Abstract—Force regulation is a challenging task of forceps used for robot-assisted surgical manipulation. To avoid excessive force applied on soft tissues, sophisticated sensors with computerized precise control are often required. Without using additional electronic elements, this paper presents a passive mechanism to maintain a constant contact force between forceps jaw tips and tissues given a pre-specified force magnitude. The mechanism consists of symmetric flexible structures specifically designed to generate a constant torque regardless of input rotation. The constant torque is converted to a constant force through an adjustable lever arm. When the force is further transmitted to jaw tips, it keeps a nearly constant contact force regardless of tissue stiffness and size. After a formulation to find the optimal mechanism configuration, the design is verified by comparing experiment and simulation results. A prototype of the adjustable constant-force forceps is finally illustrated and discussed. The novel forceps is expected to serve as a reliable alternative for robot-assisted surgeries.

Index Terms—Medical robots, robot-assisted surgery, force sensing, compliant mechanism, shape design.

I. INTRODUCTION

Minimally invasive surgeries (MIS) are advantageous to open surgeries because of reduced patient trauma and recovery time. Since a MIS is performed in a hard-to-reach place, its success requires sophisticated instruments and highly skilled surgeons. To advance the popularity and performance of MIS, numerous robot-assisted surgery instruments have been developed. One commercially available example is the da Vinci [1] surgical system. In a robot-assisted MIS, the assisting robot arms are attached with forceps for grasping soft, live tissues inside an expanded human body. Surgeons maneuver the robot-operated forceps through a human-machine interface. Unlike direct operation, surgeons lose sensing ability when using this indirect operation. Since excessive grasping force to tissues results in trauma and damage, a challenging task for manipulating the forceps inside human body is how to rebuild force perception.

The elastic moduli of soft tissues vary roughly from 100 Pa for brain to 100 kPa for soft cartilage [2]. Through an endoscope, the force applied to soft tissues can be predicted by observing tissue deformation at forceps tips. However, this visual feedback is far from accurate because tissue stiffness varies. Instead, there are several approaches to better obtain the tip force of forceps. One commonly used approach is to place a physical sensor on the forceps to detect the deformation caused by a grasping force. Since the force is only several newtons, precise sensors are often required. For example, strain gauges [3] and capacitive sensors have been used. An extensive review of forceps sensing can be found in Ref. [4]. Electronic sensors are limited by their sterilization and electromagnetic interference problems. Other researches employ inverse dynamics [5] and mechanical sensors [6] to circumvent these problems. After the force is indirectly sensed, the force signal is fed to a surgeon through a haptic device. Hence the surgeons can manually control the force on the tissue being grasped. To avoid errors caused by inexperienced operators and hand tremor, additional master-slave control algorithms may be developed to help maintain a safe contact force. Nevertheless, extra precautions have to be taken so that the surgical system is reliable and fail-safe.

Aimed at removing the needs for force sensing and control at the same time, this paper proposes an adjustable constant-force forceps for robot-assisted surgery. By using a compliant constant-toque mechanism (CTM) to transmit the required force to forceps tips, the grasping force can be maintained at a specified magnitude regardless of the size and stiffness of soft tissues. The grasping force can further be adjusted depending on the force requirements. As force regulation is achieved mechanically without using additional sensor electronics, it is useful in a MIS where compactness, cleanliness, and safety are major concerns.

A CTM is the rotary version of a linear constant-force mechanism (CFM). A CFM outputs a nearly constant force over a range of input displacements. It is useful when an unknown displacement is applied to a system the reaction force of which must be a specified constant regardless of the displacement. As the force is kept invariable without extra sensors and control electronics, a CFM can reduce control effort and increase reliability. Ready examples are the constant-force and constant-torque springs that made of one or more rolled strips of spring steel [7]. They are found in exercise and rehabilitation devices (e.g., Refs. [8–9]). A CFM has been proposed in our earlier work [10] to regulate the force of a robot end-effector.

In what follows, we begin by presenting the design concept of the adjustable constant-force forceps. This is followed by the optimal design of a CTM with an adjustable lever arm. The CTM is then verified experimentally for its constant-torque and adjustable constant-force characteristics. Finally, a prototype of the proposed forceps will be illustrated.

II. DESIGN CONCEPT OF ADJUSTABLE CONSTANT-FORCE FORCEPS

Fig. 1(a) shows a typical forceps. It has a pair of miniaturized jaws to be inserted into a patient's abdomen. A surgeon provides force at the handles to pull the cable, which actuates the jaws for grasping tissues within human body. The jaw mechanism is often designed as a slider crank where the
force is applied at the slider through an inextensible cable driven by the handles. Alternatively, the cable force may also be provided by a geared motor [11] or a pneumatic cylinder [6]. Fig. 1(b) shows a close view of the jaw mechanism. When the cable provides a force $F_c$, the transmitted tip force $F_t$ contacts the soft tissue in a direction normal to tip surface. The tip then slightly deforms the tissue to yield sufficient grasping force. The deformation causes the tip to displace $\Delta x$. Through a static analysis, the forces and displacements have the following relations.

$$F_c / F_t = L_2 \sin(\alpha/2) / 2L_1 \cos(\phi) = \Delta x / 2y$$

where $\phi = 2 \sin^{-1}(L_2 / 2) \sin(\alpha / 2)$

(1)

For a typical jaw mechanism, the ratio $\Delta y / \Delta x$ ranges from 5 to 10. Since tissue deformation $\Delta y$ is already small, the quantity of $\Delta x$ is further smaller. As $F_c$ is proportional to $\Delta x$, maintaining a constant tip force through the cable would depend on controlling the displacement $\Delta x$. This requires a high-quality actuator since $\Delta x$ is very small. An overly large cable displacement would result in an excessive grasping force on the tissue; a too smaller displacement would not provide a sufficient grasping force.

To solve the difficulty of grasping soft tissue, our idea is shown in Fig. 1(c). The cable is connected to the proposed constant-torque mechanism (CTM). It has four symmetric flexible arms that connect the outer rim to the central shaft. It exhibits a torque to angle relation as in Fig. 1(d). Unlike a linear spring, a CTM generates a nearly constant output torque over a rotation range. In Fig. 1(c), there is a lever arm extending from the outer rim. The cable is connected to slider B on the lever arm and the distance from the shaft center to slider B is denoted as $a$. When the shaft is driven, e.g., by a motor, it will deflect the flexible arms to generate a torque to balance with the cable force acting on the lever arm. When the cable is taut, the torque and cable force are related by

$$F_c = T(\theta) / (a \times \sin(\theta))$$

(2)

For a given $a$, the angle $\gamma$ can be shown to be near 90°, regardless of the position of slider A. Hence $\sin(\theta)$ is close to one. The cable force then mainly depends on $a$. It increases as length $a$ decreases. Fig. 1(e) shows the cable force $F_c$ of various arm lengths. The ability to vary the cable force makes the forceps adaptable to unknown soft tissues.

For a given cable length, sliders A and B move to adapt to various tissue sizes and stiffnesses, which results in a cable of unknown slackness. The CTM with its lever arm has to slightly rotate left or right from its upright position to reach the cable taut position before it can be driven. Shown on the bottom left corner of Fig. 1(c), this rotation range is denoted as $\Delta \theta$. The left extreme taut position is denoted as $\theta_l$. To drive the CTM, an actuator is rigidly attached to the shaft. The actuation starts at $\theta_l$ and then rotate $\theta$, clockwise. During the $\theta_l$, it first rotates the CTM to the taut position, then drives the central shaft to balance with the external cable force. The actual shaft rotation relative to the outer rim may be any value between $\theta_l - \Delta \theta$ and $\theta_l$. As long as this range falls within the constant torque region in Fig. 1(d), the actuator can provide a constant force $F_c$ to the jaw.

By using the CTM, driving the forceps requires only a less demanding actuator, e.g., a stepper motor, to give a rotation amount $\theta_l$ on the central shaft. The force will remain the same regardless of the cable taut position and tissue deformation (resulting from different tissue sizes and stiffnesses). There is no need to embed sensors on the forceps to feedback and control the grasping force. This benefit reduces the forceps complexity and makes it more reliable.

### III. DESIGN FORMULATION OF A CONSTANT-TORQUE MECHANISM

#### 3.1 Objective and constraint

The success of the proposed CTM in Fig. 1(c) relies on the shape of the flexible arms to deflect in a way that generates a range of constant torque. Ideally, the constant-torque range should be flat as indicated in Fig. 1(d). However, actual elastic mechanisms do not fully follow the ideal $T-\theta$ curve. As depicted in Fig. 2, there will be maximal and minimal peaks that make the curve fluctuate. This is similar to the structural snap-through buckling behavior, except the goal here is to reduce to gap between the peaks. To define a reasonable constant-torque (operational) region, we first characterize the $T-\theta$ curve of a typical CFM in Fig. 2. Given an input rotation where $\theta = 0 - \theta_l$, the torque first has to experience a preload
region from $\theta = 0 - \theta_1$. It is followed by the constant-torque region where the torque curve fluctuation in $\theta = \theta_1 - \theta_2$ has to be minimal. To ensure the durability of the deformation, the maximal stress at $\theta = \theta_1$ has to be as small as possible. Considering these two goals, we formulate the stress requirement as an objective function and fluctuation requirement as a constraint function. They are written on the right hand side of Fig. 2. Eq. (3a) minimizes the maximal stress at $\theta_1$. Eq. (3b) limits the fluctuation to be less than 5% of the average torque between $\theta_1$ and $\theta_2$ so that the portion in between is as flat as possible. Apparently the arm shape is the primarily design variable for the objective and constraint.

3.2 Parameterization of CTM shape

A CTM has a $T$-$\theta$ curve that differs from other elastic structures. Hence its flexural arm shape requires a subtle design. To seek for a proper design base on the objective and constraint in Eq. (3), the arm shape first has to be parameterized. We adopt Bezier curves [12] because of their ease of programming. The $x$ and $y$ coordinates of an $m$-th order Bezier curve are

$$x(t) = \sum_{i=0}^{m-1} Q_i B_i(t) ; \quad y(t) = \sum_{i=0}^{m-1} Q_i B_i(t)$$

where $B_i(t) = \frac{(m-i)!}{i!(m-1-i)!} t^i (1-t)^{m-1-i}$ and $t \in [0,1]$ (4)

Variables $Q_{xi}$ and $Q_{yi}$ are the $x$ and $y$ coordinates of control point $Q_i$. After multiplied by a Bernstein polynomial $B_i(t)$, the curve is determined by the weighted sum of the coordinates of the $m$ control points. Fig. 3 shows four curves parameterized by three to five control points. The curves always pass through the end control points and are tangent to the line between the first two and last two control points. When using three or four control points as shown in Figs. 3(a–b), a curve always lies within the convex polygon formed by the control points. It behaves smoothly without erratic oscillation. When the $x$ coordinate of $Q_1$ exceeds that of $Q_2$, the curve may form a near-cusp region or even a self-intersection. Fig. 3(c) shows one such case. To form a normal curve, a quick yet useful constraint is that the $x$ coordinate increment from $Q_1$ to $Q_2$ must be larger than a fixed value. More discussions on cusp and loop avoidance are found in Ref. [12, p. 176]. Using five or more control points enable more curve variation.

We consider a CTM that consists of four identical flexible arms. Taking advantage of symmetry, we only consider one quarter of the CTM, which consists of one flexible arm. Fig. 4(a) shows the sketch. The central rigid shaft is given a clockwise angular displacement and it generates a reaction torque $T$ in the opposite direction. The flexible arm connecting to the central shaft is divided into two segments. The initial un-deflected shape of each segment is characterized by using a four-point Bezier curve. Higher order Bezier curves may be used. The control points $Q_0 - Q_3$ and $Q_3 - Q_6$ determine the shapes of segment 1 and 2, respectively. A point $[x(t), y(t)]$ on the neutral axis of a segment is expressed as

$$[x_i(t), y_i(t)] = [B_0(t) B_1(t) B_2(t) B_3(t)] [Q_0, Q_1, Q_2, Q_3]^T ;$$

$$[x_i(t), y_i(t)] = [B_0(t) B_1(t) B_2(t) B_3(t)] [Q_3, Q_4, Q_5, Q_6]^T$$

where $B_0(t) = (1-t)^3$; $B_1(t) = 3(1-t)^2t$;

$$B_2(t) = 3(1-t)t^2 ; \quad B_3(t) = t^3.$$ (5)

The subscripts of $x(t)$ and $y(t)$ denote segment number. Each segment is further divided into $N = 10$ subdivisions and each subdivision is modeled as a straight beam element. Each beam has a rectangular cross-section with constant in-plane thickness $w$ and out-of-plane thickness $h$. The flexural rigidity of the beam is denoted as $EI$ where $E$ is the elastic modulus and $I = hw^3/12$ is the second moment of area.

Fig. 4(b) shows the model of a straight beam. The governing equations and numerical solution techniques of a large deflected beam are extensively found in the literature (e.g., Ref. [13]). When a straight beam deflects, the bending moment $M$ and maximal bending stress $\sigma_m$ are calculated by using

$$M = EI \frac{dy}{du} \quad \text{and} \quad \sigma_m = \frac{Ey}{2L} \frac{dy}{du} \quad (6, 7)$$

where $u \in [0,1]$ is a non-dimensional arc length along the beam neutral axis; the function $\psi(u)$ measures the beam rotation (or slope, in radians) along $u$. For a slender beam, the axial and shear stresses are very small and are not considered, when compared to the bending stress.

Since there are seven control points, the number of design variables is 14. Aimed at making the maximal stress as small as possible, formulation of the CTM shape design is detailed in Table 1. Besides the constraint function in Eq. (3b), there are more equality and inequality constraints in Table 1 to ensure numerical convergence and that the converged optimal shape is feasible. Specifically, constraint (b) confines the relative
positions between $Q_o$ and $Q_a$ in the x and y directions. To prevent interference of the flexible arm with the central shaft and outer rim, constraint (c) ensures that the lines tangent to $Q_o$ and $Q_a$ are normal to the outer rim and central shaft. Constraints (d) and (e) are used to limit the size of the arm and to avoid curve self-intersection that would cause stress concentration and manufacturing difficulties.

The deformation analysis required in the optimization process is carried out by using the generalized (multiple) shooting method (GMSM) \([13-14]\). This method is capable of accurate and efficient beam deflection computation. The optimization is realized by using fmincon() in MATLAB®.

Table 1 Formulation of CTM shape optimization

1. **Objective**: Given $\theta = 0-\theta$, minimize $|\sigma_a|$ at $\theta = \theta_3$

2. **Design variables**: $Q_0$, $Q_a$, $R$

3. **Constraints**:
   (a) Eq. (3b)
   (b) $Q_{0b} = r \cos \eta_1$ and $Q_{0b} = r \sin \eta_1$ where $30^\circ < \eta_1 < 60^\circ$ and $r = 0.8$ cm; $Q_{0} = [R 0]$ where $3r < R < 5r$
   (c) $Q_{0} = Q_0 \tan \eta_1$ and $Q_{0} = Q_0$
   (d) $Q_{i} = 1$ cm $< Q_{i} < 1.5$ cm;
   $Q_{i}: 1.5$ cm $< Q_{i} < 2.5$ cm; $-1.5$ cm $< Q_{i} < 1.5$ cm;
   $Q_{i}: Q_{0} = v \cos \eta_2$ and $Q_{0} = Q_{0} + v \sin \eta_2$ where $-90^\circ < \eta_2 < 90^\circ$ and $0 < \epsilon_i < 1.5$ cm
   $Q_{i}: Q_{0} = Q_{0} + v \cos \eta_2$ and $Q_{0} = Q_{0} + v \sin \eta_2$ where $1.5$ cm $< \epsilon_i < 2.5$ cm; $\epsilon_i > 1.5$ cm
   $Q_{i}: 2.5$ cm $< Q_{i} < 3.5$ cm
   (e) $Q_{i} - Q_{i-1} > 0.5$ cm for $i = 2, 5, 6$

3.3 Optimal shape of the constant-torque mechanism

Based on Table 1, we seek for the optimal shape and the corresponding relationship among $T$, $\theta$, and $\sigma_a$. Table 2 lists the simulation parameters. We use Polyoxymethylene as the material with yield stress $\sigma_y = 76$ MPa.

Table 2 Simulation parameters of the CTM

| $\theta_1$: 10°; $\theta_2$: 30°; $\theta_3$: 50° | $v$: 0.7 mm (In-plane thickness) |
| $E$: 2 GPa | $l$: 12 mm (Out-of-plane thickness) |

The optimized design variables are listed in Table 3. Using the values in Table 3, Fig. 5 shows the optimal positions of the control points and the corresponding arm shape. As can be seen, the active constraints are $\epsilon_2 = 1.5$ cm, $Q_{0b} = 3.5$ cm, and $R = 5r$. These active constraints may be relaxed to obtain a better objective value. However, this may come at the cost of larger arm footprint. The deflected shapes at $\theta = 25^\circ$ and $50^\circ$ are also plotted in Fig. 5. Fig. 6 shows the $T-\theta$ and $\sigma_m-\theta$ curves. After a 0.2575 Nm preload to reach $\theta = 10^\circ$, the $T-\theta$ curve stays within the 5% fluctuation range between $\theta = 10^\circ$–$30^\circ$. Besides, the fluctuation is still within the 5% until $\theta = 36^\circ$. In total, the constant-torque region is $26^\circ$, or 52% of the entire rotation. The maximal stress at $\theta = 50^\circ$ is 72.39 MPa, which is still below the yield stress. Note that the slope of the $\sigma_m-\theta$ curve suddenly increases at $\theta = 40^\circ$, where $\sigma_y = 51.31$ MPa. This is because the maximal stress changes location at this angle. Before $\theta = 40^\circ$, the maximal stress occurs within dotted circle A indicated in Fig. 5. After $\theta = 40^\circ$, the stress in dotted circle B is higher than that in circle A and becomes the place of the maximal stress. To avoid overact during operation, we will operate the CTM under $\theta = 40^\circ$ in the following experiments.

In Eq. (3b), only the flatness of the $T-\theta$ curve during $\theta_1-\theta_2$ is considered. Ideally it would be better to decrease the preload region in order to increase operational range. However, adding this mathematically into Eq. (3b) would cause divergence. This is expectable since the $T-\theta$ curve of typical elastic material is always continuous and cannot have a too sharp change of slope.

Table 3 Optimal configuration of the CTM

| $Q_0$: [0.6053 0.5231] | $Q_1$: [1.4986 1.2952] | $Q_2$: [2.25 0.473] |
| $Q_3$: [2.7324 0.3628] | $Q_4$: [3 0.8263] | $Q_5$: [3.5 0] |
| $Q_6$: [4 0] | $R$: 4 | Unit: cm |

3.4 Scaling of the $T-\theta$ curve

Since the flexible arms are identical, multiple arms may be placed at equally spaced positions to achieve a higher torque and structurally stronger design. This is shown in Fig. 7, where three, four, and five flexible arms are employed. The form of the $T-\theta$ curve is invariant regardless of the number of arms placed. For the three-arm CTM, its $T(\theta)$ would be 0.75 times that of the four-arm CTM. Similarly, the $T(\theta)$ function of the five-arm CTM would be 1.25 times that of the four-arm CTM.

Furthermore, the size and flexural rigidity may be changed to obtain a scaled $T(\theta)$ without altering the torque curve profile. Specifically, if the size is multiplied by $k_1$ and the flexural rigidity by $k_2$, then the new $T(\theta)$ and original $T(\theta)$ is related by

$$T_{new}(\theta) = k_2 / k_1^2 T_{ori}(\theta)$$

(8)

Enlarging the size requires multiplying the parameters in Table 3. Multiplying the flexural rigidity can be achieved by
changing the in-plane thickness \( w \), out-of-plane thickness \( t \), or material. Eq. (8) is helpful since the optimized CTM can be applied to various situations without redoing another optimization. Note that changing \( w \) would as well change the maximal stress curve; changing material would change the yield and allowable stress. These in return alter the allowable input angular displacement.

IV. EXPERIMENT VERIFICATIONS

4.1 Experiment of the \( T-\theta \) curve

The results in Sec. 3 are verified by comparing with those obtained by using ANSYS® and experiment. Fig. 8(a) shows a CTM prototype fabricated by using the parameters in Tables 2–3. Fig. 8(b) shows the experiment setup to measure the \( T-\theta \) curve. After mounting the outer rim of the CTM on a rigid bracket, the central shaft is given an angular displacement by using a motorized stage. The reaction torque at the central shaft is measured by a torque sensor from FUTEK (TFF425, maximum 1.2 Nm, 0.01 Nm resolution). Fig. 9 shows the experiment curve (with error bar) when the CTM is given a \( \theta = 40^\circ \) rotation, along with simulation results using the GMSM and ANSYS®.

Observing the curves in Fig. 9, the GMSM curve agrees well with the ANSYS curve (BEAM3). This verifies our computation methods. However, neither curve matches with the experimental curve. The experimental reaction torque has an average torque \( T_{ave} = 0.3036 \) Nm during \( \theta = 10^\circ-40^\circ \), which is 0.0306 Nm higher than the average torque predicted by simulations. The primary reason for this discrepancy is that both the GMSM and ANSYS® BEAM3 elements model beams as representative neutral axes. They do not consider the effect of beam thicknesses and corner fillets (due to the cutting tool diameter) at control points \( Q_b \) and \( Q_c \). More accurate results may be obtained by analyzing the CTM based on its true geometry. Although the experimental curve has a slightly larger constant torque than that predicted by the simulation curves in Fig. 9, the constant-torque ranges show very much resemblance. This implies that the CTM is insensitive to fabrication and modeling imperfections.

4.2 Experiment of the adjustable lever arm length

As indicated in Fig. 1(e), a CTM can be converted to an adjustable CFM through adjustable lever arm lengths. This enables adaptable force magnitude for different tissues. Fig. 10(a) shows the schematic of the lever arm. There are multiple holes for anchoring the cable. Fig. 10(b) shows the experiment setup to measure the cable force. A force sensor (FUTEK LSB200, maximal 22.27 N, 0.01 N resolution) measures the tension force of the cable. The cable direction is kept normal to the lever arm. The tension force is measured by connecting the cable at three different positions, namely \( a_1 = 3 \) cm, \( a_2 = 4.5 \) cm, \( a_3 = 6.0 \) cm. Each position is repeated three times. Fig. 11 shows the experiment result. As expected, the force remains nearly constant during \( \theta = 10^\circ-40^\circ \). The average cable forces are 8.9246 N, 6.1977 N, and 4.6161 N, respectively.

Comparing the curves in Fig. 9 and Fig. 11, the cable force in Fig. 11 is slightly smaller than that calculated from the reaction torque in Fig. 9. The reasons are that the cable (made of copper) extends and the lever arm experiences slight deflection. To match the results in Figs. 9 and 11, steel cables are suggested and the lever arm should be made stronger.

V. ADJUSTABLE CONSTANT-FORCE FORCEPS

Based on the designed CTM, a constant-force forceps is validated in this section. Fig. 12 shows the forceps prototype. The jaw mechanism is similar to that in Fig. 1(a) with initial (largest) angles denoted as \( \alpha_0 \) and \( \beta_0 \). Slider A is attached with a compression spring with stiffness \( k \) so that the jaws are open at \( \beta_0 \) with initial zero cable force. The cable with length \( \ell \) connects slider A to B through two guide pulleys. The coordinate of the center of the pulleys is denoted as \( [-b_0, a_0] \). The length from slider A to the center of the pulleys is denoted as \( s \) (initially \( s_0 \)). A stepper linear motor (Haydon, size 8) moves slider B along the slot where the extreme positions are denoted as \( a_{\max} \) and \( a_{\min} \). A rotary stepper motor (Minebea,
23LM-K) actuates the central shaft to provide the cable force. The motor first rotates the whole CTM to the cable taut position and then tautens the cable by deflecting the flexible arms. Detailed dimensions are listed in Table 4. Fig. 12 indicates that the CTM and cable form a closed loop where the sum of \( a \) (along the lever arm) and \( b \) (from slider B to pulley center) must be a constant vector. Also, the sum of the length variables \( b \) and \( s \) along the cable must be equal to the cable length \( \ell \). Joining these two conditions, we have the following. 

\[
a \cos \theta_a + b \cos \theta_b = -b_0; \quad a \sin \theta_a + b \sin \theta_b = a_0; \quad b + s = \ell 
\]

(9) 

Given length \( s \) and arm length \( a \), Eq. (9) can be numerically solved for \( \theta_a \), \( \theta_b \), and \( b \). The angle \( \gamma \) can then be obtained from \( \gamma = \pi - \theta_b + \theta_a \). To obtain the tip force \( F_e \), the forces acting on slider A have to be analyzed. The spring force \( F_s \) is a function of the jaw angle \( \beta \). With the help of Eqs. (1–2), we sum all the forces acting on slider A to derive the formula for \( F_e \).

\[
F_e = L_s \sin \left( \frac{a \theta_a}{L_s} \right) / \sin \gamma - F_s \right) / 2L_s \cos \frac{\theta_a}{2}
\]

where \( F_s = k L_s \left[ \cos \frac{\theta_a}{2} - \cos \frac{\theta_b}{2} \right] \) (10) 

At a particular position \( \beta \) and arm length \( a \), the tip force \( F_e \) can be obtained by using Eqs. (9–10). Consider the case where the torque is already in the constant-torque region, i.e., \( T(\theta) = T_{\text{ave}} = 0.273 \text{ Nm} \), we analyze the tip force by using the values in Table 4. Fig. 14 shows the effect of \( a \) and \( \beta \) on \( F_e \). It is shown that \( F_e \) primarily depends on \( a \). For each \( a \), the variation of \( F_e \) in \( \beta = 0^\circ \text{ to } 30^\circ \) is less than 2.5% of its average value. Hence the tip force can be kept constant regardless of the grasping position \( \beta \). The maximal and minimal \( F_e \) in Fig. 13 are 3.73 N and 1.23 N, respectively. This force range can be easily changed by scaling the \( T(\theta) \) curve. 

Fig. 14 shows the section view of the forceps. Two motors are required for this forceps. The linear motor adjusts the constant-force magnitude by changing the lever arm length.

Table 4 Detailed dimensions of the proposed forceps

<table>
<thead>
<tr>
<th>( L_s ) (mm)</th>
<th>( L_2 ) (mm)</th>
<th>( L_1 ) (mm)</th>
<th>( s_0 ) (mm)</th>
<th>( a_0 ) (mm)</th>
<th>( b_0 ) (mm)</th>
<th>( a_{\text{min}} ) (mm)</th>
<th>( a_{\text{max}} ) (mm)</th>
<th>( k ) (mm/mm)</th>
<th>( T ) (N)</th>
</tr>
</thead>
<tbody>
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<td>15</td>
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<td>30</td>
<td>15</td>
<td>45</td>
<td>0.5</td>
<td>6</td>
</tr>
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</table>

The rotary motor first tightens the cable, then drives the central shaft of the CTM to provide a constant torque. The forceps has a size of \( 143 \times 140 \text{ mm} \). During the actuation, the extreme values of \( \theta_a \) are 87.40° and 94.32°. Hence the range of cable taut position is 6.92°. As mentioned in Sec. 2, this range is for adapting various tissue sizes and stiffnesses. Yet, this range can be readily covered by the constant-torque region.

VI. CONCLUSIONS

This paper has presented the design and analysis of an adjustable constant-force forceps for robot-assisted surgical manipulation. The proposed forceps consists of a CTM with a lever arm so that constant forces of different magnitudes can be obtained. With the formulated design procedure, the optimized CTM exhibits a 52% constant torque range over the entire displacement. The CTM prototype has been verified by simulation and experiment. Without using sensing and control electronics, the proposed forceps has been shown to maintain a constant grasping force on tissues of various sizes and stiffnesses. The designed grasping force ranges from 1.23 to 3.73 N, which can be further adjusted by rescaling the \( T-\theta \) curve of the CTM. Our future work includes fabrication of the forceps prototype for tip force experiments. Extension to forceps of multiple DOFs will also be investigated. We expect the proposed forceps can serve as a reliable alternative to handle soft tissues in a MIS.

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