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Load bearing performance of mechanical joints inspired by elbow of quadrupedal mammals

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Abstract

One of the biggest issues of the mechanical cylindrical joints is related to premature wear appearing. Application of bioinspiration principles in an engineering context taking advantage of smart solutions offered by nature in terms of kinematic joints could be a way of solving those problems. This work is focussed on joints of one degrees of freedom in rotation (revolute or ginglymus joints in biological terms), as this is one of the most common type of mechanical joints. This type of joints can be found in the elbow of some quadrupedal mammals. The articular morphology of the elbow of these animals differs in the presence/absence of a trochlear sulcus. In this study, bio-inspired mechanical joints based on these morphologies (with/without trochlea sulcus) were designed and numerically tested. Their load bearing performance was numerically analysed. This was done through contact simulations using the finite element method under different external loading conditions (axial load, radial load and turnover moment). Results showed that the tested morphologies behave differently in transmission of external mechanical loads. It was found that bio-inspired joints without trochlea sulcus showed to be more specialized in the bearing of turnover moments. Bio-inspired joints with trochlea sulcus are more suitable for supporting combined loads (axial and radial load and turnover moments). Learning about the natural rules of mechanical design can provide new insights to improve the design of current mechanical joints.

1. Introduction

Kinematic joints are connections between two or more independent parts of a multibody system that allow relative motion between them. They are everywhere in natural and man-made systems: exoskeletons, endoskeletons, helicopters and robotic arms. In the medical and industrial fields, efforts of the scientific community are mainly focussed on improving the lifespan of joints.

This work proposes to apply bioinspiration principles in an engineering context taking advantage of the smart solutions offered by nature regarding kinematic joints. Nature has created different types of joints with different degrees of freedom (DOF), power transmission capacity and movement ranges. These joints are characterized by specialized materials, lubrication mechanisms and articular morphologies that make them efficient and durable (Picault \textit{et al} 2018, Egan \textit{et al} 2015).

The tribological functioning of biological joints have been widely studied (Nia \textit{et al} 2011). Their lubrication mechanisms, which are defined by interactions between the cartilage and the synovial fluid, allow them extremely low friction coefficient and therefore wear rates. However, the benefits of the geometry/topology of biological in wear performance has not been well addressed yet.

This work is focussed on joints of one DOF in rotation (revolute or ginglymus joints), as this is one of the most common type of mechanical joint. This type of joint can be found in the elbow of some quadrupedal mammals. In some animals, the elbow is formed by two well defined condyles: capitellum and trochlea (hereafter called morphology I) (see figure 1(a)). In the elbow of
these animals no trochlear sulcus appears. Some of the animals presenting this characteristic are, for example: Camelus bactrianus, Elephas maximus, Rhinoceros unicornis and Diceror bicornis. However, a slightly different morphology (hereafter called morphology II) can be found in the elbow of other quadrupedal mammals. This difference is the presence of a trochlear sulcus (see figure 1(b)). This is the case for example of Equus caballus, Antilope cervicapra, Connochoates gnou and Bison bison. The collection number of the analysed bones, which belong to the National Museum of Natural History in Paris, are presented in appendix A. To the best knowledge of the authors, no explanation has been proposed for the coexistence of these morphologies; neither a comparative analysis about their load bearing performance has been yet done. A better understanding of the coexistence of these morphologies and their functional advantages can provide new insights to improve the design of current mechanical joints.

Kinematic joints have been studied from a paleontological (phylogenetic purposes), medical (health purposes) and technological (industrial purposes) point of view, as described in the next paragraphs.

Paleontological studies have been carried out about the functionality of biological joints regarding to the evolution of the species. Concerning the elbow, several studies have been focussed on the analysis of the relation between its morphology and hunting style (Andersson 2004), locomotion (Patel 2005) or extinction/preservation (Fujiwara 2009) of a given species. Other research works, such as (Granotosky et al 2019), studied the range of motion of the elbow and how it affects feeding and locomotion performance. Even if these studies help to understand evolution, they usually provide qualitative results which make their exploitation difficult in a technological context through bioinspiration.

In the medical field, several studies have been performed to evaluate load transmission in biological joints. In the case of ginglymus joints, most of these studies were conducted in the human elbow (Alcid et al 2004, Mason et al 2005), hand phalange (Sancho-Bru et al 2001) and knee (Aduoni et al 2012, Amirudin et al 2014, Haut Donahue et al 2002, Massouros et al 2010; Mesfar and Shirazi-Adl 2005). These studies have motivated the design and the development of the prosthetic knee (Russell et al 2018, Sathasivam and Walker 1994), ankle (Au et al 2008) and elbow (Cil et al 2008, Prasad et al 2016). Some other works have studied load transmission in animal joints, for example on horse (Becker et al 2019, Harrison et al 2010, Panagiotopoulou et al 2016, Praet et al 2012), sheep (Lerner et al 2015; Picault et al 2018; Poncery et al 2019) and seahorse (Praet et al 2012). A limited number of scientific articles explore the technological advantages of biological joints for bioinspiration. Picault et al (2018) analysed contact pressure distribution in a sheep elbow under few cases of physiological loading conditions. In their study, however, the behaviour of the joint regarding transmission of axial loads and turnover moments was not analysed.

In the technological field, many studies have been performed with the aim of extending the lifetime of bearings, maximizing their load bearing performance and reducing premature wear appearing. One of the biggest issues of the cylindrical joints is related to premature wear appearing due to misalignment between inner and outer races of the bearing (Hili et al 2005, Messaoud et al 2011). Figure 2 presents an example of cylindrical joints present in the camshaft of the combustion engine. New bearing designs have been proposed with improved roller geometry (Dragoni 2013, Mermoz et al 2016, Poplawski et al 2001, Potočnik et al 2010, Zamponi et al 2009, Zupan and Prebil 2001), improved contact surface shape (Bryans et al 2015, Wei et al 2016), optimal surface finishing (Fesanghary and Khonsari 2013, Gherca et al 2013, Lin 2001, Qiu and Raeymaekers 2015, Wang et al 2003) and optimal dimensions (Boedo and Eshkabilov 2003, Papadopoulos et al 2011, Rezaei et al 2012). Despite these technological advances, a limit...
Figure 2. Example of cylindrical sliding bearings. The illustrated mechanism corresponds to the camshaft and the cylinder head of a combustion engine. The housing is made of two half parts that allow the assembly of the shaft.

seems to be reached, as they are still not enough to effectively prevent premature wear failure.

Reduced number of studies exist about bearings with non-cylindrical geometry (Boedo and Eshkabilov 2003), which might be due to associated manufacturing challenges. Nevertheless, current fabrication capabilities (additive manufacturing) are not technological barriers any more for the development of new designs.

In response to this, the aim of this study was to evaluate the load bearing performance of the two morphologies proposed by nature for the ginglymus elbow (with and without trochlear sulcus, see figure 1). It was assumed that the load bearing performance of these morphologies is related with a combination of their topology (number of bumps) and geometry (curvature). In order to verify this, a numerical experiment was carried out over bio-inspired joints simulating different loading conditions (radial loads, axial loads and turnover moments) using the finite element method (FEM). Load bearing was evaluated and compared by means of peak contact pressure values.

2. Methods

In this section, it is described the methodology followed to evaluate the functional advantages of the two different morphologies proposed by nature for the ginglymus elbow: morphology I (without trochlear sulcus) and morphology II (with trochlear sulcus) (see figure 1). Bio-inspired mechanical joints based on both morphologies were designed and numerically tested. For the design of these joints, the elbows of the camel (Camelus bactrianus) and the bison (Bison bison) were selected. These two animals were chosen since their elbows have similar dimensions. The use of bones of similar dimensions allow modifying as little as possible the original dimensions of the profiles during the creation of the mechanical joints. This modification is required to obtain mechanical joints with equivalent dimensions (equal radius and length) and to make a feasible comparison of their load bearing performance.

The used methodology is divided into three main steps as illustrated in figure 2. First, bio-inspired mechanical joints were designed (Section 2.1). Second, the load bearing performance of the bio-inspired mechanical joints were numerically analysed (section 2.2). This simulation results are presented in section 2.3.

2.1. Bio-inspired joints

Two groups of mechanical joints were created: one group corresponding to the camel-inspired joints (representing the group of animals of morphology I: without trochlear sulcus) and the other corresponding to the bison-inspired joints (representing the group of animals of morphology II: with trochlear sulcus). Shafts were inspired in the camel and bison humerus, while bearings were inspired in the camel and bison radius-ulna.

The creation process of the bio-inspired joints is illustrated in figure 4 and described in the next sections. The first step was the digitization of the articular geometry (section 2.1.1). The second step was the extraction of the joint revolution profiles (section 2.1.2). The last step was the creation of the bio-inspired joints (section 2.1.3).

2.1.1. Articular geometry

Dry bone specimens (bones without soft tissue) of the left elbow of the C. bactrianus and B. bison were used in this study (see figure 4(a)). These specimens belong to the National Museum of Natural History of Paris collection (Camelus bactrianus reference: MNHN-ZM-AE-2007-1435; Bison bison reference: MNHN-ZM-AC-1951-242). The articular geometry of the specimens was digitized using a 3D optical scanner (Gom ATOS III, Braunschweig, Germany) with a resolution of 0.02 mm (see figure 4(b)). Four triangle meshes were obtained: camel humerus (85.3 thousand points), camel radius-
ulna (74.6 thousand points), bison humerus (86.8 thousand points), and bison radius-ulna (96.8 thousand points).

These triangle meshes were subsequently converted into a surface and smoothed in a CAD software (CATIA V5, Dassault Systèmes, France).

2.1.2. Morphological profiles
Using CATIA V5, ten morphological profiles were obtained from each digitized humerus surface (camel and bison). One radius-ulna morphological profile was obtained from each digitized radius-ulna surface (camel and bison). The reason for doing so was to...
recreate ten different flexion-extension angular positions of the elbow. The flexion-extension angle of the elbow influences the articular contact as demonstrated by Goel et al (1982). The morphological profiles are depicted in figure 5 in blue. They were obtained performing an intersection between the digitized surfaces and equiangular planes (5° of angular step) passing through the rotation axis. The 3D surfaces were considered as revolution surfaces and their axes were extracted by best fitting.

The last step of the generation of the morphological profiles was the scaling. The objective of this scaling was to obtain equivalent mechanical joints with the same average radius and length. This is necessary to make viable the comparison of the joints in terms of load transmission. The average radius of the distal articular surface of the camel and bison humerus was of 25.3 mm and 20.1 mm respectively. Their length was of 64.3 mm and 71.7 mm respectively. A scaling was performed, and the obtained dimensions of the joints were: 68 mm of length and 22.7 mm of average radius.

The achieved morphological profiles from the camel and bison are presented in figures 6(a) and (b) respectively. The profile of the bearing (radio-ulna) is represented by the dotted line and with solid lines the profiles of the shafts (humerus). In figure 6, the anatomical regions of the morphological profiles are presented for both groups. The curvature (the inverse of the radius) of the bearing profiles (derived from the radio-ulna) is presented in a light shading. These morphological profiles were used for the creation of the mechanical joints.

2.1.3. Bio-inspired mechanical joints
The bio-inspired mechanical joints were composed of two parts: shaft and bearing. For the creation of the bearings (for camel and bison), the morphological profiles coming from the radius-ulna were used. For the creation of the shafts, the morphological profiles extracted from the humerus were used. Ten shafts were created for the camel and ten for the bison. Combining the shafts with the bearing, ten mechanical joints were created.

The creation of the parts (shaft and bearing) was done by means of a revolution of the aforementioned profiles. The bio-inspired mechanical joints can be seen in figure 7 in an exploded view with the overall dimensions of the parts. The shafts were considered to be made up of 4340 alloy steel (Young’s modulus of 193 000 MPa and Poisson’s ratio of 0.28). The bearings were considered to be made up of polyoxymethylene (Young’s modulus of 3000 MPa and Poisson’s ratio of 0.35). Both materials were considered to be linear elastic and isotropic. The average mass of the camel-inspired shafts was estimated to be 325.8 g while the average mass of the bison-inspired shafts was estimated to be 306.9 g.

2.2. Contact mechanics simulation
In this section, the numerical FEM models for the contact analysis of the camel-inspired and bison-inspired mechanical joints are presented. This section is divided into three subsections. First, the meshing of the CAD models is presented in section 2.2.1. Second, the boundary conditions and the external loads (axial load, radial load and turnover moment) are described in section 2.2.2. Third, the execution of the simulations is explained in section 2.2.3.

2.2.1. Meshing
The shaft and bearings were imported into NX 11.0 software (Siemens, Germany) and meshed individually using hexahedral linear elements. It was demonstrated that this type of elements behave better for contact mechanics simulations (Maas et al 2016). A mesh convergence analysis was performed using an equivalent model made up of two cylinders. The results of the numerical simulations with the cylinders were compared with the analytical Hertzian solution. Elements with an average edge length of approximately 1 mm were found to provide an acceptable compromise between result accuracy and computational efficiency. The average number of elements, nodes and DOF per model are indicated in table 1. The meshed models of (a) camel-inspired
joint and (b) bison-inspired joint are shown in figure 7.

2.2.2. Boundary conditions and loading set-up

The shafts and bearings were initially assembled making coincident their rotation axes and their lateral faces parallel and aligned. All the DOF of the bottom face of the bearings were restricted. Rotation around \( \vec{x} \) and \( \vec{y} \) axes was restricted on the top face of the shafts. The inner surface of the bearings was selected as slave, and the outer surface of the shafts parts was selected as master. Unidirectional and frictionless surface-to-surface contact was defined between slave and master surfaces.

Three simulations sets were defined to evaluate the bearing performance of the created joints regarding axial loads (simulation set 1), turnover moments (simulation set 2) and combined loads (simulation set 3). For a feasible comparison, the magnitude of the applied load was the same and equal to 5.2 kN for all simulations. It was assumed that all loads act in the same plane (that normal to the \( \vec{z} \) axis) in all simulations.

In the simulation set 1, the bearing performance regarding axial loads \( (F_a) \) was analysed. The studied range of axial load went from \(-2.6 \text{ kN}\) to \(2.6 \text{ kN}\). The direction of the force considered as positive is illustrated in figure 7. The turnover moment was zero for these simulations. The applied radial load \( (F_r) \) was calculated from the following relation to obtain a total constant load: \( \sqrt{F_a^2 + F_r^2} = 5.2 \text{ kN} \). Simulation set 1 was performed with all the joints of both groups (camel and bison-inspired joints).

In the simulation set 2, the bearing performance regarding turnover moments \( (M_t) \) was analysed. The axial load was zero for these simulations. The studied range of turnover moment went from \(-138 \text{ Nm} \) to \(138 \text{ Nm} \). A couple of off-centred loads was applied in radial direction to obtain the desired moment value. These loads were applied at 115 mm from the centre of the piece keeping a total magnitude equal to 5.2 kN. Simulation set 2 was performed with all the joints of both groups (camel and bison-inspired joints).

The simulation set 3 aimed to analyse the bearing performance regarding combined loads (axial loads and turnover moments). This simulation set was performed only with one geometry of each group (one for camel and one for bison) due to computational costs. This geometry was chosen as that whose response was the closest to the average response of all
Figure 7. Example of used meshing size in (a) camel-inspired joint; (b) bison-inspired joint. A detail of the meshing is presented. P0 point indicates the midpoint of the upper face of the shaft. \( \vec{F}_{a} \) represents the axial force. \( \vec{F}_{r} \) represents the radial force. \( \vec{M}_{t} \) represents the turnover moment. Represented loads are assumed positive in the axis direction. Dimensions are expressed in mm.

Table 1. Average number of elements, nodes and DOF of the numerical models of the mechanical joints.

| Model   | No of elements | No of nodes | DOF  |
|---------|----------------|-------------|------|
| Camel   | 39 600         | 53 502      | 1891 |
| Bison   | 41 400         | 55 393      | 1891 |

the tested geometries. The analysed range of axial load went from \(-2.6 \text{ kN}\) to \(2.6 \text{ kN}\) and the analysed range of turnover moment went from \(-138 \text{ Nm}\) to \(138 \text{ Nm}\). In order to generate the desired moment, a couple of off-centred loads with a total value of \(5.2 \text{ kN}\) was applied in radial direction.

The loading conditions of each simulation set can be seen in table 2. The conditions of the simulation set 1 (axial loads) are denominated as \(S_{ai}\) (with \( i = 1:11 \)) and can be seen in blue colour in table 2. The loading conditions of the simulation set 2 (turnover moments) denominated as \(S_{mj}\) (with \( j = 1:12 \)) and can be seen in brown colour in table 2. The loading conditions of the simulation set 3 (combined loads) are denominated as \(S_{ck}\) (with \( k = 1:120 \)) and can be seen in black colour in table 2.

2.2.3. Simulation execution

A total of 720 numerical simulations of contact were performed for both groups of joints (camel and bison-inspired joints). All the simulations were performed using Samcef 2015 V.17.1 (Samtech, Belgium) solver on a desktop computer (Intel (R) Xeon (R) Gold 6134, 256 Go RAM) running a 64 bit operating system. The maximum allowable penetration depth was set on 0 mm.

2.3. Result treatment

Using NX 11.0 as postprocessor, the five highest contact pressure values were averaged and considered as the peak contact pressures (\(P_{\text{max}}\)) for each simulation. This with the aim of mitigating numerical errors due to the discretisation of the geometry during the meshing process.

Confidence intervals of the mean values (\(P_{\text{max}}\)) were calculated from the results of the simulations of the different geometries with the same simulation conditions. Confidence intervals were determined as twice the standard deviation (2\(\sigma\)). From these results, the geometry whose response was the closest to the average response of all the geometries was chosen for running the simulation set 3.

3. Results

Peak contact pressures obtained from the simulations are reported in the following sections as a function of the applied external loads (axial load, turnover moment and combined loads).

3.1. Bearing performance regarding axial loads

In figure 8(a), \(P_{\text{max}}\) as a function of the applied axial load is presented. The applied turnover moment was zero for these simulations. In figure 8(a), for each axial load value, central mark points (orange triangles for camel and green circles for the bison) represent \(P_{\text{max}}\) value. The confidence interval for each set of results is represented also in figure 8(a) (orange shading for camel and green shading for bison).
Table 2. Loading conditions for the set of numerical simulations. \( \text{F}_a \) and \( \text{F}_r \) values are indicated on the top of the table. On the left column, \( M_t \) is indicated as a function of \( \text{F}_r \).

| \( M_t \) (Nm) | \( \text{F}_r \) (kN) | \( \text{F}_a \) (kN) | \( \text{F}_r^* \) (°) |
|----------------|----------------|----------------|----------------|
| 4.886          | 5.023          | 5.121          | 5.180          | 5.200          | 5.121          | 5.023          | 4.886          | 4.713          | 4.503          |
| −1.779         | −1.346         | −0.903         | −0.453         | 0.000          | 0.453          | 0.903          | 1.346          | 1.779          | 2.198          | 2.600          |

\( \text{F}_r^* \) values:
- \( \text{F}_r^* \) (−26.5) Sc1 Sc13 Sc25 Sc37 Sm1 Sc49 Sc61 Sc73 Sc85 Sc97 Sc109
- \( \text{F}_r^* \) (−22.1) Sc2 Sc14 Sc26 Sc38 Sm2 Sc50 Sc62 Sc74 Sc86 Sc98 Sc110
- \( \text{F}_r^* \) (−17.7) Sc3 Sc15 Sc27 Sc39 Sm3 Sc51 Sc63 Sc75 Sc87 Sc99 Sc111
- \( \text{F}_r^* \) (−13.3) Sc4 Sc16 Sc28 Sc40 Sm4 Sc52 Sc64 Sc76 Sc88 Sc100 Sc112
- \( \text{F}_r^* \) (−8.8) Sc5 Sc17 Sc29 Sc41 Sm5 Sc65 Sc77 Sc89 Sc101 Sc113
- \( \text{F}_r^* \) (−4.4) Sc6 Sc18 Sc30 Sc42 Sm6 Sc54 Sc66 Sc78 Sc90 Sc102 Sc114
- \( \text{F}_r^* \) (0) Sa1 Sa2 Sa3 Sa4 Sa5 Sa6 Sa7 Sa8 Sa9 Sa10 Sa11
- \( \text{F}_r^* \) (4.4) Sc7 Sc19 Sc31 Sc43 Sm7 Sc55 Sc67 Sc79 Sc91 Sc103 Sc115
- \( \text{F}_r^* \) (8.8) Sc8 Sc20 Sc32 Sc44 Sm8 Sc56 Sc68 Sc80 Sc92 Sc104 Sc116
- \( \text{F}_r^* \) (13.3) Sc9 Sc21 Sc33 Sc45 Sm9 Sc57 Sc69 Sc81 Sc93 Sc105 Sc117
- \( \text{F}_r^* \) (17.7) Sc10 Sc22 Sc34 Sc46 Sm10 Sc58 Sc70 Sc82 Sc94 Sc106 Sc118
- \( \text{F}_r^* \) (22.1) Sc11 Sc23 Sc35 Sc47 Sm11 Sc59 Sc71 Sc83 Sc95 Sc107 Sc119
- \( \text{F}_r^* \) (26.5) Sc12 Sc24 Sc36 Sc48 Sm12 Sc60 Sc72 Sc84 Sc96 Sc108 Sc120

Figure 8. (a) Peak contact pressure values as a function of axial load are presented for the ten camel and bison-inspired joints. A secondary axis represents the applied radial load. (b) Peak contact pressure values as a function of turnover moment are presented for the ten camel and bison-inspired joints. The central markers (orange triangles for the camel and green circles for the bison) represent \( P_{\text{max}} \) value for each turnover moment value and the light shading (orange shading for the camel and green shading for the bison) represent the confidence intervals.

In figure 8(a), an asymmetrical response of the \( P_{\text{max}} \) value with respect to the zero axial load value can be observed for the bison-inspired joints. The highest value of positive axial load was 65% greater regarding to the highest negative value. The camel-inspired joints had a constant and more...
symmetrical response, even if the capacity to support positive axial loads was slightly greater (19%) respect to the capacity to support negative axial loads. Comparing the response of both sets of joints, it can be noticed that the camel-inspired joints had a greater capacity (49% higher than bison) to support negative axial loads. Regarding to the supported positive axial loads, the bison-inspired joints had a greater capacity (18%) in comparison with the camel-inspired joints.

The camel-inspired joints had lower peak contact pressures almost in all ranges except from 2.19 kN to 2.60 kN. The $P_{\text{max}}$ value for all simulations of the set 1 was on average 37% lower for the camel-inspired joint than for the bison-inspired joint. The minimum $P_{\text{max}}$ value for the bison-inspired joints was 87.78 MPa, which was obtained under an axial load of $-0.45$ kN. The minimum $P_{\text{max}}$ value for the camel-inspired joint was 36% lower than that of the bison, and it was obtained under axial load of 1.34 kN.

Among the ten tested joints (for bison and camel), those whose response was the closest to the average response of all the geometries were: joint N°2 (CJ$_2$) in the case of the camel-inspired joints group and joint N°6 (BJ$_6$) in the bison-inspired joints group. These two joints were used for the following analysis of the contact areas (figure 9).

Figure 9 shows the contact distribution, from a bottom view, on the shaft of CJ$_2$ and BJ$_6$ for
different values of axial loads. The bearing is represented in a wireframe. In this figure, it can be noticed that the contact occurs mainly near the shaft ends in both BJ₆ and CJ₂. For the considered loading conditions, the deepest part of the capitellotrochlear sulcus did not enter in contact in CJ₂ neither in BJ₆. Application of negative axial loads on BJ₆ generates a high concentration of the pressure in a reduced area (edge effects). At an axial load of -1.77 kN and 2.6 kN CJ₂ presented edge effects. At axial load of -0.9 kN BJ₆ presented edge effects. At axial load of 1.77 kN, BJ₆ did not converge while CJ₂ did. The contact reaction forces could not be counteracted by the applied radial load, which generates a loss of static equilibrium and therefore of contact. For the same conditions, in most of the cases, the CJ₂ had lower $P_{\text{max}}$ values and larger contact areas than the BJ₆.

### 3.2. Bearing performance regarding turnover moments

In figure 8(b), $P_{\text{max}}$ as a function of the applied turnover moments is presented. The applied axial load was zero and the radial load was 5.2 kN for these simulations. In figure 8(b), central mark points (orange triangles for camel and green circles for the bison) represent the obtained $P_{\text{max}}$ for each turnover moment value, and the light shading (orange shading for camel and green shading for bison) represents the confidence interval.

In figure 8(b), an asymmetrical response of the $P_{\text{max}}$ value with respect to the zero turnover moment value can be seen specially in the bison-inspired joints. On the contrary, the camel-inspired joints had more constant response. When positive turnover moments were applied, the bison-inspired joints produced lower $P_{\text{max}}$ (better load distribution) than when negative turnover moments were applied. The confidence intervals of the simulation results of the bison-inspired joint for negative turnover moments were larger than for positive turnover moments.

The camel-inspired joint had lower $P_{\text{max}}$ almost in all range except from 46 Nm to 92 Nm. The $P_{\text{max}}$ value for all simulations of the set 2 was on average 30 % lower for the camel-inspired joint than for the bison-inspired joint. The minimum $P_{\text{max}}$ value for the camel-inspired joints was obtained under turnover moment of 0 Nm. The minimum $P_{\text{max}}$ value for the bison-inspired joints was obtained under turnover moment of 46 Nm. Both values were almost the same around 76.85 MPa.

Among the ten tested joints for each group, CJ₂ and BJ₆ were the closest to the average response of all the geometries. Figure 10 shows the contact distribution, from a bottom view, on the shaft of CJ₂ and BJ₆ for different values of turnover moments. The bearing is represented in a wireframe. In this figure, it can be noticed that the contact occurs mainly near the shaft extremities in both cases. Also, it was seen that under the explored range of loading conditions the deepest part of the capitellotrochlear sulcus did not come in contact in CJ₂ neither in BJ₆. At turnover moments of -138 Nm and 138 Nm CJ₂ presented edge effect. At a turnover moment of 130 Nm, the simulation with BJ₆ did not converge while CJ₂ did. The static equilibrium could not be satisfied under the applied conditions in that case. The contact reaction forces could not be counteracted by the applied radial load, which generates a loss of static equilibrium and therefore of contact. In most cases, the CJ₂ had lower $P_{\text{max}}$ values and larger contact areas than the BJ₆.

### 3.3. Bearing performance regarding combined loads

In figure 11, it can be seen $P_{\text{max}}$ as a function of the applied axial load and turnover moment for CJ₂ and BJ₆. In the left-hand side of this figure, it can be seen that CJ₂ behaves well (lower peak contact pressure values) under positive turnover moments if the axial load is also positive, and under negative turnover moments if the axial load is also negative. It can also be seen that BJ₆ behaves well (lower peak contact pressure values) under positive turnover moments and positive axial loads. Table 3 presents a comparison of the peak contact pressure values of both joints (CJ₂ and BJ₆). In 37% of the analysed cases (cells in light green colour in table 3), CJ₂ had lower peak contact pressure with respect to the BJ₆, with a mean difference of 32%. In 6% of the cases (cells in light orange colour in table 3), BJ₆ had lower peak contact pressure regarding the CJ₂, with a mean difference of 10%. In 20% of the cases (cells in dark orange colour in table 3), BJ₆ did not converge while CJ₂ did. In 15 % of the cases (cells in dark green colour in table 3) CJ₂ did not converge while BJ₆ did. Simulations did not converge for any of the joints in 22% of the cases (blank cells in table 3). Besides having a lower peak contact pressure, CJ₂ also had a wider load range (the simulations converged in more cases) and with a greater mean difference in pressure than BJ₆.

In figure 12, the contact pressure distribution over the shafts of the CJ₂ and BJ₆ can be seen from the bottom view under their preferential loading conditions. The lowest contact pressure was obtained on the CJ₂, with a value of 55.65 MPa for loading conditions presented in the bottom part of the figure. These loading conditions correspond to a carrying angle of 30°. The contact under these conditions is characterized by two contact zones, with one larger than the other. The lowest value of BJ₆ was 76 MPa. The contact under these conditions is characterized by three contact zones, two large and one more narrow. These loading conditions correspond to a carrying angle of 15°.

### 4. Discussion and conclusions

In this study, the contact response of two groups of bio-inspired joints were analysed under application of...
external mechanical loads using FEM. The mechanical joints, inspired on the camel and bison elbow morphologies, behave differently during transmission of external loads.

The load bearing performance of the bison-inspired joints showed an asymmetrical response with respect to the zero axial load value. Similarly, an asymmetrical response of the same joints with respect to the zero turnover moment value was observed. This can be explained by the mid-sagittal plane asymmetry of the geometry of the joints. This shows a preferential loading side, which might be related with the physiological loading conditions of the studied joints. The obtained results from both joints showed that the carrying angle was not perpendicular to the rotation axis (around 15° for bison and 30° for camel) with medial-lateral loading inclination. Similar results were found for the human elbow, with an angle of 11–16° of medial-lateral loading (Ayhan and Ayhan, 2020).

The different morphological profiles extracted from the real bones showed that the morphology of the articular surface varies in function of the flexion-extension angle of the elbow. This variation translates into changes on the articular contact, which is in accordance with the results presented in Goel et al (1982).

It was found that the camel-inspired joint exhibited lower peak contact pressure values in a wider range than the bison-inspired joint. The camel-inspired joint presented larger contact areas than...
Figure 11. Peak contact pressure (in MPa) as a function of the applied axial load (in kN) (radial load as secondary axis (in kN)) and turnover moment (in Nm) for the camel-inspired joint (CJ2 geometry) on the left-hand side and for the bison-inspired joint (BJ6 geometry) on the right-hand side. The colour bar shows the peak contact pressure values from 50 MPa (in blue colour) to $\geq 130$ MPa (in dark red colour). The axial load axis is represented on the horizontal axis on the left side of each graph. The range of the axial load goes from $-3$ kN to 3 kN. A secondary axis, parallel to the axial load axis represents the radial loads. The horizontal axis represents the turnover moment. Turnover moment range goes from $-138$ Nm to 138 Nm.

Table 3. Relative difference expressed in percentage of the peak contact pressure obtained with the camel-inspired and the bison-inspired joints. In light green colour: results when the bison-inspired joint had lower peak pressure. In light orange colour: results when the camel-inspired joint had lower peak pressure. With the mark ‘B’: results when the simulation with the camel-inspired joint did not converge but the bison-inspired did. With the mark ‘C’: results when the simulation with the bison-inspired joint did not converge but the camel-inspired did. Empty cells: none of the simulations converged.

| Fr (kN) | Fa (kN) | $F_r^*$ (−26.5) | $F_r^*$ (−22.1) | $F_r^*$ (−17.7) | $F_r^*$ (−13.3) | $F_r^*$ (−8.8) | $F_r^*$ (−4.4) | $F_r^*$ (0) | $F_r^*$ (4.4) | $F_r^*$ (8.8) | $F_r^*$ (13.3) | $F_r^*$ (17.7) | $F_r^*$ (22.1) | $F_r^*$ (26.5) |
|--------|--------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| M (Nm) |        |                 |                 |                 |                 |                 |                 |                 |                 |                 |                 |                 |                 |                 |                 |
| 4.886  | 5.023  | 5.121           | 5.180           | 5.200           | 5.121           | 5.023           | 4.886           | 4.713           | 4.503           | 2.198           | 2.600           |                  |                 |                 |                 |
| −1.779 | −1.346 | −0.903          | −0.453          | 0.000           | 0.453           | 0.903           | 1.346           | 1.779           | 2.198           | 2.600           |                  |                 |                 |                 |

The bison-inspired joint, which might be caused by a greater congruence between the bearing and the shaft geometry (see figure 6). The congruence of biological joints has been interpreted by MacConaill (1953) as a functional gap intentionally created to allow an efficient lubrication of the joint. This might suggest that the elbow of the bison is more suitable for high-speed operation than the camel elbow. Additionally, the maximal tangential speed on the bison-inspired joint surface is lower than that of the camel-inspired joint given that the maximum radii of this joint are smaller, and therefore less wear is produced (assuming equal material properties).

The bison-inspired joint showed to be more specialized for bearing positive turnover moments in absence of axial loads. This can be explained by the geometry at the right end of the joint, which presents a smaller curvature regarding the same zone of the camel-inspired joint (see capitellum in figure 6(b)).

Results showed that the camel-inspired joint is more suitable for supporting axial and combined loads. The capacity of the camel-inspired joint to bear
axial loads can be explained by the groove derived from the capitellotrochlear sulcus that acts as a thrust collar. The depth of this groove is almost three times larger than that of the bison-inspired joint (see $\Delta r_C$ and $\Delta r_B$ in figure 6). This generates a larger projection surface in the radial plane (a plane orthogonal to the rotation axis) and therefore a better capacity for bearing axial loads. Similar results were found by Willing et al. (2014), who tested a similar geometry to that of the camel elbow (hourglass) against other geometries (cylindrical and concave).

In technological applications of load bearing at high speeds (i.e., those requiring the use of roller bearings), certain arrangements of bearings present an equivalent contact topology to that of the camel and bison elbow. The topology of the camel-inspired profile can be associated with a back-to-back bearing arrangement (see figure 13(a)), while the bison-inspired profile can be associated with a back-to-back and tandem bearing arrangement (see figure 13(b)). In accordance with our results, the back-to-back arrangement allows axial force absorption in both directions and presents a good rigidity against turnover moments. On the other hand, back-to-back and tandem bearing arrangement present a reinforced capacity to absorb turnover moments. This kind of arrangements has not been tested yet in the case of sliding bearings (or bushing bearings) because of the associated difficulties in the manufacturing processes (machining and assembly). Nevertheless, technologies of nowadays allow to perform complex geometries by additive manufacturing or 5-axes CNC machining. So technological barriers have been knocked down opening the way for development of new designs.

In synovial joints, muscles and ligaments preserve their stability. In contrast, in technological cases, there are three feasible options to preserve the stability of revolute joints. The first option is to let a unilateral joint (the housing covers less than half of the shaft and hence can be separated from the housing), this option is feasible if the directions of the efforts are well known and they do not change during operation. This is case for example of some rotatory drum dryers. The second option is to build a bilateral joint which its housing (female part) is made of two half parts to allow the assembly process of the shaft (see for example figure 2). The third option is to build a non-assembly joint by means of additive manufacturing. In this non-assembly joint, the male part is fabricated inside the female part with a controlled clearance. A manufacturability proof was done with the bison-inspired joint using additive manufacturing technology (selective laser melting (SLM)). The complete part was splinted at the end the process using electrical discharge machining for visualization purposes. The obtained joint is presented in figure 14. This type of procedures is just being explored by the industrial and scientific community and needs more study (see, for example, Boschetto and Bottini (2019)).

This study presents several limitations. First, cartilage and soft tissues, which may influence load transmission performance, were not considered. However, as this is a comparative study, the cushioning effects generated by a softer material affect both joints in equal manner (assuming small deformations). Therefore, contact pressure values will change while the global tendency of the contact pressure regarding the external loads will not. This can be assumed if the topology of the contact (number of contact zones) does not change. This is the case when the deformations are small regarding the curvature of the contacting profile and it corresponds to the assumption of the present work. Therefore, in the case of very soft materials or very high loads, the topology of the contact (the number of contact zones) may change, and the load bearing response of the joint will be different. The former case was not addressed in this work, but
it merits further investigation. Therefore, future work is required to determine the role of cartilage in pressure distribution, which might help to improve the designs of bio-inspired mechanical joints. The study of the contact mechanics of articular cartilage can be done in vitro, for example, using x-ray reconstruction of moving morphology as demonstrated by Tsai et al. (2020).

A second limitation is that the application of the applied external load was assumed to belong to the same plane. In most industrial application the radial load and turnover moment act in the same plane, which is in accordance with this assumption. For future studies, the load bearing behaviour of the joints must be explored while loads are applied from different planes. Future work is also required to develop a methodology for the design of bio-inspired mechanical joints that allows to synthesize geometrical parameters (congruency, geometrical symmetry, profile curvatures, depth of grooves) of kinematical joints in function of the loading conditions.

In conclusion, kinematically equivalent biological joints can present different morphologies that make each one the optimal compromise for the loading conditions. Learning about the natural rules of mechanical design can provide new insights to improve the design of current mechanical joints.

**Competing interests**

The authors have declared that no competing interests exist.

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Data availability statement

All data that support the findings of this study are included within the article (and any supplementary files).

Appendix A.

Reference to the collection of National Museum of Natural History in Paris.

| Species                          | Reference to the collection of National Museum of Natural History in Paris |
|---------------------------------|--------------------------------------------------------------------------|
| Antilope cervicapra             | MNHN-ZM-AC-1888-734                                                    |
| Bison bison                     | MNHN-ZM-AC-1951-242                                                    |
| Connochaetes gnu                | MNHN-ZM-AC-1976-344                                                    |
| Dicerorhinusicornis             | MNHN-ZM-AC-1944-278                                                    |
| Equus caballus                  | MNHN-ZM-AC-1932-46                                                    |
| Rhinoceros unicornis            | MNHN-ZM-AC-1885-734                                                    |
| Camelus bactrianus              | MNHN-ZM-AL-2007-1435                                                   |
| Elephas maximus                 | MNHN-ZM-1896-17                                                       |

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