the strong dynamic loadings due to the landing phase, the cleanliness and shock-less request associated to the application on an interferometer, made it already compatible with any following proposed usage [23]. Design requirements for the single-shot actuator are summarized in the following:

- A linear displacement of 0.6 mm to unlock the instrument scanning pendulum; and
- a holding force of 50 N, (required to warrant the locked position under the quasi-static acceleration of 1000 m/s² at landing on Mars); and
- mass (derived from the single-shot off-the-shelf actuator) of 15 g and volume limited to 40x40x20 mm³; and
- 7W maximum power consumption; and
• survival temperature range between -120 and 40 °C, operating range between -80 and 40°C;

• no usage of organic lubricants.

2.2 Kinematic model

Forward kinematics allowed selection of the disk parameters to achieve required stroke, i.e. disk external diameter and initial inclination angle. Disks shown in Figure 2 are symmetrical, therefore two hinged rigid beams allow modelling the kinematic of each sector. The model is shown in Figure 4. Moreover, considering that a symmetry exists also in the horizontal plane of the actuator, the model is further simplified focusing on half actuator. Sketch of the simplified model is provided in Figure 5.

Knowing actuator geometry, i.e. inclination angle \( \alpha \) and actuator diameter \( D_i \), vertical displacement is obtained as:

\[
\Delta h = \sqrt{\left( \frac{D_i}{2 \cos(\alpha)} \right)^2 - \left( \frac{D_i}{2} - \Delta s \right)^2} - \frac{D_i}{2} \tan(\alpha)
\]  

(1)
\[ \Delta s \] is the actuator radial displacement that depends on the SMA wire contraction. As worst case, \( \Delta s \) was restrained to 2\% of the initial radius. Moreover, this value allows up to \( 10^5 \) cycles of actuation considering the SMA aging with cycling. Besides, this limit well above the required value for the intended application and most locking devices. The static analyses performed are bound to verify the feasibility of the proposed concept. In Figure 5, results of the kinematic analyses shows that by increasing the wrapping diameter the output stroke increases as well. Moreover, in order to maximize the amplification, disk inclination should be minimized. However, minimizing the angle reduces the force exerted on the pin especially at the beginning of the actuation. In fact, the force \( V \) (along the direction of the pin) and the radial one \( H \) (related to the wire contraction), are linked by:

\[ H = \frac{V}{\tan(\alpha)} \]  

(2)

where \( \alpha \) is the inclination angle of the disk. Equation 2 has been used in the following to compute tensile stress on SMA material in the different actuation phases. The trade-off between output stroke and force led in our case to define the preliminary disk geometry with an inclination angle of 27\(^{\circ}\) and initial diameter of 25 mm. Un-deformed disk diameter was set 28 mm and the disk was divided in 8 sectors. The kinematic analysis evidenced that even in the worst case of minimum wire contraction (i.e. 2\%), the actuator still provides the required output displacement of 0.6 mm. This result had been verified in the following by FE analyses on the actuator. Models have been developed with PTC Creo Simulate software © PTC Inc.

### 2.3 Disk design

Disks design has been performed developing a FE model based on shell elements (elements 4782 and 4610 nodes) as shown in Figure 6.
Figure 6 (Left) Disk FE model and (right) geometrical parameters.

Disk initial configuration has been selected by the simplified kinematic analyses presented above, in which the disk sectors were considered as stiff elements. Disks thickness and cuts have to be defined in order to completely define the disk geometry. Figure 6 shows remaining parameters to be defined. In order to warrant resistance against expected loads in operative condition, buckling analyses were performed. Maximum wire pull has been considered (i.e. 98 N), considering austenite plateau stress of manufacturer datasheet [24]. Half disk has been considered; displacements in the horizontal plane and rotations have been left free for the external diameters whereas hinge constraint has been applied at the inner disk circumference. Two different materials have been considered, aluminum (Al7075T6) and titanium (Ti6Al4V) alloys. Material characteristics are summarized in Table 1. Geometry of the disk has been varied in order to minimize disk stiffness. This has been achieved with optimization analyses having set maximum Von Mises stress for each material and minimum buckling coefficient, as for ECSS design [25].

<table>
<thead>
<tr>
<th>Material Property</th>
<th>Unit</th>
<th>Al 7075 T6</th>
<th>Ti6Al4V</th>
<th>Macor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young Modulus</td>
<td>GPa</td>
<td>70</td>
<td>114</td>
<td>66.9</td>
</tr>
<tr>
<td>Density</td>
<td>kg/m³</td>
<td>2800</td>
<td>4430</td>
<td>2520</td>
</tr>
<tr>
<td>Poisson Ratio</td>
<td></td>
<td>0.33</td>
<td>0.33</td>
<td>0.29</td>
</tr>
<tr>
<td>Ultimate Tensile Strength</td>
<td>MPa</td>
<td>572</td>
<td>950</td>
<td>345</td>
</tr>
<tr>
<td>Yield Tensile Strength</td>
<td>MPa</td>
<td>380</td>
<td>880</td>
<td>n.a.</td>
</tr>
<tr>
<td>thermal conductivity</td>
<td>W/(m K)</td>
<td>156</td>
<td>6.7</td>
<td>1.46</td>
</tr>
</tbody>
</table>

Table 1 Material properties of the FE model.
Optimization results are summarized in Table 2. Buckling Von Mises stress distribution is shown in Figure 7.

<table>
<thead>
<tr>
<th>Thickness [mm]</th>
<th>d [mm]</th>
<th>r [mm]</th>
<th>d1 [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimization limits</td>
<td>0.35</td>
<td>1</td>
<td>0.5</td>
</tr>
<tr>
<td>0.6</td>
<td>2</td>
<td>1</td>
<td>5</td>
</tr>
<tr>
<td>Ti6Al4V optimum</td>
<td>0.35</td>
<td>1.5</td>
<td>0.75</td>
</tr>
<tr>
<td>Al7075T6 optimum</td>
<td>0.38</td>
<td>1</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Table 2 Buckling analyses, geometry of the disks.

It can be seen that the titanium alloy allows for thinner elements. This is expected because of the higher mechanical strength and stiffness. In order to evaluate the force budget for the actuator feasibility verification, stiffness of the disk has been evaluated by means of FE model. In fact, the elastic deformation of the actuator due to the movement during wire contraction gives a resistance to be overcome. Two contributions are evaluated; these are related to the deformation of the disks once wire is recovering (upwards movement) or when a force from the top is applied (downwards movement). The latter force is present when actuator 2 is positioning back the pin. Thus, a FE model with two disks has been developed: disks have been matched with weighted links to allow relative rotation of the disks during the simulated conditions. A static analysis has been performed, with 1 N loading in radial direction. Deformed configurations is shown in Figure 8. In table 3 are summarized computed radial and vertical stiffnesses.
Figure 8 Disks deformation with radial loading of 1N.

<table>
<thead>
<tr>
<th>Material</th>
<th>Radial stiffness [N/mm]</th>
<th>Axial stiffness [N/mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ti6Al4V</td>
<td>5.15</td>
<td>10.26</td>
</tr>
<tr>
<td>Al7075T6</td>
<td>6.90</td>
<td>13.89</td>
</tr>
</tbody>
</table>

Table 3 Single-shot actuator stiffnesses.

Comparison between candidate materials allowed evidencing that titanium alloy is the chosen material thanks to the lower axial stiffness while providing larger safety margin during actuation.

It has to be reminded that a complete actuation is defined by three different phases:

- Opening phase: the pin puller unlock the pendulum and open position is achieved; beside the bias spring force, deformation of the disks and wire 2 deformation have to be overcome; and

- Static phase: the system is stuck and the movement is prevented by the resistance to deformation of wire 1; and

- Closing phase: the actuator 2 is activated, wire 1 is deformed again with the contribution of the force due to the bias spring; the actuator is reset and ready for the next actuation cycle.

According to ECSS design standard [25], friction force has to be considered with safety factor 3, elastic ones have to be multiplied or divided by 1.2 depending on the role of resistance or motor in the actuation. Forces can be computed for each phase and depend on the disk design, whose geometry
and material have been selected on the basis of the performed analyses. Resulting stresses in the final configuration for the wires of the actuator 1 and 2 are summarized in Table 4. The detailed computation of the stresses and the forces in each movement phase can be found in [26].

<table>
<thead>
<tr>
<th>Movement Phase</th>
<th>Actuator 1 [MPa]</th>
<th>Actuator 2 [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Opening</td>
<td>277.29</td>
<td>82</td>
</tr>
<tr>
<td>Static</td>
<td>59.92</td>
<td>0</td>
</tr>
<tr>
<td>Closing</td>
<td>82</td>
<td>335.98</td>
</tr>
</tbody>
</table>

Table 4 Stress budget for actuator 1 and 2 wires.

The analysis evidence that actuation is feasible for each phase, so the external reset is no longer needed. In fact, austenitic and martensitic phases plateau stresses specified by SMA wire manufacturer, ranging between 70 and 650 MPa, are compatible with required tensile stresses during each actuation phase. Anyway, the SMA wire selected for the actuator development underwent an individual characterization, which is described in the following.

2.4 Actuator design

A FE model of the actuator 1 was built in order to verify the output displacement based on solid tetrahedrons and shell elements (44047 elements and 36708 nodes). The FE model comprises the two identical disks linked together, the elastic structure and the insulating supports. As previously mentioned, the rotation between the disks border is allowed. Forces and constraints are shown in Figure 9. Resistance load of 133.5 N is applied on the actuator and a radial force of 70 N resulting from the wire pull has been distributed over the insulating sectors.
The bias spring $h$ has been added on the top of the actuator, as well. A single-shot actuator has been optimized using as cost function the maximization of the output displacement $[27, 28]$; disk parameters in Figure 5 are left free. Figure 10 provides output displacement with optimal configuration and Table 5 summarizes optimized disk geometry.

<table>
<thead>
<tr>
<th>Thickness [mm]</th>
<th>$d$ [mm]</th>
<th>$r$ [mm]</th>
<th>$d_1$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.283</td>
<td>1</td>
<td>0.5</td>
<td>4.08</td>
</tr>
</tbody>
</table>

Table 5 Optimized single shot actuator parameters

A thermal model of the actuator was developed with the aim of determining the power required to actuate the SMA wire in cryogenic condition. In fact, mission thermal analysis evidenced that in the case of cold Martian condition, expected lowest temperature is about $-120 \, ^\circ C$ $[23]$. Actuators have
been thermally linked together by means of null thermal resistance in order to simulate the worst case. In fact, increasing the thermal resistance would reduce the power supply required for the actuation. Moreover, radiation heat exchange has been imposed on the disks, insulating supports and wire. The disks emissivity has been set to 0.05, assuming a gold coating of the titanium disks while unitary emissivity has been assumed for the insulating ceramic supports. In order to reduce the model complexity, wires have been modelled by separated circles on which heating has been uniformly distributed, simulating Joule heating effect. The two extreme conditions “cold” and “hot” have been analyzed; radiative and conductive interfaces have been set to -120°C and 40°C, respectively. The predicted temperature distributions in cold and hot cases are shown in Figure 11. The transformation temperature of 90°C is achieved on actuator1 heating the wire with 1.86 and 0.62 W, for cold and hot case respectively. These values are fully compliant with the mission requirements. Moreover, it can be seen that in the simulated cases temperature of the wire2 is always by far than the one required for complete austenite transformation, so there is no risk of simultaneous actuation of the two wires that would lead to a failure.

Final configuration for the single shot actuator has 7g mass and radial size of 34 mm and height of 15mm.
3 Experimental activity

3.1 Wire characterization

SMA wire from Memory-metalle GmbH/ Saes Group has been purchased for the actuator development. Material characteristics have been measured to identify the transformation temperatures in operative conditions and the stress-strain curves for the material different phases. DSC Seiko calorimeter model 220 has been used for the calorimetric characterization. DSC samples with 20 mg mass have been tested between -50 and 150°C. Temperature variation has been performed with 10 °C/min rate and three cycles have been done. Specific heat flux curves are provided in Figure 12.

![Figure 12 Results of the DSC testing.](image)

MTS testing facility allowed functional testing to determine stress-strain curves in the martensitic and austenitic phases. Material austenitic and martensitic states had been obtained by means of thermal chamber at high (about 100 °C) and low (about RT) temperatures respectively. Tensile loading/unloading cycles were performed, at constant temperatures, with 25 mm strain gauge extensometer and a preload of 2N. Deformation rate was set to 2 mm/min with maximum deformation of 3%. Temperature variation between 22 and 24 °C was accepted for the martensite testing whereas 95°C was the kept as minimum for the austenitic phase characterization. Results of the tensile tests are shown in Figure 13.
Figure 13 Functional testing: martensitic (left) and austenitic (right) states.

Table 6 provide mechanical characteristics derived from stress-strain curves.

<table>
<thead>
<tr>
<th>Test ID</th>
<th>Elastic Modulus [GPa]</th>
<th>Plateau Stress [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1_M</td>
<td>28</td>
<td>80</td>
</tr>
<tr>
<td>2_M</td>
<td>43</td>
<td>80</td>
</tr>
<tr>
<td>3_M</td>
<td>30</td>
<td>70</td>
</tr>
<tr>
<td>4_A</td>
<td>73</td>
<td>602</td>
</tr>
<tr>
<td>5_A</td>
<td>70</td>
<td>628</td>
</tr>
<tr>
<td>6_A</td>
<td>63</td>
<td>560</td>
</tr>
</tbody>
</table>

Table 6 DSC and functional testing transformation temperatures. Units are °C.

Strain recovery heating/cooling loops, under constant load (simulating working condition with dead mass of 5 kg, i.e. nominal tensile stress of 250 MPa) and temperature cycling has been performed in a thermal chamber (ACS Angenlantoni Industrie type), equipped with a LVDT (Linear variable differential transducer) to measure the wire deformation under heating and cooling. Temperature variation rate was set to 2°C/min between -50 and 150 °C and 10 cycles between minimum and maximum temperatures had been performed. Results are shown in Figure 14.
Obtained transformation temperatures are compared with DSC results in Table 7.

<table>
<thead>
<tr>
<th>Temperatures</th>
<th>DSC</th>
<th>1st cycle</th>
<th>2nd cycle</th>
<th>10th cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ms</td>
<td>-3.6</td>
<td>38.46</td>
<td>38.51</td>
<td>46.52</td>
</tr>
<tr>
<td>Mf</td>
<td>-45</td>
<td>32.34</td>
<td>35.68</td>
<td>43.20</td>
</tr>
<tr>
<td>As</td>
<td>39</td>
<td>87.31</td>
<td>87.07</td>
<td>85.74</td>
</tr>
<tr>
<td>Af</td>
<td>57</td>
<td>89.63</td>
<td>88.61</td>
<td>87.66</td>
</tr>
</tbody>
</table>

**Table 7** DSC and functional testing transformation temperatures. Units are °C.

3.2 Discussion

Heat peaks in Figure 12 identify phase transformation temperatures. During cooling two peaks have been found, since between martensite and austenite the phase R is present. During heating just one peak is found since the transformation peaks are coincident. Transformation temperatures, summarized in Table 7, have been derived looking to the intersection between the tangent of the DSC curves at the measured peaks. Austenitic phase is achieved at relatively low temperature, about 60 °C. This is important for our design, since a low heating power is expected even in cryogenic condition (as confirmed by thermal modelling). As expected, for a non-trained wire, there is not a thermal cycling stability particularly for the firsts heating/cooling cycles (see DSC cooling curves) [29]. However, after three DSC scans a stabilization is achieved. Typical thermal hysteresis (about 40°C) of NiTi wire for shape memory application is shown also.
Figure 13 depicts mechanical behaviors of the material and the characteristic plateaus related to the de-twinning of the martensite (low temperature) and stress induced martensite (high temperature) are well shown. Mechanical modulus as well as stresses plateau values are summarized in Table 6. It can be seen that measured values are compatible with datasheet specifications and are in agreement with previous literature studies. For the martensitic phase the elastic modulus varies between 28 and 43 GPa and plateau stress is varying between 70 and 80 MPa. For the austenitic phase, the elastic modulus and plateau stress increase up to 70 GPa and 600 MPa, respectively. Worst case combination of the measured mechanical properties had been used in the feasibility and optimization design phases.

Finally, except for the first cycle, the wire under constant load of 250 MPa exhibits a stable functional performance after few heating/cooling cycles (see Figure 14). This means that with proper training (lasting no more than 10 cycles), the wire is ready for the actuation. Another important behavior is that the transformation temperatures change as consequence of the applied stress. Table 7 evidences that a general increase of the temperature is obtained. In particular, \( M_t \) is about 40°C and \( A_f \) achieves 87 °C. The latter values had been used as reference for the design described in Section 2.

### 3.3 Actuator Testing

With the aim of verifying the design and highlighting drawbacks and possible improvements, a mock-up for the single actuator has been realized. Figure 15 shows the single-shot actuator breakdown and the experimental setup.
Actuating force was measured by a load cell (range 150 N, linearity 0.5% of the measurement range) mounted between the hold-down pin and the ground. Displacement has been measured by laser Micro Optronic ILD 1400-05 (range 5mm, maximum linearity error 9 μm). The single-shot actuator was loaded with a dead mass of 10 kg (as worst case loading) and powered with a constant current of 1.8 A, value was selected to achieve complete transformation but avoid the wire overheating. Figure 16 shows measured force and displacement during an actuation cycle.

Once that the actuation starts, the hold-down pin reaches the maximum displacement of 0.9 mm after about 140 s. This is confirmed by the load increasing, due to the spring element, up to 110 N.
Once the electrical power is switched off, the system comes back to the initial position in about three minutes. This is expected since the load is larger than the one that causes system closing. Moreover, the measured stroke is higher than the nominal 0.6 mm. This result was expected too, because the actual SMA recovery was larger than the 2% value, assumed as end of life figure in the design. This happened despite the wire had been “aged” with more than 1000 cycles.

The obtained result evidenced the need of an accurate control of the actuator geometry and wire contraction, to match the expected performances. As positive outcome, the obtained result also highlighted the intrinsic flexibility of the proposed concept that can be adjusted to different displacement requirements with small changes of the starting wire length.

4 Conclusions

A light and small bidirectional actuator was designed to lock the mechanism of a miniaturized infrared spectrometer. The actuator is based on SMA technology and can provide 0.6 mm output displacement against 50 N loading. The actuator is based on elastic structures whose geometry can be changed to face different requirements in terms of force or displacement, which amplify the contraction of the SMA wire. The bidirectional working is achieved without requiring power either than that necessary for the position commutation. Moreover, the actuator concept flexibility can be exploited to match different working condition, i.e. by changing the wire diameter to achieve higher forces or the actuator geometry to allow larger displacements. A commercially available wire has been selected and tested to develop a mockup of the proposed actuator. Testing of the manufactured mockup allowed preliminary validation of the proposed concept. Future activity is foreseen to fully characterize the designed actuator in terms of force vs displacement performance in more general working conditions; this would enable evaluating the actuator suitability for applications different from the designed locking system.


