Abstract — A new variable stiffness actuator (VSA) is presented, designed for reproducing the full versatility of human motions from delicate tasks to precise positioning tasks through teleoperation, and highly dynamic tasks like hammering in particular. Existing VSAs show good performance in terms of either quick stiffness changing time, high output velocity, or variable damping, but not in the combination required for highly dynamic tasks. Goal of the presented design is to reach with one system a stiffness changing time of 50 ms, a peak output velocity of 20 rad/s and variable damping, based on the requirements of literature and our previous research. A prototype was built and its performance measured with three motors in parallel configuration: two responsible for changing the VSAs neutral position and effective stiffness through a lever arm mechanism, and the third acting as variable damper. Its effective stiffness can be changed continuously within 50 ms to 120 ms for small to large stiffness steps, its output velocity is up to 1100 rad/s and its oscillation behavior can be controlled precisely with the variable damper. Its effective stiffness range is 0.2 N m/rad to 313 N m/rad and its maximum continuous torque 9.4 N m. This unique combination makes the new actuator particularly suitable for highly dynamic tasks, while being also very versatile, and makes it especially interesting for teleoperation and human-robot collaboration.

Index Terms—Actuators, Telerobotics, User centered design

I. INTRODUCTION

HUMANS can perform motions with their arms within a wide range from very gentle motions like petting a cat to destructive motions like punching. In our research, we are interested to reproduce human motion through a robotic teleoperator with high fidelity, and in particular in highly dynamic motions on the example of hammering, as one extreme of the human motion bandwidth. Hammering will be useful in human-robotic exploration to cope with unexpected situations in unstructured environments, e.g., to clear the path in a disaster site inspection like the Fukushima Daiichi nuclear disaster site.

A robot suitable to reproduce human hammering efficiently needs to deploy high power efficiently in order to accelerate the high mass of the hammer head and needs to be able to absorb the high shock load at the impact of the hammer. Therefore, variable stiffness actuators (VSAs) with their mechanically adaptable compliance are currently the best candidate technology for building a telemanipulated robot appropriate for hammering. They can be used as rigid actuators for positioning tasks, as soft actuators for delicate tasks, and for highly dynamic tasks, as has been shown for various applications [1]–[10].

Garabini et al. have shown analytically that the peak output velocity achieved when hammering with a VSA can be increased by 30% if the stiffness is adapted during the hammering motion [11]. We have measured in our previous experiments that humans execute hammering motions at a frequency around 5 Hz and a peak velocity of 20 rad/s [12]. Therefore, a stiffness changing time under 50 ms is required to perform the hard-soft-hard-soft cycle described in [11] within a typical hammering period of 200 ms. Furthermore, a compliant actuator will strongly oscillate after a hammer strike, which is unpleasant for the human operator when the actuator is used for bilateral telemanipulation and can make the teleoperation system unstable. But adding constant damping to the actuator would undermine the benefit of the spring in the hammering, thus the actuator should have variable damping.

The actuator for our future research should thus provide the combination of fast stiffness changing time (50 ms), fast peak output velocity (20 rad/s) and variable damping. Some existing VSAs like the BAVS [13] and the FAS [14] from DLR allow a stiffness change within these times, but BAVS only reaches an angular velocity of 12.6 rad/s, and FAS is too small as it was designed to actuate a finger tendon, and neither has variable damping.

Goal of this paper is to design a dynamic robotic actuator (Dyrac) with these characteristics able to perform telemanipulated hammering. While the reference task for this design is the hammering task, the actuator should also be useful for other tasks like precise positioning tasks and delicate tasks, and therefore have an output torque and stiffness range comparable to other VSAs.

II. DYRAC DESIGN

A. Requirements

Table II lists the design requirements and compares them to the performance of selected existing VSA designs. The performance requirements for the design were derived from the data obtained from the teleoperated hammering experiments [12]. The primary design goal was to match these requirements, and the secondary design goal was to develop a very versatile actuator, allowing to explore the parameter space velocity – stiffness – damping extensively. Existing actuator designs with similar performance as the requirements are among others: the Bidirectional Antagonistic Variable Stiffness joint (BAVS) [13], the Floating Spring Joint (FSJ) [15], the Mechanically

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Adjustable Compliance and Controllable Equilibrium Position Actuator (MACCEPA) [16], the Actuator with Adjustable Stiffness II (AwAS-II) [17], and the Variable Stiffness Actuator University Twente II (vsuUT-II) [18].

B. Kinematics

The architecture chosen for Dyrac was a parallel motor configuration design with moving pivot point lever arm stiffness variation mechanism, similar to the vsuUT-II.

Fig. [1] shows the name conventions used in the following. The angular polar coordinate of the pivot point \( P \) will be referred to as pivot angle \( \varphi \) and the radial polar coordinate as pivot radius \( r \). Fig. [2] and [3] show the kinematic principle of the actuator and the three basic operations: change of neutral position, change of stiffness, and deflection.

The neutral position is changed by equal rotation of motor 1 and 2. This causes a change of the pivot angle \( \varphi \) while keeping the pivot radius \( r \) constant. The stiffness is changed by rotation of motor 2 relative to motor 1. This causes a change of the angle \( \angle POB = \varphi_d \). Thus, in the triangle \( POB \) defined by the two constant lengths \( OB = r_D \) and \( BP = b \) and the angle \( \angle POB = \varphi_d \), the length \( OP = r \) has to change. This changes the effective stiffness of the spring by changing the position of the lever pivot point \( P \). Indeed, if \( P \) is on one axis with the output axis going through \( O \) for \( r = 0 \), the output axis can rotate freely around the pivot axis and the spring has no effect, which corresponds to an effective stiffness of zero. On the other hand, if \( P \) is on one axis with the spring rotation axis going through \( A \) for \( r = l \), then the output axis is locked to the motor 1 axis and the effective stiffness of the spring is infinite. For any other values of \( r \) between these two extremes, the effective stiffness has a certain finite value, with the special case of \( r = l/2 \), for which the effective stiffness equals the nominal stiffness of the spring. Equations [1] to [16] describe the relationships to calculate the output torque \( T_O \) (eq. [10]) and the torques \( T_1 \) and \( T_2 \) of motor 1 and 2 (eq. [15] and [16]) depending on the design parameters \( r_D, b \) and \( l \), the spring stiffness \( k \), the stiffness setting angle \( \varphi_d \), and the deflection \( \alpha \) for a static equilibrium of forces.

Some helping variables \( \beta, \gamma, \) and \( \delta \) can easily be obtained from the basic trigonometric relations:

\[
\beta = \frac{\pi}{2} - \alpha - \theta, \tag{1}
\]
\[
\gamma = \frac{\pi}{2} - \delta, \tag{2}
\]
and
\[
\delta = \pi - \varphi_d - \epsilon. \tag{3}
\]

From the law of sines, one obtains \( \epsilon \) and \( r \):

\[
\epsilon = \arcsin \left( \frac{r_D}{b} \sin \varphi_d \right), \tag{4}
\]

![Fig. 1. Geometric relations of the kinematics (view from the left compared to Fig. 2). The neutral position \( \varphi \) is controlled by motor 1 (purple). The angular position \( \eta \) of the output axis (orange) is the sum of neutral position \( \varphi \) and the deflection angle \( \alpha \). The triangle \( OBP \) is defined by the lengths \( b \) and \( r_D \) and the angle \( \varphi_d \). Changing \( \varphi_d \) by moving motor 2 (green) relative to motor 1 will modify the pivot radius \( r \). The triangle \( OPA \) is defined by the lengths \( r \) and \( l \) and the angle \( \alpha \). Changing \( \alpha \) by deflecting the output axis (orange) compared to the neutral position will modify \( \theta \), which is the spring deflection angle, and result in a counter-force from the spring deflection. The other measures in the figure are indicated to help obtain equations [4] to [10].](image-url)
and

\[ r = b \frac{\sin \delta}{\sin \varphi_d}. \]  

Using Pythagoras’ theorem in the triangle PAC allows to calculate \( c \) as

\[ c = \sqrt{(l \cos \alpha - r)^2 + l^2 \sin^2 \alpha}, \]  

which helps to calculate \( \theta \) using the law of cosines:

\[ \theta = \arccos \frac{c^2 + l^2 - r^2}{2 cl}. \]

With \( \theta \), the torque \( T_A \) produced by the spring is easily obtained from Hooke’s law applied to rotational springs as

\[ T_A = k \theta, \]

which gives the force of the spring in the point \( P \) by division through the lever length \( c \):

\[ F_P = \frac{T_A}{c}. \]

This needs to be projected into the tangential direction and multiplied with the lever arm to obtain the output torque:

\[ T_O = \sin \beta \frac{k \theta}{c} r. \]  

The effective stiffness \( k_e \) can be defined from this equation as

\[ k_e = \sin \beta \frac{k \theta r}{c \alpha}, \]

such that \( T_O = k_e \alpha \).

Further, the motor torques \( T_1 \) and \( T_2 \) can be obtained by projection of the forces and the relation between torque and force:

\[ F_D = \frac{\cos \beta F_P}{\cos \epsilon}, \]

\[ F_{T_1} = \sin \beta F_P + \text{sign}(\alpha) \sin \epsilon F_D, \]

and

\[ F_{T_2} = \frac{F_D}{\cos \gamma}. \]
Fig. 4. Semi-logarithmic plot for the proposed actuator kinematics of the effective stiffness ratio, defined as effective stiffness $k_e$ divided by spring stiffness $k$, over the pivot radius ratio, defined as pivot radius $r$ divided by lever length $l$. The plot is cropped as a pivot radius ratio of zero yields zero effective stiffness ratio and pivot radius ratio of 1 yields infinite effective stiffness ratio.

$$T_1 = \left( \sin \beta + \text{sign}(\alpha) \sin \frac{\cos \beta}{\cos \epsilon} \right) \frac{k \theta}{c} r,$$

(15)

and

$$T_2 = \frac{\cos \beta}{c \cos \epsilon \cos \gamma} k \theta r_D,$$

(16)

Fig. 4 shows a semi-logarithmic plot of the resulting effective stiffness ratio over the pivot radius ratio, with the stiffness ratio defined as effective stiffness $k_e$ over nominal spring stiffness $k$, and the pivot radius ratio defined as pivot radius $r$ over lever length $l$. For the pivot in the center position (pivot radius ratio of 0.5), the effective stiffness equals the nominal spring stiffness (stiffness ratio of 1). In practice, a pivot radius of zero is not useful to implement with a slider-crank linkage, as it would result in a dead-lock position from which one cannot recover by changing the relative position of stiffness changer and positioning motors. In the final design, the design parameters were set to $r_D = 10$ mm $b = 9.5$ mm, and $l = 20$ mm, resulting in an effective stiffness ratio range from 0.0006 to 1500.

C. Component Selection and Assembly

Although the kinematic principle of the actuator is simple, its realization turned out to be rather complex. If the reachable pivot radius should be as small as possible, the pivot axis has to be very close to the shared rotation axis of the motors. Thus, a hollow shaft approach was adopted, meaning that all bearings and driving components were placed on the outside of the actuator, leaving the center region for the pivot mechanism (cf. the exploded view in Fig. 5). This was achieved by using large diameter thin section bearings (INA CSEA030) and frameless large diameter thin section motors (ThinGap TG5151). The motors were integrated in direct drive configuration as a concession to not add further complexity by the necessity to also integrate gears.

The final design therefore consists of three disc-shaped modules for the positioning motor, the stiffness changer motor and the damper, which are mounted on a two-parted chassis. For position sensing, large diameter magnetic ring absolute position encoders were mounted directly to the respective discs (RLS AksIM). A CAD model of the actuator is provided as Solidworks files in the supplementary material [19].

D. Spring Design

A monolithic part consisting of a rigid cantilever section and an elastic torsion spring section was designed to act as elastic element, as shown in Fig. 6. A nominal stiffness of $k = 60 \text{ N m/rad}$ was chosen as design goal for the spring, resulting in a theoretical effective stiffness range of 0.06 N m/rad to 41 000 N m/rad. Considering the selected motors’ maximum torque of 12 N m, this corresponds to a maximum spring deflection of 15.1° and a spring torque of 15.8 N m.

Titanium grade 5 (Ti6Al4V) was selected as the spring material because of its capacity to store high potential energy per volume elastically before being affected by plastic deformation. Several torsion spring shapes were considered. Because of spatial constraints, the resulting spring has only one winding. The diameter and thickness of the spring were determined to fit in the design when deformed, and the spring
thickness profile was dimensioned as following, such that those values are achieved without exceeding material limits. First, a finite element simulation was performed in ANSYS Workbench 18 on a spring model with a uniform thickness of 5 mm and a load of 15.8 N m. The simulation result indicated a non-uniform stress distribution over the circumference of the spring. The spring winding thickness was then altered iteratively until an even stress distribution was achieved. By doing so, the stress is uniform along the outer edge of the spring. Second, the spring shape and thickness was optimized further. The height, width and thickness were changed iteratively until the spring had the desired deflection at maximum load (15.1°), the maximum stress did not exceed the titanium grade 5 fatigue strength (450 MPa), and the spring did not touch any surrounding parts at maximum deformation. Since the spring is asymmetric, these conditions were checked in both directions. The spring was manufactured through wire electric discharge machining (EDM) and CNC milling.

E. Variable Damper

A third motor is acting as a variable damper with the stator mounted to the chassis and the rotor mounted to the output axis. The motor torque \( T_3 \) is controlled to mimic the behavior of a variable damper with damping factor \( b \) between motor 1 and output axis:

\[
T_3 = b (\dot{\varphi} - \dot{\eta}) .
\]

F. Controller

The motors are controlled by three motor drivers (Ingenia JUP-40/80-E) communicating via EtherCAT with a real-time Linux based high-level controller. The EtherCAT loop runs at 1 kHz with the drivers of motor 1 and 2 configured in position control mode and the driver of motor 3 configured in torque control mode. The internal current controllers of the drivers run at 10 kHz.

III. System Performance

In the following, we report the performance measures in the way suggested for VSAs by Grioli et al. A column was added to the mechanical performance summary to indicate whether values were obtained by calculation from the design or by measurement on the prototype. Table II shows the mechanical performance measures, as a combination of values calculated from the manufacturer data sheets and measured values. For torque and power, the values are different in the positive and negative direction, as motor 1 and 2 either work together or against each other (cf. Fig. 4). Fig. 7 shows the measured output torque over output deflection plots for different pivot radii, showing the output torque hysteresis. Fig. 8 shows the measured effective stiffness over torque plots for different pivot radii with the effective stiffness calculated as output torque over output deflection. Fig. 9 shows the measured output torque over output deflection plots for different pivot radii. Fig. 10 to 12 share the same measurement data collected with motor 1 fixed and an external torque applied at the output. The torque was measured with an ATI Gamma force-torque sensor mounted to the output axis (calibrated for ± 10 N m torque measurement in z-axis). For Fig. 8 and 9 the data points were reduced for better visualization by calculating the center line of the hysteresis and applying the Douglas-Peucker line simplification algorithm (Matlab function reduce). The dashed lines indicate the theoretical values for reference. Fig. 11 shows the measured pivot radius in response to a stiffness set point step, once for a small step and for a large step. The 90% step response time is 50 ms for a small step and 120 ms for a large step.

IV. Application Example

As a simple application example, the actuator was programmed to execute a fast back and forth swinging motion similar to hammering at approximately 3.1 Hz with a cylindrical brass load of 1.6 kg at the output of the actuator. Together with the torque sensors, this resulted in an inertial load of approximately 0.0125 kg m². The actuator was programmed to execute the motion once at a high stiffness setting with a pivot radius of 19.1 mm and

### Table II: Actuator Mechanical Performance Summary

| No. | Measure Defined | Unit | Value Calculated | Value Measured |
|-----|-----------------|------|------------------|----------------|
| 1   | Cont. Output Power (pos.) | W | 1085 |                  |
| 2   | Nominal Torque (pos.) | N m | 7 |                  |
| 3   | Nominal Speed | rad/s | 155 |                  |
| 4   | Nominal Stiffness Variation | s | 0.09 |                  |
| 5   | Nominal Stiffness Variation | s | 0.12 |                  |
| 6   | Peak Torque (pos.) | N m | 23 |                  |
| 7   | Maximum Speed | rad/s | 1100 |                  |
| 8   | Maximum Stiffness | N m/rad | 41 000 | 313 |
| 9   | Minimum Stiffness | N m/rad | 0.06 | 0.2 |
| 10  | Maximum Damping | N m/s/rad | 2 | 2 |
| 11  | Maximum Elastic Energy | J | 5.7 | 0.7 |
| 12  | Maximum Torque Hysteresis | % | 40 |                  |
| 13  | Maximum Deflection with max. stiffness | ° | 120 |                  |
| 14  | Active Rotation Angle | ° | ∞ |                  |
| 15  | Angular Resolution | ° | 6.9 x 10⁻⁴ |                  |
| 16  | Weight | kg | 10.3 |                  |

Fig. 7. Measured torque over deflection hysteresis plot for different pivot radius settings.
once with changing the stiffness to a low setting with a pivot radius of 6.9 mm at the beginning of the motion. In both cases, the variable damping setting was set to a low value at the beginning of the motion (0.01 N m s/rad) and triggered to a high value (0.5 N m s/rad) at detection of a negative output position $\eta_{\text{actual}}$ to simulate an impact.

Fig. 10 shows plots over time of the position, velocity, stiffness and damping data that were measured during execution of the motion (3.3 s of data were cropped at time 0.5 s, corresponding to a return to initial conditions and waiting for manual start of the next run). At high stiffness ($r_{\text{set}} = 19.1$ mm), the deflection is lower than 0.03 rad allowing precise motions, whereas at low stiffness ($r_{\text{set}} = 6.9$ mm) the deflection is up to 0.3 rad, helping to reach a higher output velocity than with the stiff setting. The velocity gain obtained with low stiffness is 175% from input velocity $\dot{\varphi}$ to output velocity $\dot{\eta}$. The stiffness changing time measured was 58 ms for this stiffness change.

V. DISCUSSION

In the analysis of whether the design meets the requirements of the tasks, we differentiate between the design and the prototype implementation, which was slightly simplified for budgetary reasons. Most notably, the prototype was implemented without gears for motors 1 and 2, meaning that the position has to be held by the motors in stall operation. This is not an efficient operation mode for electric motors, and negatively affects the positioning performance of the actuator in the presence of external disturbances. However, this was deemed acceptable, as the main mode of operation aimed at with this actuator are highly dynamic tasks. For delicate tasks and highly dynamic tasks, a direct drive configuration is deemed acceptable, as the main mode of operation aimed in the presence of external disturbances. However, this was

The requirement of 50 ms was only met by the prototype for small stiffness changes, whereas the time measured for large
stiffness changes was up to 120 ms. We think that this is not an issue in our targeted bilateral hammering application, because it only requires small stiffness changes as the maximum human arm stiffness was measured to be in the range of 40 N/m/rad [21].

Also, while the calculated maximum stiffness of the design was 41,000 N/m/rad, a maximum stiffness of only 313 N/m/rad was measured. This difference is most likely due to play and material deformations in the force transmission. The calculations are based on the assumption that only the spring deforms, while that cannot be guaranteed for this prototype. However, the maximum actuator stiffness is still sufficiently high compared to the maximum human arm stiffness of 40 N/m/rad, and therefore the use of Dyrac for bilateral teleoperation is not limited.

Play is also an issue of the prototype’s performance for explosive movements, as it induces a high torque hysteresis (cf. Fig.7), resulting in energy losses in resonance mode. This can explain the lower velocity gain of 175 % in the example application compared to results measured with other VSAs of 272 % [22] and the optimum velocity gain according to analytic calculations of 400 % [11].

Another effect of the design is that for equal motor torques, the output torque is higher in the negative direction than the positive direction (9.4 Nm to 7 Nm) because the respective torques of the position changer motor and the stiffness changer motor add up in the negative direction, whereas they subtract in the positive direction. This is a direct consequence of the slider-crank linkage mechanism. While this is in general undesirable, it is in many practical applications not an issue as there is often a working direction requiring a large force and a disengaging direction requiring much less force. Moreover, the configuration can be changed on-the-fly by setting $\phi_d$ to a negative value to swap the stronger and the weaker direction.

Mechanical end stops were added at ±120° deflection in the prototype to avoid situations that cannot be recovered from by the motors alone. This is mainly due to the slider-crank linkage mechanism leading to very unfavorable configurations for low stiffness settings. Indeed, for a low stiffness setting $\phi_d$ has to be large, thus a higher torque in motor 2 is required to maintain the equilibrium of forces in Fig.1 (the cosine terms in the denominator of equation [16] tend to zero for small pivot radii).

The actuator weight was not optimized for this prototype, leading to a heavy actuator (10.3 kg) compared to existing variable stiffness actuators with similar performance, like the FJS or the AwAS-II (both 1.4 kg). This could be improved in a future design iteration.

The application example shows that the actuator is suitable for precision tasks as well as highly dynamic tasks and that it can dynamically adapt the stiffness as needed. Also the variable damping is very useful to suppress unwanted oscillations without affecting the desired ones.

These are very promising results, as this actuator will allow performing also human-like dynamic motions such as hammering, shaking, jolting and throwing intuitively and efficiently through teleoperation additionally to delicate tasks and precision tasks.

VI. CONCLUSION

A new variable stiffness actuator design was presented having some attractive performance features for dynamic actuation research: (1) fast stiffness changing time of 120 ms for full-range steps (50 ms for small steps), (2) a wide effective stiffness range of 0.2 N/m/rad to 313 N/m/rad, and (3) variable damping.

A first prototype of the new design was manufactured and characterized to show the feasibility and validating the performance requirements in hardware, albeit with small limitations. The benefits of this actuator for teleoperated applications have successfully been demonstrated in an application example, showing that the actuator is suitable for both precise positioning tasks and highly dynamic tasks like hammering.

VII. ACKNOWLEDGEMENTS

This project was supported by the Dutch Research Council (NWO) under the project grant 12161.

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