Compliant mechanism for ankle rehabilitation device. Part II: optimization and simulation results

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Abstract. As rehabilitation robotics become more popular, a great amount of research is focused on safety measures. In the design of an ankle rehabilitation device multiple safety techniques must be embedded. A solution is represented by a compliant mechanism that could take the overload from the motor, in case of emergency, and not further damage the patient’s foot. In this paper two designs of compliant mechanisms are studied with the help of finite element analysis, providing two solutions of elastic elements.

1. Introduction
The integrity of the human ankle joint complex is critical for everyday activities. The high frequency of ankle injuries is explained by the fact that at this level multiple movements occur. After the initial treatment and rest period some recovery exercises are required, performed with the help of a therapist. But the exercises are long-term, repetitive and require constant supervision. To combat these drawbacks some rehabilitation devices were developed, that offer the required movements for a complete recovery.

As robotic rehabilitation field grows larger the systems must fulfil more and more requirements. For every device safety plays a major part in design and control. Safety measures for rehabilitation devices founded in literature can range from stop buttons, to automatic stop control achieved through software programming. As an ankle rehabilitation device that offers two degrees of freedom was proposed in previous work ([2]-[3]), some safety measure are required. A stop button is implemented that allows the therapist to interrupt the rehabilitation exercise. As extra safety measures are necessary, a compliant mechanism is studied, that will receive the extra load and will not allow further damage of the ankle joint.

Some compliant mechanism for rehabilitation devices were developed around the world ([7]-[10]), but they are not offering the necessary requirements for our device. As main specifications one can mention: range of motion, operating live, accuracy, the ability to be integrated into other subsystems and light weight. Starting with these considerations a new compliant device for ankle rehabilitation is presented in this paper.

2. Design and optimization
The mechanical design of the ankle rehabilitation device was conceived in order to obtain lightweight and easy to use mechanism. The device should provide two degrees of freedom (DOF) for dorsiflexion/plantar flexion movement and inversion/eversion movement. These DOF are obtained
with the help of a spatial quadrilateral mechanism that is actuated using two servomotors. In order to prevent the overloading of the ankle joint a compliant mechanism is required to be positioned between the crank and the motor (Figure 1(a)-green represents the emplacement of the elastic element).

In designing the elastic element the first step is represented by the systematic synthesis. After kinematic approach a structural optimization is required ([11]). This allows one to design the compliant mechanism that satisfies an objective function for a set of parameters and constrains. One of the main methods consists in searching for the mechanical advantage, geometrical advantage and mechanical efficiency. These formulations refer to the functional specifications rather than their structural requirements, allowing maximizing the required functions. This can be achieved through finite-element analysis where loads and displacements can be studied.

An ankle model for the passive moment-displacement relationships under static conditions was developed by Jamwal et al. [12] and maximum ankle moments were found between -10 Nm and 7 Nm. Based on this studies the compliant mechanism must offer a maximum of 15 Nm torque, after this value one can assume that the exercise is becoming painful for the patient. Since the crank position versus motor position is required, a potentiometer can be placed between the elastic element and the crank (Figure 1(b)).

The elastic system will consist of: (1) a servomotor (that already has encoders in order to determine the position); 2) an adaptable shaft coupler (provided through servomotor’s vendor) that allows the extension of the shaft; 3) a shaft that provides support for the potentiometer; 4) the elastic element; 5)
a rotary potentiometer (that is connected to the other end to the crank); 6) a casing that contains all the
above components, with one end connected to the elastic element and the other end connected to the
crank (Figure 2).

The elastic element received a disc-shaped design, in order to minimize weight and size. It is
represented by a monolithic disc, where two rings are interconnected by flexible elements. A
symmetric structure was chosen in order to assure that the input torque will produce a rotation of the
inner ring, without a translation. Another requirement is represented by the ability to rotate in both
directions, with same amplitude (in order to obtain the required motion: flexion/extension and
inversion/eversion).

3. Simulation results

Multiple designed were pursued, based on cartwheel hinge and the double spring Archimedes spiral.
Dimensions of the inner and outer ring were established based on the space requirements and shaft
dimension and have the following values: \( r_1=35 \) and \( r_1=90 \) mm (two studied values), \( r_2=6 \) mm. Stress
distribution and stiffness depend on optimizing the thickness \( d \) and dimension of the outer ring.

3.1. Cartwheel hinge design and simulation

Several variation of the cartwheel designs were studied centralized in Table 1. A FEA analysis was
performed using ANSYS Student software, to ensure the maximum stress is less than the yield stress
of the material, and to observe the maximum deformations. The analysis consisted in fixing the outer
ring (as it would happen when the patient would oppose the movement, due to pain) while torque (15
Nm) was applied tangential to the inner ring. For each design four materials were analysed from
programs’ database, their characteristics are shown in Table 2. Also the rotation of the inner circle
versus the external circle was measured, in degrees.

| Table 1. Numbering of the proposed compliant mechanism |
|------------------------------------------------------|
| Dimensions [mm] | Cartwheel hinge/4 elastic elements | Cartwheel hinge/8 elastic elements | Cartwheel hinge/variable width |
| \( r_1=35 \) | A1 | B1 | C1 |
| \( r_1=35 \) | A2 | B2 | C2 |
| \( r_1=90 \) | A3 | B3 | C3 |
| \( r_1=90 \) | A4 | B4 | C4 |

| Table 2. Main characteristic of programs’ database materials |
|-----------------------------------------------------------|
| Name                          | Tensile Yield Strength [MPa] | Tensile Ultimate Strength [MPa] | Young’s Modulus MPa |
| Aluminium Alloy              | 280                          | 310                          | 71000               |
| Magnesium Alloy              | 193                          | 255                          | 45000               |
| Stainless Steel             | 207                          | 586                          | 193000              |
| Titanium Alloy              | 930                          | 1070                         | 96000               |

The first studied design is the cartwheel hinge presented in Figure 3a,b and it is characterized by
length \( l=11.2\)mm, depth \( d=5\) mm and constant width \( t=3\) mm. The \( \phi \) angle (45\(^{\circ}\)) is a very important
parameter because it affects the stiffness and precision of the hinge. As demonstrated in [8] the
stiffness of the cartwheel hinge could be reduced by increasing this angle. Stiffness is also related with
the material of the hinge and the section dimension of the element.
As it is shown in Figure 4 for this first design (A1) maximum equivalent (von-Misses) stress varied between 762.44 and 779.38 MPa (for the studied materials), with titanium alloy being the only material suitable (equivalent stress < tensile yield strength). The angular deflection of the inner circle for this material was only of 0.8259°. For this reason, the depth $d$ was increased to 10 mm (A2). Results showed a lowered maximum equivalent stress (449.13 to 463.05 MPa) but also lower angular deflection (maximum of 0.40618° for magnesium alloy). Increasing the radius $r_1=90$ mm and lowering the depth $d=5$ mm (A3) an increased angular deflection was obtained (maximum of 4.2899°) but with the disadvantage of high equivalent stress. The last studied version of this model (A4) had $d=10$ mm and $r_1=90$ mm, with the best results regarding the maximum equivalent stress. Still, the angular deflection showed low values, thus more designs were studied.

In order to increase the stiffness of the hinge the $\varphi$ angle was reduced to 22.5°, increasing the number of elastic elements to 8, to maintain a symmetric structure (Figure 5a,b). Same analysis conditions were used, with a fixed outer ring and an applied torque of 15 Nm to the inner ring (Figure 5c). The characteristic dimensions are length $l=11$ mm, depth $d=5$ mm and constant width $t=2$ mm.
Figure 5. Cartwheel hinge design, eight elements, constant width: (a) isometric view; (b) structural parameters; (c) analysis configuration

For this design (B1), following values were obtained (Figure 6):
- Magnesium Alloy: angular deformation of 1.7335° and equivalent stress of 404.86MPa;
- Aluminium Alloy: angular deformation of 1.1026° and equivalent stress of 409.46MPa;
- Stainless Steel: angular deformation of 0.40685° and equivalent stress of 413.99MPa;
- Titanium Alloy: angular deformation of 0.811° and equivalent von-Misses stress of 402.53MPa.

One can observe that the values for maximum equivalent stress are larger than tensile yield, resulting in deformations without returning to the initial shape. Titanium Alloy is the only material that has a lower equivalent stress value, but the deformations are very small and not easy to collect.

To decrease the values for equivalent stress depth was increased to \(d=10\ mm\) (B2). With these modifications the maximum equivalent von-Misses stress was found between 206.93 and 212.35 MPa, with a maximum angular deflection of 0.84259° for magnesium alloy. Although the two of the values obtained for stress (titanium and aluminium) are within the limits of yield strength, the deformation for these materials are too small to be noticeable.

Figure 6. Simulation results for cartwheel hinge with 8 elastic elements: (a) maximum equivalent stress; (b) angular deflection

Increasing the radius \(r_1=90\ mm\) and lowering the depth \(d=5\ mm\) (B3) some increased angular deflection values were obtained (maximum of 5.5665° for magnesium alloy) but with the disadvantage
of high equivalent stress (483 MPa). The last studied version of this model (B4) had $d=10 \text{ mm}$ and $r_1=90 \text{ mm}$, recording a 224.9 MPa stress for aluminium alloy (at 1.949° angular deflection) and 224.83 MPa for titanium alloy (at 1.4334° angular deflection), both materials presenting maximum stress within the limits of yield strength.

A new design is proposed, with constant width and variable thickness (Figure 7a,b) since the sensitive axis lies in the cross-section of minimum thickness where maximum bending compliance is present.

The length and angle have the same value as previous design ($l=11 \text{ mm}$, $\varphi=22.5^\circ$), the depth was maintained to $d=5 \text{ mm}$ and thickness is variable. The shape of the elastic elements is obtained by connecting circle $r_3=2\text{ mm}$ with circle $r_4=0.5\text{ mm}$ (situated at half length) and again at the base with another $r_3$ circle, with the help of a 3-point arc (C1). After the simulation the following values were obtained (Figure 8):

- Magnesium Alloy: angular deformation of 5.7913° and equivalent stress of 1169.9MPa;
- Aluminium Alloy: angular deformation of 3.6845° and equivalent stress of 1173.1MPa;
- Stainless Steel: angular deformation of 1.3601° and equivalent stress of 1175.8MPa;
- Titanium Alloy: angular deformation of 2.7091° and equivalent von-Misses stress of 1168.2MPa.

![Figure 7. Cartwheel hinge, variable width: (a) isometric view; (b) structural parameters; (c) analysis configuration](image)

None of the stress values situate in the required range, although deflection angles have greater values than the designs studied above. For this reason, the depth $d$ was increased to $10 \text{ mm}$ (C2). Results show a lowered maximum equivalent stress (569.41 to 576.39 MPa) with 1.306° angular deflection for titanium alloy (the only material with stress value in the required range). Increasing the radius $r_1=90 \text{ mm}$ and lowering the depth $d=5 \text{ mm}$ (C3) very large angular deflection values were obtained (maximum of 36.135°) but none of the equivalent stress values were lower than the yield strength. The last studied version of this model (C4) had $d=10 \text{ mm}$ and $r_1=90 \text{ mm}$, with titanium alloy the only material that has a maximum equivalent stress (840.01 MPa) lower than the yield strength value and has a deflection angle of 8.2206°.

For the cartwheel hinge design several models were found to be acceptable, with lower equivalent von-Misses stress than yield strength and with angular deflection of the inner ring versus the outer ring of minimum $2^\circ$: A3, B3 and C4, all based on titan alloy material. The material has a linear elastic deformation, presented in Figure 9. Although the stress value of this material is within range, the relatively small deformation and high acquisition price represent the main drawbacks, thus another design was proposed, based on double Archimedes’s spiral.
3.2. Double spring compliant mechanism

The double spring compliant mechanism, based on Archimedes’s coil, offers large displacements due to its shape. Multiple designs were studied, altering the thickness $d$ and width of the elastic area $t$. The main properties of the spiral are presented in Figure 10a,b. For the first simulation (D1) the following dimensions were used: $r_1=35\ mm$, $r_2=6\ mm$, $t=2\ mm$, depth $d=15\ mm$ and $n=1$ (number of coils). Simulation results show extremely large values for maximum equivalent stress (Figure 11). In order to lower this values some dimensions were modified to $r_1=90\ mm$, $r_2=6mm$, $t=3\ mm$, depth $d=20\ mm$, maintaining the same number of coils (D2). It can be noticed that the maximum equivalent stress has half of the value, with 716.57 MPa for titanium alloy and an angular deflection of 11.24 degrees.
Titanium alloy represents a rather expensive material. In order to obtain results with a more accessible material, some more dimensions were studied. If $t$ dimension is increased to $t=4\text{mm}$ an even lowered equivalent stress is obtained (632.16 to 636.89 MPa) but also lower angular deflection.

Another approach would lead to connect two elastic elements, in different configurations: with same coil beginning or with angle between the beginning of coils of 45 degrees and 90 degrees. The results are synthesized in Table 3. The largest angular deflections are obtained in the 90 degrees between he beginning of the coils configuration but also the equivalent stress rose. When the two elastic elements are placed in mirror configuration almost no deflection is recorded, thus resulting an unsuitable design. From these designs two models were found to be acceptable D2 and D4, both based on titanium alloy material. The angular deflection versus torque is presented in Figure 12.

Numerous simulations were performed and it was observed that an increase in the coil number (1.5 or 2) would lead to an increase of deflection values, but also an increase in equivalent von-Misses stress. The optimized shape would consist in a double spiral Archimedes’s coil (Figure 13), with a coil number of $n=1.7$ revolutions, pitch of the coil $22\text{mm}$, $r_1=90\text{mm}$, depth $d=50\text{mm}$, $t=4\text{mm}$. After the simulation the following values were obtained:

- Magnesium Alloy: angular deformation of $8.0303^\circ$ and equivalent stress of $276.77\text{MPa}$;
- Stainless Steel: angular deformation of 1.9146° and equivalent stress of 279.75 MPa;
- Aluminum Alloy: angular deformation of 5.1491° and equivalent stress of 278.26 MPa;
- Titanium Alloy: angular deformation of 3.7411° and equivalent von-Misses stress of 276.03 MPa.

Table 3. Different results obtained for the double spiral spring configurations

| Configuration | Material     | Angular deflection [°] | Von-Misses equivalent stress [MPa] |
|---------------|--------------|-------------------------|-----------------------------------|
| 0 degrees angle between the beginning of coils (D4) | Aluminum Alloy | 7.4566 | 352.15 |
|               | Magnesium Alloy | 11.623 | 350.34 |
|               | Stainless Steel | 2.774 | 354.04 |
|               | Titanium Alloy | 5.4137 | 349.45 |
| 45 degrees angle between the beginning of coils (D5) | Aluminum Alloy | 7.6152 | 379.54 |
|               | Magnesium Alloy | 11.918 | 378.79 |
|               | Stainless Steel | 2.8225 | 380.34 |
|               | Titanium Alloy | 5.5626 | 378.38 |
| 90 degrees angle between the beginning of coils (D6) | Aluminum Alloy | 7.6589 | 464.66 |
|               | Magnesium Alloy | 11.99 | 459.87 |
|               | Stainless Steel | 2.8381 | 469.18 |
|               | Titanium Alloy | 5.5968 | 457.36 |

Figure 12. Angular deflection versus torque for Archimedes’s spiral spring mechanisms:
(a) D2, D4; (b) D7

For this design (D7) two materials have equivalent stress lower than tensile yield stress: aluminum and titanium alloy. The relationship between angular displacements and torque, for these materials, is presented in Figure 12 b. This linear character will ease the control and calibration of the compliant mechanism. An aluminum mechanism would present advantage of a low cost and ease of replacement in case of deterioration. Future work will focus on practical validation of the chosen model.
4. Conclusions

In this paper some simulation results with finite element method, of different elastic elements for an ankle rehabilitation device, were presented. The elements are conceived from one building block, easy to manufacture, and their aim is to fulfil a set of requirements such as: compact lightweight design and compatibility with the rehabilitation device. The compliant mechanism should not allow the extra load from the motors to further damage the patients’ lower limb. Some solutions were found where the maximum stress is less than the yield stress of the material. Future work will focus on choosing the proper design and preparations for practical validation of the models.

5. References

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