RESUMEN

En este proyecto se presenta el diseño mecánico de un banco de pruebas para el sistema de actuación de una ortesis activa de rodilla-tobillo-pie (knee-ankle-foot orthosis en inglés) para lesionados medulares incompletos. El objetivo de dicho banco es caracterizar con precisión el comportamiento dinámico del sistema de actuación de la rodilla: estudiar su funcionamiento directo e inverso y la posibilidad de emplear materiales elásticos en alguna de las piezas que transmite el movimiento a la pierna.

Para la realización del diseño es necesario conocer los rangos de trabajo del sistema de actuación y las especificaciones mecánicas del banco de ensayo. Para ello, después de realizar una búsqueda bibliográfica sobre el estado del arte de los bancos de pruebas para ortesis de rodilla, se estudia el análisis dinámico de la marcha humana realizado por Sistiaga [2012] a partir de una captura en el Laboratorio de Biomecánica de ETSEIB (Escola Tècnica Superior D’Enginyeria Industrial de Barcelona). Una vez conocidas las especificaciones, se seleccionan los sensores adecuados para medir las magnitudes que permiten caracterizar el sistema de actuación.

El diseño del banco busca ser fácilmente adaptable para que se pueda utilizar no sólo con el sistema de actuación que se está desarrollando actualmente en el Departamento de Mecánica de ETSEIB, sino también con otros sistemas de actuación. Dentro del análisis del diseño mecánico, se comprueba la resistencia de los rodamientos empleados en la articulación de la rodilla y la resistencia de las uniones atornilladas más críticas.

Después de la fabricación de las piezas y del montaje del banco de pruebas, habrá que realizar unas pruebas para conocer los parámetros necesarios de cara a los experimentos posteriores: masa, centro de gravedad y momento de inercia de la parte del banco que hace las veces de pierna.

Finalmente, se lleva a cabo un diseño preliminar y una descripción de las pruebas a realizar en el banco de pruebas para caracterizar el sistema de actuación. También se analiza el impacto social y ambiental del proyecto, y el aspecto económico.
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This final degree project, which has the title *Design of a test bench to evaluate the dynamic performance of the actuation system of an active knee-ankle-foot orthosis*, has been done during an internship in the Biomechanical Engineering Group (BIOMEC) of the Department of Mechanical Engineering of ETSEIB (School of Industrial Engineering) at UPC (Universitat Politècnica de Catalunya). This work is a contribution to the project *Diseño de una ortesis activa innovadora para la marcha de lesionados medulares incompletos mediante métodos de análisis y predicción de movimiento y modelos músculo-esqueléticos complejos* of the National Plan of R+D (ref. DPI2012-38331-C03-02).

This project was approved by the Spanish Ministry of Economy and Competitiveness for the years 2013-2015, and it is being developed by the BIOMEC group in coordination with the Laboratory of Mechanical Engineering at University of La Coruña, and the Department of Mechanical, Energetic and Materials Engineering at University of Extremadura.

The UPC is responsible for the biomechanics and the mechanical design, the University of La Coruña simulates with multibody dynamics techniques, and the University of Extremadura designs the control system.

The work presented below is the extension of other final degree projects in which the two first prototypes of a SCKAFO (stance-control knee-ankle-foot orthosis) were built. The first project, *Modelo de ortesis activa SCKAFO para asistir la marcha de lesionados medulares*, was presented by Guillermo Arroyo in 2011. The second one, *Disseny mecànic d’una ortesi activa per a lesionats medul·lars a partir de l’anàlisi dinàmica de la marxa humana*, was presented in 2012 by Javier Sistiaga. This project continues the tasks of designing the third prototype of the SCKAFO in the Department of Mechanical Engineering of ETSEIB.
INTRODUCTION

Orthoses are exoskeleton devices that help many people with reduced mobility to improve their function to have a life similar to the one of a healthy person. Thanks to biomechanics, it is possible to study the pathological movement of a patient and the normal one for a healthy subject in order to design effective assistive devices.

In the case of an active knee-ankle-foot orthosis, it is designed for assisting human gait. During the gait cycle, there are two phases: stance and swing [Font-Llagunes, 2010]. In the stance phase, the orthosis has to lock the rotation of the knee and hold the whole body weight while the other leg is swinging. During the swing phase, the orthosis has to assist the knee flexion-extension rotation.

The aim of this project is to design and build a test bench to assay the actuation system of the third active knee-ankle-foot orthosis prototype that is being developed in the Department of Mechanical Engineering at ETSEIB. The objective is to know the dynamic performance of the actuation system in the two phases of human gait: when it has to work as a motor to swing the leg and when it has to work as a brake to lock knee rotation.

The project includes a study of the operating ranges of the orthosis actuation system and the selection of the sensors for the test bench. Besides the mechanical design of the parts that make up the test bench (seeking a design that could be used with other knee orthosis prototypes), and its construction, a review of other test benches used to study the mechanical performance of different knee orthoses is made.

Finally, this work also includes a preliminary design of the experiments intended to be done by means of the test bench, and a study of the environmental and social impact of the project.
1. STATE OF THE ART

This chapter describes the state of the art of knee active orthoses and of the tests that are carried out with them to evaluate their mechanical performance.

In section 1.1, there is a brief description of various types of orthoses with actuation at the knee joint, which are usually designed for spinal cord injured patients. All the devices are described from a mechanical point of view, without considering its control, which is a complex issue and deserves a deeper study. Section 1.2 summarizes some tests done by research groups to evaluate the performance of knee orthoses.

1.1. Knee orthoses

Orthoses are exoskeletal devices used to help patients with reduced mobility to increase their capacity of movement. Sistiaga [2012, pp. 26-30] gives an overview of these devices from its origin some decades ago to last years development.

The lower extremities orthoses designed for spinal-cord-injured patients can be classified in different groups: ankle-foot orthosis (AFO), knee-ankle-foot orthosis (KAFO), stance-control knee-ankle-foot orthosis (SCKAFO) and hip-knee-ankle-foot orthosis (HKAFO). The differences are related to the joints that the orthosis consists of.

Another criterion that differentiates orthoses is the kind of control and actuation they have. There are active orthoses (they have a power supply) and passive orthoses (they do not need power supply because they use the movement to store energy, using for instance springs).

During the last years, several orthoses that combine active and passive actuation systems have been developed. One example are the Series Elastic Actuator (SEA) orthoses, which usually consist in an actuator in series with a spring.

Some authors, as Fatone [2006, pp. 137-158] and Yakimovich et al. [2009, pp. 257-268], give a panoramic view of the research in lower extremity orthoses and the challenges for future design. Dollar and Herr [2008, pp. 144-158] also study which kinds of lower limb orthoses have been designed and studied during the last decades.
1.2. Test benches for knee orthoses

Most of the orthosis prototypes are tested on people (healthy or injured subjects) in order to analyse the performance of the device and the improvement of the gait. The gait of the subject with the prototype is captured and compared with the one of a healthy subject without orthosis.

However, some research groups test orthosis prototypes without subjects, in customized test benches, to know with precision the performance and the mechanical characteristics of the actuation system: the maximum moment they can exert with respect to the knee, the passive resistance, the number of cycles the mechanical components can bear, or the response time of the actuation system.

The possibilities and mechanisms to characterize the actuation systems of knee orthoses are diverse. Yakimovich et al. [2006, pp. 361-369] used a material testing machine to apply a controlled moment to a SCKAFO knee joint in order to determine the highest flexion moment that could be supported by the joint before failure, the amount of knee flexion permitted by the joint, and the amount and type of wear that occurs with repeated loading (see Fig. 1.1, left).

Spring et al. [2012, pp. 678-687] also used a tensile testing machine to determine the spring force of a KEA (Knee-Extension Assist) during spring compression and KEA extension moment during joint flexion and extension, and to verify the proper function of the locking mechanism (Fig. 1.1, right).
Chen’s group [2008, pp. 512-517] developed a knee brace which consists of a DC motor to give active torque together with a magnetorheological actuator that works as a clutch to transfer the torque when the DC motor is active, and as a brake when passive torque is needed. They used a testing structure similar to the ones described above to analyse the motion of the brace and the torque it can provide (Fig. 1.2), with the difference that the force was made by some weights (load disks).
Fig 1.2. Photograph of the structure used by Chen and Liao [2008, p. 514] to test a knee brace composed by a DC motor and a magnetorheological actuator.

Other research groups, which want to apply directly an external moment on the knee and study the actuation system performance, placed a rotatory actuator on the knee. This is the case of Bulea et al. [2012, pp. 822-832], who employed a robotic dynamometer to actuate the knee motion of the orthosis and to characterize the resistive torque and the ability to lock and unlock the knee under high loads of a Variable Impedance Knee Mechanism (VKIM) to provide controllable resistance to knee motion (Fig. 1.3).

This device substitutes the quadriceps to allow controlled levels of knee flexion during stance phase of gait and stair descent. It uses a magnetorheological fluid damper with a four-bar linkage transmission to provide controllable resistance to knee motion.
Finally, Dollar [2014, pp. 258-268] built a quasi-passive Compliant Stance Control Orthosis (CSCO), which implemented a linear spring in parallel with an impaired knee joint to compliantly support it during stance phase and then allowed the leg to freely swing to initiate the next step. He designed and built a mechanical knee simulator, consisting of a four-bar linkage actuated by a large three phase servomotor and servo controller (Fig. 1.4), in order to evaluate the moment-angle performance and its reliability, and to measure the response time of the Compliance Control Module (CCM).

This orthosis intends to provide the natural shock absorption function of the knee in order to reduce compensatory movements both in the affected and unaffected limbs, in opposition to other stance control orthoses which rigidly lock the knee during the stance phase.
The above examples show different mechanical solutions to test knee orthoses and measure their dynamic parameters. Some researchers use linear actuators (tensile testing machines or weights) to apply a torque on the knee. Others use rotatory actuators to directly exert a torque on the knee, and finally there are researchers that make a mechanical structure to simulate the performance of the leg and the knee during human gait.

The simplicity or complexity of the test bench design depends mainly on the magnitudes needed to be recorded and calculated and on how the orthosis works. Though, the crucial thing is to know with accuracy the external load applied on the orthosis, in order to characterize with precision the actuation system performance.
2. REQUIREMENTS OF THE TEST BENCH

2.1. Description of the actuation system prototype

This project is related with the research of the Biomechanical Engineering Group (BIOMEC) at ETSEIB, which is currently developing the design of the third stance control knee-ankle-foot orthosis (SCKAFO) for people with spinal cord injuries.

This device is compounded by a knee module coupled thorough some supports to a commercial AFO. That module actuates on the knee, substituting or complementing the muscles, and helps (together with the AFO) the subject to perform a more natural motion.

The design of this third knee actuation system prototype for the orthosis (Fig. 2.1, left) and the need of having an accurate characterization of it, led the group to the proposal of designing and building a test bench in order to know with precision the mechanical performance of this device.

Fig. 2.1. Left: CAD of the third knee actuation system for an active knee-ankle-foot orthosis developed by the Biomechanical Engineering Group (BIOMEC) of ETSEIB. Right: CAD of the AFO and the supports where the knee actuation system is coupled.
This knee actuation system has the following main elements (see Fig. 2.2):

(a) An electric motor and a position control unit (it does not appear in Fig. 2.2).
(b) A ball screw actuated by the motor, with a nut that moves along it. The rotation of the ball screw produces a linear movement of the nut.
(c) A rod which transfers the movement from the nut to the shank support.
(d) Two couplings to link with the supports that are tied to subject’s thigh and shank.

Fig. 2.2. Principal elements of the actuation system: (a) electric motor; (b) nut and ball screw; (c) rod; and (d) couplings with the supports that are tied to subject’s thigh and shank.

2.2. Objectives of the test bench

The principal objectives of studying the dynamic performance of the actuation system are three:
a) Know how much power is dissipated in the ball screw transmission when the actuation system has a forward performance and the electrical motor works as a motor.

b) Know how much power is dissipated in the ball screw transmission when the actuation system has an inverse performance and the electrical motor works as a brake.

c) Know the performance of the actuation system if it is used a rod made up of an elastic material (instead of steel) in order to store energy during some phases of human gait and provide it during others.

In order to find out the dissipated power and characterize the actuation system dynamic performance, the sensors needed for the test bench are very important. The bench must be provided with the appropriate sensors to measure accurately the desired magnitudes.

However, before selecting the sensors and beginning to design the test bench, some hypotheses and starting points have been settled:

a) The movement of the leg during human gait and the external forces exerted on it are considered to be in the sagittal plane.

b) The human gait and the forces captured in the Laboratory of Biomechanics at ETSEIB [Sistiaga, 2012; Arroyo, 2011] and taken as reference to design the test bench are the ones of a healthy subject walking on a flat surface.

c) The main objective of the test bench is not to simulate the exact kinematics and dynamics of the leg during human gait, but to characterize the actuation system of an active knee-ankle-foot orthosis through a well-measured motion (whose magnitudes should have similar ranges to the ones during human gait).

2.3. Test bench measured magnitudes

2.3.1. Measured magnitudes

The electric motor of the actuation system (Annex A.1), Fig. 2.3, is equipped with an encoder (Annex A.2), Fig. 2.3, and it is controlled with a position control device (Annex A.3), Fig. 2.3. Consequently, angular position, angular velocity and angular acceleration of the shaft can be easily recorded.
Another magnitude of the motor that should be measured is the electric current consumption, which is directly related with the torque provided by the motor (this relationship is given by the manufacturer [Maxon Motor, 2012, p. 197]). With these two magnitudes, the mechanical power developed by the actuator is perfectly known.

Regarding the test bench, the following measurements are needed to determine the mechanical energy variation rate: angular position, velocity and acceleration of the part of the test bench which acts as leg.

In order to know the power developed by the external wrench on test bench foot (which represent the foot-ground contact forces), the value of the components of the wrench exerted at the foot and the angular velocity of test bench leg are needed.

Once the power developed by the motor, the mechanical energy variation rate and the power developed by external wrench are known, the power dissipated in the ball screw transmission can be calculated either in forward or inverse performance.

**2.3.2. Operating ranges of measured magnitudes**

In order to choose the most suitable angular sensor for the test bench it is essential to look at the operating ranges (in terms of angle position, angular velocity, angular acceleration) of the knee joint during human gait.
Fig. 2.4 shows the evolution knee angle (considering “knee angle” the angle between the shank and the thigh), angular velocity and angular acceleration during a cycle of human gait. The measurements have been taken from a capture at the Laboratory of Biomechanics of the Department of Mechanical Engineering of ETSEIB [Sistiaga, 2012; Arroyo, 2011].

Fig. 2.4. Evolution of the angle (of the shank with respect to the thigh), the angular velocity and the angular acceleration of the knee during a cycle of human gait. The angle is considered 0° when the shank is aligned with the thigh. The first 60% of the cycle corresponds to stance phase, and the last 40% to swing phase. [Sistiaga, 2012; Arroyo, 2011].

From Fig. 2.4 it can be observed that the maximum and minimum values of those magnitudes are:
<table>
<thead>
<tr>
<th>Angle</th>
<th>Angular velocity</th>
<th>Angular acceleration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Units</td>
<td>[rad]</td>
<td>[deg]</td>
</tr>
<tr>
<td>Maximum</td>
<td>0,98</td>
<td>56,4</td>
</tr>
<tr>
<td>Minimum</td>
<td>0,12</td>
<td>6,9</td>
</tr>
</tbody>
</table>

Table 2.1. Maximum and minimum values of the angle (of the shank with respect to the thigh), the angular velocity and the angular acceleration of the knee during human gait [Sistiaga, 2012; Arroyo, 2011]. The angle is considered 0° when the shank is aligned with the thigh. Positive sign corresponds to knee flexion.

Those ranges are the starting point to set an approximate rotation speed for the part of the test bench that acts as a human leg.

In order to select the appropriate sensor to measure the external wrench applied at the foot of the test bench, it is necessary to know the range of forces and torque at the knee during human gait. Fig. 2.5 shows the evolution of the forces and the torque on the knee during a cycle of the same human gait capture [Sistiaga, 2012; Arroyo, 2011]. Table 2.2 summarizes the maximum and minimum values.

<table>
<thead>
<tr>
<th>Horizontal force</th>
<th>Vertical force</th>
<th>Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>Units</td>
<td>[N]</td>
<td>[N]</td>
</tr>
<tr>
<td>Maximum</td>
<td>60,97</td>
<td>57,44</td>
</tr>
<tr>
<td>Minimum</td>
<td>-89,18</td>
<td>-752,50</td>
</tr>
</tbody>
</table>

Table 2.2. Maximum and minimum values of forces (horizontal and vertical with respect to the ground) and torque at the knee (considering it part of the shank) during human gait [Sistiaga, 2012; Arroyo, 2011]. Positive sign of force corresponds horizontally to the direction of the gait and vertically upward; positive sign of torque corresponds to knee flexion.
The most important magnitude is the torque at the knee, which has its maximum value (in absolute value) during the stance phase. During that phase, the foot-ground contact forces create a moment with respect to the knee, and it will be imitated with an external force applied at test bench foot. This torque is balanced by the orthosis actuation system.

Considering shank’s length to be 0.45 m, the external force (perpendicular to the shank) needed in the experiments at the foot of the test bench to create a 38 Nm torque should be 85 N. So, during the tests, that will be the force to be applied along the direction perpendicular to the leg. As it is unknown how that load will be exerted, the bench should...
have force sensors to measure all the load components that could appear (two for the force, and one for the torque).

2.4. Sensor alternatives

2.4.1. Angular sensor

Attending to these operating ranges, different types of sensors to measure the needed magnitudes have been considered. Table 2.3 shows the angular sensors considered.

<table>
<thead>
<tr>
<th>Photograph</th>
<th>Type</th>
<th>Characteristics</th>
<th>Model/Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angular potentiometer</td>
<td>Ranges: 90°, 180°, 345°, 1800° and 3500°&lt;br&gt;Linearity: 0.1% and 0.05% optional&lt;br&gt;Output: amplified 4-20mA and 0-10V</td>
<td>ASM AWS1, ASM AWS2&lt;br&gt;www.asm-sensor.com</td>
<td></td>
</tr>
<tr>
<td>Optical Incremental Encoder</td>
<td>Resolution: 12500 pulses/revolution&lt;br&gt;Maximum frequency: 150kHz&lt;br&gt;Output: two square waves 90° ±15° push-pull</td>
<td>ELAP E30&lt;br&gt;www.elap.it</td>
<td></td>
</tr>
</tbody>
</table>
Table 2.3. Different kinds of angular sensors

<table>
<thead>
<tr>
<th>Absolute encoder</th>
<th>Range: singleturn, 90°</th>
<th>Linearity error: 0,07% of active angle</th>
<th>Hohner HS-10</th>
<th><a href="http://www.hohner.es">www.hohner.es</a></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Resolution: 12 bits, 4096 positions for 360°</td>
<td>Output: 0-10V and 0-20mA or 4-20mA</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

All the previous sensors have enough quality for the application at hand because of two reasons: first, the rotation velocity is not too high or demanding, and second, the performance will not be continuous, as the test bench will be used to make punctual experiments. It has been chosen the Hohner HS10 angular sensor because it was available at the Department of Mechanical Engineering of ETSEIB, and it fitted adequately the specifications for the test bench. Hohner is also a well-known brand of sensors.

2.4.2. Force and torque sensors

Table 2.4 shows some sensors for measuring forces and torques. Two main groups of sensors have been considered: sensors that measure only force in one direction and sensors that could measure forces and torques in several directions.

<table>
<thead>
<tr>
<th>Photograph</th>
<th>Type</th>
<th>Characteristics</th>
<th>Reference/Model</th>
<th>Approx. price</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load cell</td>
<td>Range: 2.5kg to 200kg in F1 model and to 2000kg in F3</td>
<td>Combined error: 0,026%</td>
<td>AEP F1, AEP F3</td>
<td>100 €</td>
</tr>
<tr>
<td></td>
<td>Output: 2mV/V</td>
<td><a href="http://www.aeptransducers.com">www.aeptransducers.com</a></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
| **Load cell** | Range: 3, 5, 10, 20, 50, 100 kg  
Number of divisions: 6000 ... 15000  
Output: 2mV/V | Siemens WL260 SP-S AA  
www.automation.siemens.com | 100 € |
|---|---|---|---|
| **Load cell** | Range: 5 to 50 kg  
Combined error: ±0,017%  
Accuracy: 3000 divisions  
Output: 2mV/V | Utilcell 102  
www.utilcell.es | 50 € |
| **Force and torque sensor (multiaxial)** | Force range: 50, 125, 250 lb. (222, 556 and 1112N).  
Torque range: 100, 250 and 500 inchlb (11, 28 and 56 Nm)  
Linearity: ±0,2% f.s.  
Small size and low weight | AMTI FS6-100  
www.amti.biz | 6975 € |
| **Force and torque sensor (multiaxial)** | Force range: 50, 125, 250 and 500 lb. (222N, 556N, 1112N and 2224N)  
Torque range: 100, 250, 500 and 1000 inchlb (11Nm, 28Nm, 56Nm y 113 Nm)  
Linearity: ±0,2% | AMTI MC3A-100  
www.amti.biz | 4900 € |
| **Tri-axial load cell** | Measurement range: ±400 lb to ±50000 lb (±1.8 kN to ±222 kN)  
Linearity: 1 % f.s.  
Output: 1,5mV/V | Althen 70048  
www.althensensors.com | 5980 € |

**Table 2.4.** Different kinds of force and torque sensors

The force sensors type selected are load cells because of their simplicity, their availability, and their low price for a reasonable accuracy. The fact that the range of forces hold by the test bench is relatively low and that there are not size restrictions, favours the choice.
Three load cells adequately arranged will be needed to characterize the external wrench applied at the foot, which includes the two forces and the torque.

The selected load cell is the Utilcell model 102. It is a double bending beam load cell, usually used for platforms, and with high accuracy for off-centre loads. It is one of the cheapest load cells, it has enough accuracy and capacity, and the headquarters of the manufacturer are in Barcelona.
3. TEST BENCH MECHANICAL DESIGN

The specific models of sensors chosen for the test bench are:

a) Hohner absolute encoder HS10-31312117-1024 (see datasheet in Annex A.4).
   The Hohner encoder permits a maximum angular velocity of 6000 min\(^{-1}\), which is
   more than enough for this application, that has a maximum angular speed of 3.80
   rad/s = 36.3 min\(^{-1}\). It has a resolution up to 12 bits, 4096 positions, for 360\(^{\circ}\) (0.088
   \(^{\circ}\)/position) and an updating frequency of 100 kHz. So, it fits perfectly with test bench
   requirements.

b) Utilcell Load Cell 102; two with a nominal capacity of 100 N and one with a nominal
   capacity of 200 N (see datasheet in Annex A.5).
   The one of 200 N is arranged along shank’s direction, and it will measure the
   external force perpendicular to this direction. The other two load cells, with a
   nominal capacity of 100 N), are arranged perpendicularly to shank’s direction, in
   order to measure the force in the direction of the leg and the torque.
   In this way, according to nominal specifications, the test bench could measure
   forces with the same value in the two directions and a nominal torque that depends
   on the definitive geometrical arrangement of those two load cells. The safety factor
   of the load cells is calculated in section 3.2.

Once selected the sensors taking into account the ranges of performance of the actuation
system and the test bench, the next step is to design the different parts that will make it
up.

3.1. Considerations

The test bench must be, on the one hand, adequate to achieve the objectives of this project
and, on the other hand, with the simplest possible manufacturing process. First of all, there
should be taken into account some characteristics of the test bench that have a significant
influence on the design:
a) The mass and the inertia of the parts of the test bench that rotate (shank and foot) should be similar to the ones of a human leg in order to have operating ranges comparable to the ones for the leg during human gait.
b) There are not constraints of space and volume and the dimensions of the test bench do not need to be optimized.
c) Due to the former reason, the test bench could be designed with wide security factors in terms of parts strength.
d) The forces are not high, so the materials used for the bench can be common ones, and they do not need to be specific high-performance materials.
e) If possible, the test bench should be easily adaptable to other knee actuation system prototypes.

And some essential parts or elements of the test bench, independently of the design, are the following:

a) An element of union with the structure (external fixed reference).
b) A joint which acts as knee (made up of bearings).
c) A coupling with the supports of the active orthosis actuation system to be tested;
d) A coupling with the angular sensor.
e) A coupling with the load cells (which should have an adequate geometric arrangement to measure properly the wrench).
f) An element where the external wrench can be easily applied.

Taking into account the former considerations, the test bench has been designed. Fig. 3.1 is a drawing of the test bench definitive design and Fig. 3.2 shows the CAD design of the test bench linked with the active orthosis actuation system intended to be tested.
Fig. 3.1. Drawing of the test bench. The most notable parts are indicated in blue. Its similarity with a human leg is also remarked.

Fig. 3.2. CAD of the knee actuation system coupled with the test bench. The sensors of the test bench are marked.
The assembly drawing and the bill of materials are shown in Annex D.1. In the bill of materials it is indicated which parts are standard and should be bought and which ones should be custom manufactured according to the drawings contained in Annex D.2.

To design the test bench and make it similar to a human leg (in terms of mass and moment of inertia), different works on human anthropometric parameters have been taken into account [De Leva, 1996, pp. 1223-1230; Dumas, 2007, pp. 543-553; and Winter, 2005, pp. 59-74]. Usually, it is defined the length, weight and centre of mass position for each body segment based on subject’s height and weight. Annex B includes a comparison between the values from those authors and the ones of the test bench final design.

Table 3.1 shows the values of mass and inertia of the definitive design, taking into account the geometry and the materials used. They have been calculated with the CAD program Solidworks. After building and assembling the test bench, some preliminary tests must be done in order to obtain the real values of mass, position of the centre of mass, and moment of inertia.

<table>
<thead>
<tr>
<th>CAD model</th>
<th>Length</th>
<th>Mass</th>
<th>Centre of mass</th>
<th>Inertia</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$L$</td>
<td>$M$</td>
<td>$L_{\text{CoM}}$</td>
<td>$I_0$</td>
</tr>
<tr>
<td>Units</td>
<td>[mm]</td>
<td>[kg]</td>
<td>[mm]</td>
<td>[kgmm$^2$]</td>
</tr>
<tr>
<td>Shank</td>
<td>428.0</td>
<td>4,353</td>
<td>195.2</td>
<td>54256</td>
</tr>
<tr>
<td>Foot</td>
<td>302.0</td>
<td>1,400</td>
<td>1.0</td>
<td>12779</td>
</tr>
<tr>
<td>Shank + foot</td>
<td>-</td>
<td>5,775</td>
<td>324.2</td>
<td>211090</td>
</tr>
</tbody>
</table>

**Table 3.1.** Length, mass an inertia of the elements of the test bench that act as shank and foot. The foot is perpendicular to the shank. $L_{\text{CoM}}$ is the distance between the centre of mass and the proximal (nearest to the trunk) joint. $I_0$ is the moment of inertia with respect to the centre of mass of the segment, and $I_S$ with respect to the knee.

Regarding the design of the bench, it is appropriate to make the following considerations:

a) The joint has been designed with two ball bearings to give it more robustness.

b) All the couplings of the different parts have been made with bolts, so that the test bench can be easily assembled and disassembled.
c) This is also a good solution to adapt it to other actuation system prototypes: only two parts should be redesigned (proximal separator, see Annex D.2, drawing 03; and distal separator, see Annex D.2, drawing 06).

d) The test bench will be fixed to a structure through four bolts placed in the bottom of the tube that contains the bearings and the shaft (see Annex D.2, Drawing 01).

3.2. Strength calculations

An important section of the design of the test bench are the calculations to check its mechanical resistance under the expected loads and the resistance of all the parts and couplings that compose it.

For the strength study, the worst situation that will take place during the assays has been considered in order to determine the loads that will appear. This allows to know the security factor of the test bench. The configuration considered is: the leg of the bench is held in horizontal position with the maximum external load exerted at the foot, which is balanced by the active orthosis actuation system. The involved loads are:

(a) The self-weight of the parts of the test bench that rotate around the joint (the centre of mass is point G in Fig. 3.3).

(b) The external wrench applied at test bench foot (point P in Fig. 3.3.). It has been considered that it consists only of a force that is perpendicular to the shank and creates a moment of 37.85 Nm about the knee (see Table 2.2). As the distance between the knee and the point of application of the force is 0.49 m, the force will be 78 N, which can be rounded to 80 N. The torque applied is considered to be zero.

(c) The force exerted by the actuation system through the rod of the active orthosis on the test bench (point Q in Fig. 3.3), which has the direction of the rod.

(d) The forces of contact between the shaft and the two rolling bearings of the test bench (point S in Fig. 3.3).
**Fig. 3.3.** Points of application of the loads on the test bench: point S, forces of contact between the shaft and the rolling bearings; point Q, forces made by the actuation system of the active orthosis; point G, centre of mass; point P, external wrench.

The total mass of the parts of the test bench that rotate will have an approximately value: $M \approx 6.2\, \text{kg}$. All parts are made of steel except the shaft, which will be made of aluminium.

The maximum force exerted at point P during the experiments will be of 80 N, in the direction perpendicular to the leg. The torque applied at point P will be zero, because the external load intended to be applied on that point is a pure force (even though, the torque should be measured).

The two components of the force at point Q will be performed by the actuation system of the active orthosis. During human gait, this force is the one that accelerates and moves the leg, substituting or complementing the work of the muscles.

Finally, the two components of the force at the joint, point S, are the reactions with the rolling bearings.

For loads (a) and (b) some rough values have been already taken. And, for loads (c) and (d), it has been considered the static equilibrium of the leg in the horizontal position, and the values of these loads calculated as a function of the known loads (a) and (b). Applying
the equilibrium equations of a rigid body it is obtained the system of equations shown in Eq. (3.1).

\[
\begin{align*}
F_S^x - F_Q^x &= 0 \\
F_S^y - F_Q^y &= M \cdot g + F \\
x_G \cdot M \cdot g + x_P \cdot F &= -y_Q \cdot F_Q^x - x_Q \cdot F_Q^y
\end{align*}
\]

(3.1)

where \(x_G = 0.255\) m is the horizontal distance between points G and S, \(x_P = 0.490\) m is the horizontal distance between points P and S, \(x_Q = 0.052\) m is the horizontal distance between points Q and S, and \(y_Q = 0.046\) is the vertical distance between points Q and S.

It is also known that force \(F_Q\) has the direction of the rod, so the two components are related as shown in Eq. (3.2).

\[
\frac{F_Q^y}{F_Q} = \tan \varphi
\]

(3.2)

The angle \(\varphi\) is the one between the rod and the horizontal axis. For the position considered it is \(\varphi_0 = 2.59^\circ\).

With Eqs. (3.1) and (3.2), all the unknown values are calculated (the complete process is in Annex C.1) and shown in Eq. (3.3):

\[
\begin{align*}
F_Q^x &= 1253 \text{ N} \\
F_Q^y &= 57 \text{ N} \\
F_S^x &= -1253 \text{ N} \\
F_S^y &= 198 \text{ N}
\end{align*}
\]

(3.3)

It is also useful to calculate the module of those two loads (Eqs. (3.4) and (3.5)):

\[
F_Q = 1254 \text{ N}
\]

(3.4)

\[
F_S = 1267 \text{ N}
\]

(3.5)
Now that there is an approximate value for each of the loads, the strength and the safety factors of some parts of the test bench must be checked. The load cell arranged in shank’s direction, which measures the force perpendicular to it, has a load nominal capacity of 200 N and a safe load limit of 200% (400 N). So, the safety limit is:

\[ S_{f, \text{load cell}} = \frac{392 \text{ N}}{80 \text{ N}} = 4.90 \]  

The mechanical resistance of the ball rolling bearings and the bolted couplings are checked in sections 3.2.1 and 3.2.2, respectively.

Regarding the resistance of materials, as it has been said before, all the parts that make up the test bench are of steel, except the shaft and the tube containing the bearings, which are of aluminium. Taking into account the material, the sections of the parts and the relatively low loads that will bear, the tensions that will appear during test bench performance will be quite lower than the elastic limit of those two materials.

### 3.2.1. Bearings loads and life

Both dynamic and static bearing load conditions should be independently verified. However, as the performance of the bench has a low maximum speed for the experiments (36 min\(^{-1}\), see Table 2.1) and the swing range is of about 90° (it does not perform full revolutions), the most relevant verification is the static bearing load one. In all checking calculations for the bearings, the catalogue of one well-known manufacturer has been used [SKF, 2012, pp. 63-92]. The selected bearings for the test bench correspond to the model SKF 6002 (see annex A.6).

First of all, it must be studied how the load is distributed between the two rolling bearings. It is considered that the maximum load at the end of the shaft is \( F_S = 1267 \text{ N} \), in absolute value. Fig. 3.4 shows the loads applied on the shaft:
After solving the equations (see Annex C.2), the loads at the bearings are:

Bearing A: \[ F_A = 1868 \, \text{N} \] \hspace{1cm} (3.7)

Bearing B: \[ F_B = 601 \, \text{N} \] \hspace{1cm} (3.8)

So the most loaded rolling bearing is A, and it is the one which will be checked. It should be noted that the loads are only radial, because the movement of the leg is supposed to be performed in the sagittal plane and the axial forces that could appear (out of that plane) are negligible.

The basic static load rating \( C_0 \) is used under the following conditions:

- Very slow rotational speeds \( (n < 10 \, \text{min}^{-1}) \).
- Very slow oscillating movements.
- Stationary bearings under load for extended periods.

In the present situation, the first condition is not fulfilled, because the rotational speed could arrive to 36 \( \text{min}^{-1} \) in the experimental tests (see Table 2.1). However, it will normally run at lower rotatory speeds under oscillatory conditions.
The load done by the actuation system and the external load will be present only during the tests (and not in all of them), and the load of the weight of the bench will always be exerted on the bearings.

Due to the former reasons, it is reasonable to say that this is a case of static bearing load. The verification is performed by checking the static safety factor of the application, which is defined as:

\[ s_0 = \frac{C_0}{P_0} \]  

(3.9)

Where \( s_0 \) is the safety factor, \( C_0 = 2.85 \text{ kN} \) is the basic static load rating and \( P_0 = 1.87 \text{ kN} \) is the equivalent static bearing load. The safety factor is then 1.52. This coefficient is above the minimum value for a ball bearing with normal performance requirements (\( s_0 = 1 \) [SKF, 2012, p. 89]).

Moreover, although the dynamic bearing load checking is less indicative for this situation, it can be done and see which are the conclusions. In terms of dynamic load, the life of a rolling bearing is expressed as the number of revolutions or the number of operating hours at a given speed that the bearing is capable of enduring, before the first sign of metal fatigue occurs on a raceway of the inner or outer ring or on a rolling element.

Now, the life for the bearings can be calculated considering the dynamic load, which has been supposed always the maximum (the same as the static load). The basic rating life of a bearing in accordance with ISO 281 is:

\[ L_{10} = \left( \frac{C}{P} \right)^p \]  

(3.10)

where \( L_{10} \) is the basic rating life (at 90% reliability) [million revolutions], \( C = 5.85 \text{ kN} \) the basic dynamic load rating, \( P = 1.87 \text{ kN} \) the equivalent dynamic bearing load, and \( p = 3 \) the exponent for ball bearings. The rating life results 30,616 million of revolutions, which, considering an oscillation of \( \gamma = 90^\circ \), corresponds to 61,231 million oscillation cycles [SKF, 2012, p. 70].

This life is more than enough for the use that will be given to the bench, especially taking into account that the use will not be continuous and the external load exerted at the foot will be lower than the one considered here.
3.2.2. Bolt couplings

All the parts of the test bench are assembled by bolts to make it more flexible to be used with other knee actuation systems. For that reason, bolted couplings are a feature of the design that must be carefully analysed. The whole checking process has been done following Fenollosa [2009, pp. 41, 75, 81-84].

The most loaded bolt coupling is the one that joins the actuation system to the test bench (detail in Fig. 3.5), and it is the one that has bolts of the lowest metric (M5) because of the actuation system design. The parts joined are the distal support of the actuation system and the distal separator of test bench (see Annex D.2, drawing 06).

![Fig. 3.5. Front and back details of the coupling between the actuation system and the test bench by two bolts and two nuts (yellow circles).](image)

The peak load considered to be transmitted through this coupling is the one calculated in Eq. (3.4): $F_Q = 1254 \text{ N}$. In this way, the two bolts are working under a transverse load condition (see Fig. 3.6).
The union resists the transverse force by the friction between the two parts thanks to the clamping force exerted by bolts. In Annex C.3 it is described the checking process of bolt couplings with transverse forces.

It is obtained that the safety factor for this bolted coupling is:

\[
S_f = \frac{3 \cdot n \cdot m \cdot \mu \cdot F^\prime_{C_{\min}}}{4 \cdot F_Q}
\]  
(3.11)

where \( n = 2 \) is the number of bolts, \( m = 1 \) is the number of contact surfaces, \( \mu = 0.15 \) is the coefficient of friction for steel surfaces, \( F^\prime_{C_{\min}} = F_{C_{8.8 \text{lim}}} = 6350 \text{ N} \) is the clamping force (considering a bolt of class 8.8 and metric M5, clamped without lubrication [Fenollosa, 2009, p.75]), and \( F_Q = 1254 \text{ N} \) is the transverse force for this application. So, the safety factor is:

\[
S_{f\text{ bolt coupling}} = 1.14
\]  
(3.12)

If used bolts are of class 10.9, the clamping force can be higher \( (F^\prime_{C_{\min}} = F_{C_{10.9 \text{lim}}} = 8950 \text{ N}) \) and the safety factor results:

\[
S_{f\text{ bolt coupling}} = 1.61
\]  
(3.13)
It is usual to consider a minimum safety factor $S_{f_{min}} = 1.25$ for machines and general applications [Fenollosa, 2009, p. 83]. So, for that bolted union, it is recommended to use bolts of class 10.9.

The other coupling between the actuation system and the test bench is exactly the same: it has to bear a similar load and has also two bolts of metric M5. So, there, bolts of class 10.9 should also be used. And the coupling between distal separator and distal support (see Annex D.1, Assembly drawing), both of them parts of the test bench, that has to bear more or less the same load has also two bolts, but of metric M6, so it is stronger and will have a higher safety factor.

To sum up, the safety factors are: for the load cells 4.90; for the bearings 1.52; and for the most critical bolted coupling 1.61. In that way, the bearings and the bolted coupling are the elements with the lowest safety factors, but they are high enough to guarantee a safe performance of the test bench with the actuation system.
4. CHARACTERIZATION OF THE TEST BENCH

The use of the test bench has several objectives, for instance: know the efficiency in forward performance of the actuation system (forward efficiency), and know the efficiency in inverse performance of actuation system (inverse efficiency). These are very important to characterize the autonomy of the active orthosis during the real operation.

To achieve the test bench objectives, there have been selected two types of sensors to measure the adequate magnitudes of the test bench (see Chapter 2), besides the encoder included in the active orthosis actuation system. There are two angular encoders (including the one of the electrical motor): one that measures the angle of motor’s shaft and other that measures the angle of the leg. That way, experiments using an elastic rod (which is the part that transfers the movement from the actuation system to the leg) can be done to see how can it stores and gives energy. That last kind of experiments will not be considered here.

In order to perform the experiments, it is necessary to know which magnitudes will be measured and the relationship between those quantified magnitudes and the ones needed to calculate forward and inverse efficiency through the calculation of mechanical power and mechanical energy variation rate.

The magnitudes needed to find out the efficiency are:

<table>
<thead>
<tr>
<th>Magnitude [units]</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Actuation system</strong></td>
<td></td>
</tr>
<tr>
<td>Rotation speed [rad/s]</td>
<td>Rotation speed and torque of the electrical motor allow to calculate the mechanical power developed by the motor of the actuation system</td>
</tr>
<tr>
<td>Torque [N·m]</td>
<td></td>
</tr>
<tr>
<td><strong>Test bench leg</strong></td>
<td></td>
</tr>
<tr>
<td>Rotation speed [rad/s]</td>
<td>The rotation speed of the leg and external wrench allow to find the mechanical power performed by this external wrench and the variation rate of kinetic energy of the test bench</td>
</tr>
<tr>
<td>External wrench [N, N·m]</td>
<td></td>
</tr>
<tr>
<td>Angle [rad]</td>
<td>Leg angle permits to calculate the variation rate of potential energy due to gravity</td>
</tr>
</tbody>
</table>

*Table 4.1. Magnitudes necessaries to calculate the efficiencies of the actuation system.*
Table 4.2 reflects, for each measured magnitude, which one can be calculated in order to characterize the efficiency of the actuation system:

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Magnitude measured [units]</th>
<th>Magnitude wanted [units]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ammeter</td>
<td>Current consumed by the electric motor [A]</td>
<td>Torque of the motor [Nm]</td>
</tr>
<tr>
<td>Motor encoder</td>
<td>Angle rotated by the motor [rad]</td>
<td>Angular velocity of the motor [rad/s]</td>
</tr>
<tr>
<td>Load cell 1</td>
<td>Force on load cell 1 [N]</td>
<td>Wrench at test bench foot [N] [Nm]</td>
</tr>
<tr>
<td>Load cell 2</td>
<td>Force on load cell 2 [N]</td>
<td></td>
</tr>
<tr>
<td>Load cell 3</td>
<td>Force on load cell 3 [N]</td>
<td></td>
</tr>
<tr>
<td>Analogue encoder</td>
<td>Leg angle [rad]</td>
<td>Angle of leg [rad]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Angular velocity of leg [rad/s]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Angular acceleration of leg [rad/s²]</td>
</tr>
</tbody>
</table>

Table 4.2. Magnitude measured by each sensor and which magnitude allows to find.

The relationship between the measured and the wanted magnitude in all cases exposed in Table 4.2, is very simple except for the load cells. The electric current consumed by the motor is directly related with its torque (see Annex A.1, motor’s catalogue). The encoder of the motor gives an electric signal (current or voltage) proportional to the angle rotated by its shaft. And the analogue encoder of the leg also gives an electric signal proportional to the angle between the proximal and distal parts. To get leg angular velocity and acceleration it is necessary to derivate that angle once and twice with respect to time.

For loads cells, it is a bit more complex to get the force and the torque applied on the test bench foot from the signals measured by the three devices. First, the electric signal (voltage) of each load cell is converted easily to force using the sensor sensitivity. But then, it has to be studied how to get the wrench (two components of the external force and the torque) applied at point P (Fig. 4.1) according to the geometrical arrangement of the load cells.
The forces measured by the load cells are:

\[ F_{lc1}, F_{lc2}, F_{lc3} \]  \hspace{1cm} (4.1)

The components of the wrench at point \( P \), according to the reference system used, are defined as:

\[ F^x, F^y, T \]  \hspace{1cm} (4.2)

The wrench at point 1 (Fig. 4.3, point of the load cell 1) is:

\[ F_1^x, F_1^y, T_1 \]  \hspace{1cm} (4.3)

The wrench at point 2 (Fig. 4.4, point of the load cell 2) is:

\[ F_2^x, 0, T_2 \]  \hspace{1cm} (4.4)

The wrench at point 3 (Fig. 4.5, point of the load cell 3) is:

\[ F_3^x, F_3^y, T_3 \]  \hspace{1cm} (4.5)

The wrench at point 4 (Fig. 4.2, point of upper ear) is:

\[ F_4^x, F_4^y, 0 \]  \hspace{1cm} (4.6)
The foot support, in the lower joint, has a sliding pin in vertical direction (see Annex D.2, Drawing 12). So, the wrench at point 5 (Fig. 4.2, point of lower ear) is:

\[ F_5^x, 0, 0 \] (4.7)

Considering the equilibrium of foot support (see Fig. 4.2), the following system of equations is obtained:

\[
\begin{align*}
F_x^x - F_4^x - F_5^x &= 0 \\
F_4^y - F_y^y &= 0 \\
F_4^x \cdot \frac{L_e}{2} - F_5^x \cdot \frac{L_e}{2} + T &= 0
\end{align*}
\] (4.8)

where \(L_e\) is the distance (in the \(y\) axis) between the two joints.

Fig. 4.2. Free diagram of the foot support.

Considering the equilibrium of upper ear (see Fig. 4.3), the following system of equations is obtained:
where $L_{ear}^y$ is the distance in the $y$ axis between points 1 and 4, and $L_{ear}^x$ is the distance in the $x$ axis between points 1 and 4.

\[ \begin{align*}
    F_4^x - F_1^x &= 0 \\
    F_1^y - F_4^y &= 0 \\
    T_1 - F_4^x \cdot L_{ear}^y - F_4^y \cdot L_{ear}^x &= 0
\end{align*} \] (4.9)

Considering the equilibrium of lower ear (Fig. 4.4), the equations are the following:

\[ \begin{align*}
    F_5^x - F_2^x &= 0 \\
    T_2 + F_5^x \cdot L_{ear}^y &= 0
\end{align*} \] (4.10)

where $L_{ear}^y$ is the distance in the $y$ axis between points 2 and 5, and $L_{ear}^x$ is the distance in the $x$ axis between points 2 and 5.

Considering the equilibrium of the rigid body formed by foot support, ears, load cells 1 and 2, and foot coupling (Fig. 4.5), the equations obtained are:

\[ \begin{align*}
    F_3^x - F_3^x &= 0 \\
    F_3^y - F_3^y &= 0 \\
    T_3 - F_3^x \cdot L_3^y - F_3^y \cdot L_3^x - T &= 0
\end{align*} \] (4.11)
where \( L_3^y \) is the distance in the \( y \) axis between points P and 3, and \( L_3^x \) is the distance in the \( x \) axis between points P and 3.

![Free diagram of the system composed by: foot support, upper and lower ears, load cells 1 and 2, and foot coupling.](image)

**Fig. 4.5.** Free diagram of the system composed by: foot support, upper and lower ears, load cells 1 and 2, and foot coupling.

It is known that: \( F_1^x = -F_{lc1} \), \( F_2^x = -F_{lc2} \), and \( F_3^y = -F_{lc3} \). So, from the previous equilibrium equations, the external force and torque at the foot (point P) are determined:

\[
F^x = -F_{lc1} - F_{lc2} \quad (4.12)
\]

\[
F^y = -F_{lc3} \quad (4.13)
\]

\[
T = F_{lc1} \cdot \frac{t_e}{2} - F_{lc2} \cdot \frac{t_e}{2} \quad (4.14)
\]

Writing it in matrix notation, to go from the values measured by load cells to the values of the external wrench is easier:
The wrenches at the studied points can also be obtained:

\[ F_1^x = -F_{lc1} \]  
\[ F_1^y = -F_{lc3} \]  
\[ T_1 = -F_{lc3} \cdot L_{e_1}^x - F_{lc1} \cdot L_{e_1}^y \]  
\[ F_2^x = -F_{lc2} \]  
\[ F_2^y = 0 \]  
\[ T_2 = F_{lc2} \cdot L_{e_1}^y \]  
\[ F_3^x = -F_{lc1} - F_{lc2} \]  
\[ F_3^y = -F_{lc3} \]  
\[ T_3 = F_{lc1} \cdot \frac{L_e}{2} - F_{lc2} \cdot \frac{L_e}{2} - F_{lc3} \cdot L_3^y - F_{lc1} \cdot L_3^x - F_{lc2} \cdot L_3^y \]

Writing them in matrix notation:

\[
\begin{bmatrix}
F_1^x \\
F_1^y \\
T_1
\end{bmatrix} = \begin{pmatrix}
-1 & 0 & 0 \\
0 & 0 & -1 \\
\frac{L_e}{2} & -\frac{L_e}{2} & 0
\end{pmatrix} \begin{bmatrix}
F_{lc1} \\
F_{lc2} \\
F_{lc3}
\end{bmatrix}
\]  
\[ (4.15) \]

\[
\begin{bmatrix}
F_2^x \\
F_2^y \\
T_2
\end{bmatrix} = \begin{pmatrix}
0 & -1 & 0 \\
0 & 0 & 0 \\
L_{e_1}^y & L_{e_1}^x & 0
\end{pmatrix} \begin{bmatrix}
F_{lc1} \\
F_{lc2} \\
F_{lc3}
\end{bmatrix}
\]  
\[ (4.26) \]
\[
\begin{bmatrix}
F_3^x \\
F_3^y \\
T_3
\end{bmatrix} = \begin{pmatrix}
-1 & -1 & 0 \\
0 & 0 & -1 \\
\frac{l_e}{2} - l_3^y & -\frac{l_e}{2} - l_3^y & -l_x
\end{pmatrix}\begin{bmatrix}
F_{lc1} \\
F_{lc2} \\
F_{lc3}
\end{bmatrix}
\] (4.27)

In order to collect and record the electric signals, all test bench sensors will be connected to the universal data acquisition instrument Dewetron DEWE-43 (Fig. 4.2); see datasheet in Annex A.7. This device can be connected by USB to a computer, and includes a software for data processing.

Fig. 4.2. Dewetron universal data acquisition instrument DEWE-43. All sensors will be connected to it, and it will be connected by USB to a computer. (www.dewetron.com)
5. **DETERMINATION OF TEST BENCH INERTIAL PARAMETERS**

The custom parts of the test bench have been fabricated in the “Laboratori Comú” of the Department of Mechanical Engineering at ETSEIB, and the standard parts have been bought directly to the manufacturers. The custom steel parts have been blackened to avoid rusting. Then, the test bench has been assembled and placed in the Department.

After that, the test bench should be tested in order to know the parameters of mass and moment of inertia and the exact coordinates of the centre of mass of the parts that rotate around the knee and act as the leg.

### 5.1. Mass

The parts that compound the leg of the test bench (they are the ones that rotate around the joint) should be weighed carefully.

### 5.2. Centre of mass

It is important to know the position of the centre of mass and the distance between it and the centre of rotation (the knee). Hanging adequately the leg from one point and taking into account the symmetries of the design, it can be found the exact place of the centre of mass.

### 5.3. Moment of inertia

Every rigid body hanged from a point which is not the centre of mass, oscillates when it is displaced from its position of equilibrium. This system receives the name of physical pendulum, and allows to calculate the moment of inertia of the body about the centre of rotation. Fig. 5.1 shows the body considered here.
Fig. 5.1. Dynamic parameters of the system: $I_S$ is the moment of inertia about the centre of rotation, $L_G$ is the distance between centre of rotation and centre of mass, $M$ is the mass of the leg, $g$ is the gravity constant and $\theta$ is the angle displaced from the position of equilibrium.

According to the angular momentum theorem it is obtained that:

$$-MgL_G \sin \theta = I_S \cdot \dot{\theta} \quad (5.1)$$

And, if small displacements are considered ($\sin \theta \approx \theta$), the pendulum describes a simple harmonic motion whose period is:

$$T = 2\pi \left( \frac{I_S}{MgL_G} \right)^{1/2} \quad (5.2)$$

So, the moment of inertia about the centre of rotation can be calculated from the measurement of the oscillation period:

$$I_S = \frac{T^2 MgL_G}{4\pi^2} \quad (5.3)$$
6. PRELIMINARY DESIGN OF THE EXPERIMENTS

The objective of the test bench is to study with precision the dynamic performance of the knee actuation system presented in Chapter 2. That includes knowing how much power is dissipated in the ball screw transmission during forward and inverse performance. It can be calculated through the application of the energy theorem, which is presented below.

6.1. Application of the energy theorem

The energy theorem states that the mechanical energy time derivative of a system is equal to the power developed by non-conservative forces:

$$\frac{dE_m}{dt} = P_{ncf} \quad (6.1)$$

The system considered is the test bench leg and the transmission of the actuation system from the electrical motor to the leg, but not the motor itself (Fig. 6.1).

Fig. 6.1. The energy theorem is applied to the system composed by the test bench leg and the transmission (which is not shown in the figure). Relevant coordinates and parameters are shown.
The mechanical energy includes the kinetic energy and the potential energy:

\[ E_m = K + U \quad (6.2) \]

The forces and torques that develop power are: the torque produced by the electrical motor, the friction forces in the ball screw transmission (which are unknown) and the wrench exerted on test bench foot. So, substituting in Eq. (6.1):

\[ \dot{K} + \dot{U} = P_T + P_{\text{diss}} + \dot{P}_F \quad (6.3) \]

The kinetic energy of the test bench and its time derivative are:

\[ K = \frac{1}{2} I_s \cdot \dot{\theta}^2 \quad (6.4) \]
\[ \dot{K} = \frac{dK}{dt} = I_s \cdot \dot{\theta} \cdot \ddot{\theta} \quad (6.5) \]

And the potential energy of the test bench and its derivative are:

\[ U = M \cdot g \cdot L_g \cdot (1 - \cos \theta) \quad (6.6) \]
\[ \dot{U} = \frac{dU}{dt} = M \cdot g \cdot L_g \cdot \sin \theta \cdot \dot{\theta} \quad (6.7) \]

The power developed by the electrical motor is calculated thanks to its torque and its angular velocity:

\[ P_T = T \cdot \omega \quad (6.8) \]

As there is no movement of point P in the direction of \( F_2 \) and there is no rotation of the solid about point P, the power developed by the external wrench exerted on the foot is the power developed by force 1:

\[ \dot{P}_F = F_1 \cdot v \quad (6.9) \]

where \( v \) is the velocity perpendicular to the leg of point P of the solid.

The power dissipated by the friction forces at the ball screw transmission is unknown (but always negative), and it will be found with the experiments in order to find the system efficiency.
6.2. First kind of experiments

There is no external wrench exerted on the foot (so $F_1 = 0$) and the movement of the leg is controlled:

$$\dot{K} + \dot{U} = P_T + P_{\text{diss}} \quad (6.10)$$

Two situations could happen during the performance of the actuation system:

(a) The mechanical energy derivative is positive: the power of the electrical motor is also positive and the actuation system has a forward performance. The dissipated power in the transmission is negative. In this case, that dissipated power can be expressed in another way: as a forward efficiency of the transmission ($\eta_f < 1$), which means that not all the power given by the motor arrives to the leg:

$$\eta_f = \frac{\dot{K} + \dot{U}}{P_T} \quad (6.11)$$

(b) The mechanical energy derivative is negative: the power of the motor is negative, which means that it is working as a brake and it has an inverse performance. It can also be defined an inverse efficiency of the transmission ($\eta_i < 1$) as another way to express the power dissipated in it:

$$\eta_i = \frac{P_T}{\dot{K} + \dot{U}} \quad (6.12)$$

In that way, knowing the power dissipated and the two efficiencies of the transmission, it is known how will perform the actuation system to swing the leg in the two conditions described.

6.3. Second kind of experiments

There is an external wrench exerted on the foot and the movement of the leg is controlled. The energy theorem has the general form:

$$\dot{K} + \dot{U} = P_T + P_{\text{diss}} + P_F \quad (6.13)$$
As before, there could happen two things:

(a) The power of the electrical motor is positive and the actuation system has a forward
performance. The power dissipated in the transmission is negative (representing a
power loss) and so its forward efficiency ($\eta_f < 1$) can be defined as:

$$\eta_f = \frac{K + U - P_F}{P_T} \quad (6.14)$$

(b) The power of the motor is negative, so it works as a brake and it has an inverse
performance. In this case, the power loss in the transmission can be represented
by means of its inverse efficiency ($\eta_i < 1$) as:

$$\eta_i = \frac{P_T}{K + U - P_F} \quad (6.15)$$

The same actuation system efficiencies can be found for this situation.

6.4. Third kind of experiments

The third type of experiments that could be done correspond to static configurations of the
system. There is an external wrench exerted on the foot and the leg is locked in a certain
configuration. The torque of the motor necessary to maintain that configuration can be
measured and the torque exerted by the friction forces at the ball screw calculated.
7. SOCIAL AND ENVIRONMENTAL IMPACT AND ECONOMIC STUDY

7.1. Social impact

This project is focused on the design and the assembling of a test bench to analyse an active orthosis actuation system in order to improve its design and performance. Its social impact corresponds to the influence that could have the prototype of the active knee-ankle-foot orthosis with this actuation system, which could be commercialized in the future and be helpful for thousands of spinal cord injured patients. Without any doubt, the life of those patients and their caregivers would be greatly improved by means of this rehabilitation device.

Sistiaga [2012, pp. 67-68] made a scope of the situation of spinal-cord-injured patients in Spain and studied the number of potentials customers for a commercialized SCKAFO. The latest data of the National Statistical Institute [INE, 2008] is that there are 108000 people with some kind of spinal cord injury in Spain. It should be taken into account that only some of those patients could benefit from the orthosis (because of the kind of the injury).

7.2. Environmental impact

The analysis of the environmental impact in Spain is regulated by the Law 21/2013 (Ley 21/2013, de 9 de diciembre, de Evaluación Ambiental). Due to the characteristics of the project, it is not necessary to make a complete environmental impact study.

During the design phase, there has been a responsible use of electricity, pen and paper. It has been used also a computer for the CAD program, the calculations and the writing. Each product (pen, paper, etc.) has been adequately recycled in the corresponding container after its use.

The test bench parts have been produced in the machining centre associated to the Department of Mechanical Engineering of ETSEIB. During this manufacturing phase, a coolant and a synthetic oil have been used for the machines. The coolant is usually re-used in the machines and after two years it is replaced by the workers and sent to a specialized
company for its management. The synthetic oil is also recycled and, after one year, replaced by the workers and sent to an external company which manages it.

At the end of test bench life cycle, it should be correctly recycled: electronic elements should be separated and recycled, and steel and aluminium parts should be disassembled and sent each one to the specific container for its recycling.

7.3. Economic study

The economic cost of this project consists in variable and fixed costs. Some costs are associated to the designing process and other costs to the manufacturing of the device.

The design phase, considered of 6 months, includes the following costs: hardware and software used, office equipment, salaries of the student and the supervisors, administrative costs, energy and consumable material.

The hardware used is a computer of 600 € whose life expectancy is of 4 years. The software consisted in: the CAD program Solidworks®, with a license of 1 year for 3000 €; and Matlab® for the calculations, which has a 1 year license for 2000 €. The salaries are 15 €/h for the student (with a project working time of 750 hours) and 50 €/h for each of the two supervisors (with a supervision time of 50 hours each of them). The office equipment has a value of 240 € and a life expectancy of 10 years. The administrative costs have been of 35 € and the consumable material costs, mainly paper, of 10 €. The energy consumed corresponds to the electricity for the light and the computer. It has been supposed that they were on during the whole working time (750 hours). The lights have a power of 240 W and the computer of 90 W, so the total energy consumed is of 247,5 kWh. And the price of electricity is 0,2 €/kWh.

It should be mentioned that the cost of the working place has not been considered because it is unknown. Table 7.1 contains a summary of all those costs.
The manufacturing phase has lasted two weeks and includes: the production of the custom parts (11 of steel and 2 of aluminium) in the machining centre and the standard parts and sensors bought. It also includes the blackening of the steel parts. Table 7.2 contains all the manufacturing costs.

Table 7.1. Costs during the design phase.

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
<th>Quantity</th>
<th>Total cost [€]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Computer</td>
<td>150 €/year</td>
<td>0,5 years</td>
<td>75,00</td>
</tr>
<tr>
<td>Solidworks license</td>
<td>3000 €/year</td>
<td>0,5 years</td>
<td>1500,00</td>
</tr>
<tr>
<td>Matlab license</td>
<td>2000 €/year</td>
<td>0,5 years</td>
<td>1000,00</td>
</tr>
<tr>
<td>Student</td>
<td>10 €/h</td>
<td>750 hours</td>
<td>7500,00</td>
</tr>
<tr>
<td>Supervisors</td>
<td>50 €/h</td>
<td>100 hours</td>
<td>5000,00</td>
</tr>
<tr>
<td>Office equipment</td>
<td>24 €/year</td>
<td>0,5 years</td>
<td>12,00</td>
</tr>
<tr>
<td>Energy</td>
<td>0,2 €/kWh</td>
<td>247,5 kWh</td>
<td>49,50</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td></td>
<td><strong>15136,50</strong></td>
</tr>
</tbody>
</table>
Table 7.2. Costs of the manufacturing and assembling phase.

<table>
<thead>
<tr>
<th>Material</th>
<th>Quantity</th>
<th>Unit Cost</th>
<th>Total Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Utilcell load cell</td>
<td>3</td>
<td>41,02</td>
<td>123,06</td>
</tr>
<tr>
<td>Plain bearings IGUS</td>
<td>2</td>
<td>10,01</td>
<td>20,02</td>
</tr>
<tr>
<td>Fasteners</td>
<td>-</td>
<td>-</td>
<td>25,00</td>
</tr>
<tr>
<td><strong>Total cost</strong></td>
<td></td>
<td></td>
<td><strong>1208,30€</strong></td>
</tr>
</tbody>
</table>

Therefore, the total cost of the project has been of **16344,80 €**.
CONCLUSIONS AND FUTURE WORK

In this project, a test bench to characterize the dynamic performance of the orthosis actuation system that is being developed at the Department of Mechanical Engineering of ETSEIB has been designed and built.

After studying the state of the art, the operating ranges of the actuation system and the requirements of the bench, the necessary sensors have been selected, that is, an absolute encoder to measure knee angle and three load cells to determine the external wrench exerted on the foot. The sensors have enough resolution and the load cells have a safety factor of 4.90.

For the mechanical design, the mass and the moment of inertia of a human leg have been taken as reference. The design permits an easy adaptation, changing only two simple parts, and can be used to test different knee actuation systems. The strength of all the parts has been checked and the safety factors of the bearings and the most critical bolted union are 1.52 and 1.61, respectively. The test bench has been manufactured, assembled and calibrated. And a summary of all the costs involved in the design and manufacturing has also been done.

A brief description of the experiments that will be done with the device designed in this project and the orthosis actuation system developed by the Department of Mechanical Engineering of ETSEIB has been given in this work. That is the starting point for the tests that should be completed to characterize the actuation system. Finding the efficiencies and analysing its dynamic performance will lead to an improvement on its design. And it also will allow to make modelling and optimization of the mentioned actuation system.

This project expects to be a contribution towards the development of the stance control knee-ankle-foot orthosis that is being designed by the Biomechanical Engineering Group of CREB-UPC in the framework of the National R+D Project: Diseño de una ortesis activa innovadora para la marcha de lesionados medulares incompletos mediante métodos de análisis y predicción de movimiento y modelos músculo-esqueléticos complejos (ref. DPI2012-38331-C03-02), for the years 2013-2015.
BIBLIOGRAPHY

Bibliographic references


SISTIAGA, J. Disseny mecànic d’una ortesi activa per a lesionats medul·lars a partir de l’anàlisi dinàmica de la marxa humana. Projecte de Fi de Carrera, ETSEIB, UPC. 2012.


Other bibliography


