

DESIGN AND DEVELOPMENT OF TORSION SPRING BASED SERIES ELASTIC ACTUATOR WITH NESTED ENCODERS FOR WEARABLE EXOSKELETON ROBOT

A Thesis

by

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To my family

ABSTRACT

This thesis proposes a series elastic actuator design to power an assistive exoskeleton for the paraplegic or stroke patient whom incapable to walk or provide any motion at lower extremity portion of the body. Commonly, active exoskeleton robots are powered with the rigid actuators which do not provide torque controllability any physiological compliance and physical elasticity with the patient.

Physical elasticity in the actuator can be used to compensate the environmental external forces during motion of the exoskeleton. Therefore, controlling the torque generated at the joints of the exoskeleton would provide safer human-robot interaction. Series elastic actuators are able to provide advanced control and high fidelity control with the elastic element located between the motor and the mechanical output of the series elastic actuator. The main goal of this thesis is to develop a series elastic actuator for a powered exoskeleton to provide torque measurement and control during locomotion and optimizing the elastic element of the actuator for high fidelity control.

To provide torque controllability a custom torsional spring has been produced and placed at the output side of the series elastic actuator. Especially developed SEA modules are designed to be modular, lightweight and have a high torque-to-mass ratio. In addition, measurement of the angular displacement of the elastic element is challenging in terms of mechanical design. To circumvent this design problem, a double shaft mechanism was proposed. In this mechanism, the first shaft, which connects the spring and the spring encoder, goes through the second shaft, which is connected to the motor and the motor encoder. This way both encoders are placed on the same side of the SEA. In addition to design, results of the cascaded proportional integral derivative (PID) controller with a disturbance observer (DoB) applied on the actuator is presented.

ÖZETÇE

Bu tez, yürüyemeyen veya vücudun alt ekstremitte kısmında herhangi bir hareket sağlayamayan felçli veya felçli hasta için güçlendirilmiş bir dış iskelete güç sağlamak için bir dizi elastik aktüatör tasarımı önermektedir. Genellikle, aktif dış iskelet robotları, sert aktüatörlerle çalıştırılır. Bu nedenle rijit aktüatörler tork kontrol edilebilirliği hasta ile herhangi bir fizyolojik uyum ve fiziksel elastikiyet sağlamaz.

Eyleyicideki fiziksel esneklik, dış iskeletin hareketi sırasında çevresel dış kuvvetleri telafi edebilir. Bu nedenle, dış iskeletin eklemlerinde oluşan torkun kontrol edilmesi, güvenli insan-robot etkileşimi sağlar. Seri elastik aktüatörler, motor ve seri elastik aktüatörün mekanik çıkışı arasında bulunan elastik eleman ile gelişmiş kontrol ve yüksek sadakat kontrolü sağlayabilmektedir. Bu tez araştırmasının ana amacı, hareket sırasında tork ölçümü ve kontrolü sağlamak için güçlendirilmiş bir dış iskelet için bir seri elastik aktüatör geliştirmek ve yüksek sadakat kontrolü için Seri Elastik Aktüatör ile donatılmış elastik elemanı optimize etmektir.

Tork kontrol edilebilirliği sağlamak için özel bir burulma yayı üretilmiş ve seri elastik aktüatörün çıkış tarafına yerleştirilmiştir. Özellikle geliştirilen SEA modülleri, modüler, hafif ve yüksek tork-kütle oranına sahip olacak şekilde tasarlanmıştır. Ayrıca, bu elastik elemanın açısal yer değiştirmesinin ölçümü mekanik tasarım açısından zordur. Bu tasarım sorununu önlemek için çift şaftlı bir mekanizma önerildi. Bu mekanizmada yayı ve yay kodlayıcıyı birbirine bağlayan birinci mil, motora ve motor enkoderine bağlı olan ikinci milden geçer. Bu şekilde her iki kodlayıcı da SEA'nın aynı tarafına yerleştirildi. Ayrıca, aktüatöre uygulanan bir bozulma gözlemcisi (DoB) ile kademeli PID kontrolörünün sonuçları da bu tezde sunulmaktadır.

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CHAPTER I

INTRODUCTION

Active exoskeleton robots are a trending research area to rehabilitate the walking capabilities of paraplegic patients [6, 7]. These patients can wear these robotic systems and able to move their lower extremities according to exoskeleton joint movements [8, 9]. However, the most common type of actuator that has been used within the design of the powered exoskeletons is rigid actuators [9, 10]. Furthermore, the rigid actuators are incapable of torque control and their suitability for physical human-robot interaction in terms of safety and physiological fitness with the patient [11].

The existence of torque control and physical elasticity properties of an assistive device is beneficial for the systems that consist of human-robot interaction [12, 13]. The designed exoskeleton is no longer used for experiments but also for the rehabilitation of patients which requires human-robot interaction during the motion [14]. Furthermore, the physical elasticity of rehabilitation robots compensates for the reaction forces resulting from the environment [15]. The resulting external forces caused by the uncontrolled torque of the joint cause uncomfortable robot-human interaction and in the worse cases position-controlled actuators result in injuries to the patients [2]. To provide safe, comfortable, and effective robot-user interaction the series elastic actuators (SEA) are used in a vast area of legged robotics and human assistive devices [16].

The SEA consists of a brushless DC motor, a spring mechanism, Harmonic Drive gear, three high precision position encoders for the motor, and an elastic element that measures torque [17]. The working principle of the SEA initiates with the rotation of the rotor then the power is transmitted to the regulator which reduces the speed of the rotor and increases the torque transmitted to the

elastic element and output such as end-effectors or exoskeleton joints.

Furthermore, a directly driven exoskeleton with a simple actuation system such as DC motor, and regulator assembly may result in torque spikes during the swing phase or sit-to-stand phases. In addition, the design of the spring is essential to provide tolerance to impact loads related to joint movements during locomotion of the exoskeleton also providing low mechanical output impedance and passive energy storage during the control of the torque of the joint [18]. Moreover, the high peak torque values overcome the friction of the actuator and provide a smooth motion during the locomotion of the assistive device. In addition, various type of series elastic actuators exists in the area such as prismatic series elastic actuators [19], the linear series elastic actuator for hydraulic systems [20].

To provide a lightweight and modular actuator that can be used on an exoskeleton, Lagoda et al. used a harmonic gear set to reduce the higher motor speed and additionally used a torsional spring structure that incorporates linear springs to render a high torque control bandwidth limit [1]. The mechanical structure of the SEA can be seen in 1 There are also custom-designed torsional springs and planet gears to increase the efficiency of SEA [2]. The designed series elastic actuator assembly and its test bench can be seen in 2 In another design, the position of the DC motor is measured with the optical incremental encoder located behind the SEA and the deflection of the spring was measured with a magnetic incremental encoder located inside the SEA to make the design more compact at the actuator's output side [21]. The main design challenge in SEAs is to make them compact and lightweight while ensuring the delivery of high torque output. Placing sensitive and large size sensors in the SEA complicates the design. Accordingly, to solve the measurement problem of the deflection of the torsion spring while keeping the dimensions compact, a double shaft SEA design is proposed in this article.

In the proposed design, two absolute encoders are used to measure the spring deflection and angular position of the motor shaft. A compact design is achieved by placing the encoders on the same side of the SEA. A special casing is designed

to attach the torsional spring. This casing is used to transmit the deflection of spring to the secondary encoder shaft which goes through the hollow motor shaft.

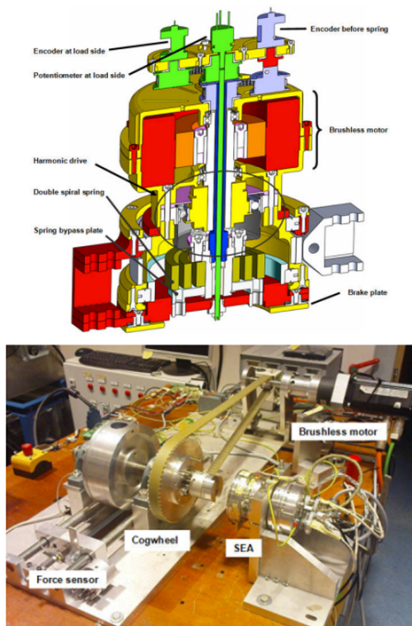


Figure 1: Design of a Series Elastic Actuator with Magnetic Encoders [1]

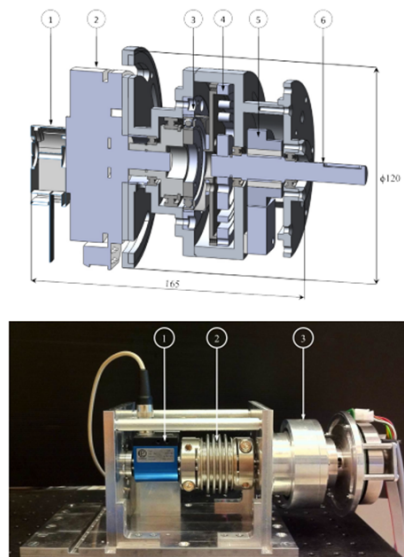


Figure 2: Design of a Series Elastic Actuator with Optical Encoder [2]

1.1 Scope of the Thesis

This study focuses on two main topics: i) development of a SEA, ii) design and optimization of elastic element for a series elastic actuator, and iii) torque control of the actuator. In this section, the literature on these topics is reviewed distinctly and provided in Section 1.1.1 and Section 1.1.2 respectively for SEA and Control Methods for SEA.

1.1.1 Series Elastic Actuators

Non-elastic actuators are ideal for high force and position control. However, the system includes small displacements and high forces the force control becomes difficult. Furthermore, the position measurements errors resulted with the errors during high forces. Therefore, to obtain accurate measurement the resolution of the sensors must be high [22].

Standard robot actuator system cannot provide precise forces acting on the robot joints. This problem caused by the friction, stiction and backlash occurred at the actuators [23]. Precise force control is needed for positioning and actuation of human joints. To attain the force control a force bandwidth, mechanical output impedance and dynamic range should be measured. To provide a high fidelity force control a trade-off have been made and elasticity of the system becomes more dominant [24]. SEAs are widely studied to present advanced control capabilities also consist of elastic elements which let the synthesis of high fidelity torque control. These elastic elements are located between the motor and mechanical output [25]. Since the SEAs are capable of torque control they contribute physical elasticity of the rehabilitation device and they preferably used in legged robots [26, 4]. The assembly of the rehabilitation robot with a human subject is given in 3 In addition, a humanoid robot designed by NASA's Johnson Space Center in collaboration with several external partners which is named Valkyrie. It has many degrees of freedom and to provide torque controllable motion and safe human - robot environment they were used series elastic actuators and used decentralized

approaches to control series elastic actuators of the Valkyrie [3]. The humanoid robot Valkyrie is shown in 4. Moreover, a tendon driven robotic arm designed and to provide a human robot interaction series elastic actuators has been used in order to manipulate the links of the robotic arm with the torque controllability aspect of the SEA [5]. The final version of the tendon driven robotic arm can be seen in 5

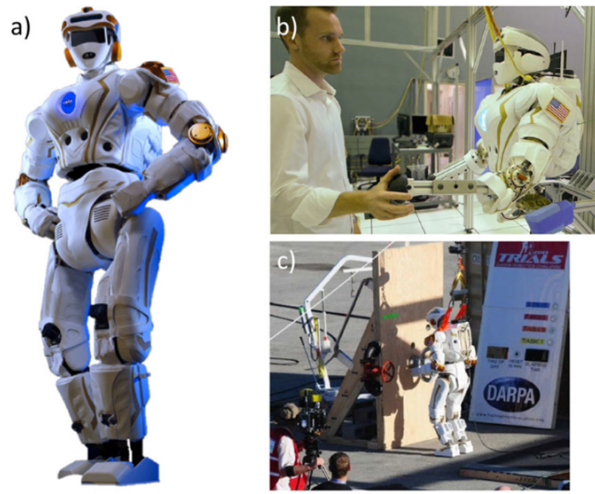


Figure 3: Design of a Humanoid Robot Valkyrie a), Human - Robot interaction of Valkyrie b), Walking Tests of Valkyrie (c) [3]



Figure 4: Design of a Rehabilitation Robot with Human Subject [4]

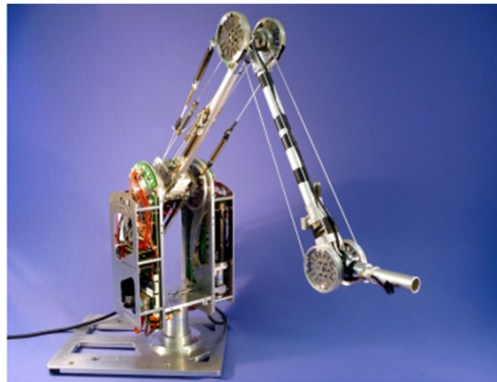


Figure 5: Design of a Tendon Driven Robotic Arm with SEA [5]

SEAs designs may appear in different forms [17]. For instance, Lagoda et al. considered harmonic drive as regulator of the SEA and designed a planar torsional spring to generate high torque control bandwidth limit also, aimed to provide

light and modular device that can be used on exoskeleton [1]. Oh and Kong et al. developed a worm gear mechanism as reducer to minimize the systems impedance [27]. Taylor et al located the elastic element at the reduction phase to decrease the torque requirement of the system [28]. Accoto et al. devised custom torsional spring and planetary gear to increase the efficiency of the SEA for lower limb exoskeleton [2].

During the design of the series elastic actuators these parameters should be considered; power output, volumetric dimension, weight, efficiency, back drivability, backlash, and passive energy storage.

1.1.2 Control Methods for Series Elastic Actuators

Series elastic actuators consist of torque control capability rather than rigid actuators. However, the complex structure of the series elastic actuator resulted in complex control methods. SEAs are modeled to be two inertia systems, considering the motor side and the load side or the link side of the legged robot [29]. The interaction of the motor side and the load side is provided by the elastic element of the SEA. The transmitted load causes the spring to deflect and the deflection can be measured by rotary encoders which can be converted in terms of torque. The force control for the rehabilitation robots is separated into two levels. For the low-level controllers, the force control mainly depends on human motion which requires fast computation and immediate reaction. Therefore, SEAs require advanced control methods instead of simple methods.

To attain torque tracking, earlier research proposed a simple torque control method for SEAs such as conventional Proportional-Integral-Derivative (PID) control. Yet, the torque control performance was decreased inadequate, In addition, the feed-forward (FF) term is added to the model to negate the effects of unmodeled dynamics [28]. For elastic joints, a similar approach has been followed with a PD and a feed-forward control method was used. However, the overall performance of the system decayed due to nonlinear disturbances [30]. Hence, the conventional methods do not indicate robust torque tracking for SEAs.

To propose a robust and stable control and minimize the nonlinearities of the SEA such as stiction generated by the DC motor and the backlash caused by the gearbox of the SEA a cascaded control algorithm consists of an inner loop velocity control and outer loop torque control method has been used [31]. Utilizing inner loop velocity control can compensate for the internal and external disturbances that occurred in the system with cascaded control method [32].

1.2 Motivation

The literature review indicates the problem of the current rehabilitation robots research field. Two main problems are i) the trade-off between torque measurement and position control bandwidth and ii) human-robot interaction. On the other hand, the literature explains and provides an engineering approach to point out several solutions to these problems; A series elastic actuator with an elastic element can measure the torque hence the soft springs limit the position control bandwidth which becomes more dominant on the dynamic system. In addition, the modularity and control method of the series elastic actuator should track the torque with high fidelity. By considering these two problems, a series elastic actuator with a stiffer spring is considered. By having two solutions in mind:

i) Optimization of Torsional Spring: Conducting a structural and deflection analysis via simulation software and experimental test benches a stiff torsional spring will be produced to measure the torque at the link side of the series elastic actuator.

ii) Robust Control of SEA: The designed series elastic actuator will power the robot which requires high fidelity torque control to provide safe and physiologically consistent human-robot interaction.

From this motivation, a modular series elastic actuator with a custom torsional spring which has a maximum deflection of 2 degrees has been determined.

Furthermore, the deflection of the spring is also tuned through conducted simulations. The designed SEA's control method has been determined as Cascaded

PID to track the input torque given to the system to fulfill the requirement of the minimum tracking error. Design and development approach of the SEA unit and design process of the torsional spring see Chapter 2. In addition, the simulation results and the application of simulation were presented in Chapter 3, the torque control strategy of SEA is decided in Chapter 4, experimental results were mentioned in Chapter 5, and conclusion is presented in Chapter 6.



CHAPTER II

MECHANICAL DESIGN OF SERIES ELASTIC ACTUATOR

The realized SEA design is produced with a long-term and iterative design process and with many trials and errors. In this chapter, initial work done on the SEA design, development of the elastic element, and final results with the implemented controllers are shared respectively through Sections 2.1 to 2.3.

2.1 Mechanical Hardware of SEA

For the wearable exoskeleton robot, two SEA units with different torque outputs were designed. Series elastic actuators are an assembly of DC motor unit and gear box unit with the elastic element placed at the output of the SEA. The reduced speed of the DC Motor transmitted as high torque at the output of the gear box unit the measured deflection of the elastic element provides torque controllability by feeding it as torque at the link side of the SEA. The basic representation of SEA is illustrated in Figure 6.

These designed SEA units consist of four main components: i) a frameless brushless DC motor (Kollmorgen TBMs-7615 or 7631), ii) a strain wave generator (Harmonic Drive (HD) CSG-25 1:100 gear ratio), iii) a torsional spring and double shaft mechanism which is used as a torque sensor [21], and iv) 21-bit absolute encoders (FENAC AS40HB3K). The The boundary conditions of the design of the SEA is given in Table 1.

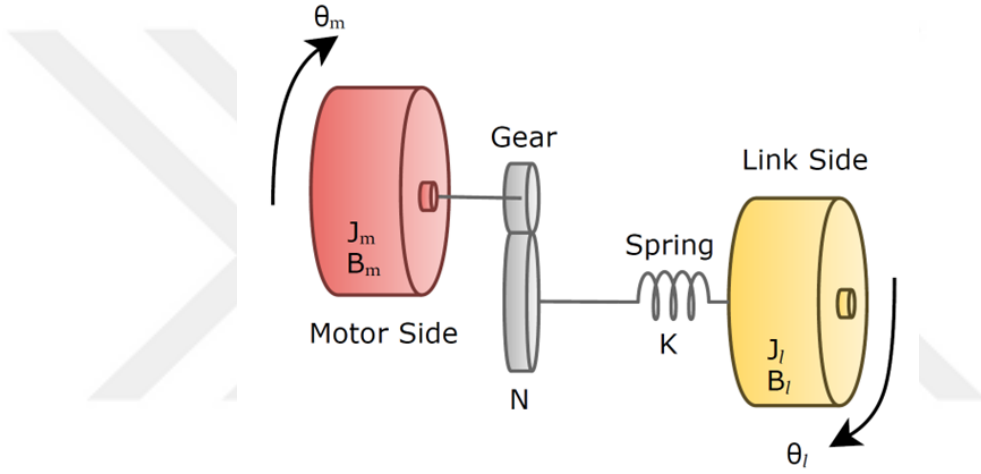
Table 1: Boundary Conditions of SEAs

Boundary Conditions	
Maximum Operating Torque	$\tau \leq 164$
Maximum Length (mm)	$L \leq 180$
Maximum Mass (kg)	3
Operating Voltage (V)	48

The deflection of the torsional spring is obtained by measuring the angle difference between the two encoders, one fitted at the motor output, and the secondary encoder connected to the actuator output through a secondary shaft passing through the hollow motor shaft. The maximum length of the high torque SEA is 168 mm and the length of the low torque SEA is 147 mm which is more modular and shorter than the compared to Lagoda et al. and Sergi et al. designs [1, 33]. In addition, in terms of weight and torque to mass ratio, this design is superior compared to other SEAs. Since the output torque can differ according to payload or study conducted. Hence the weight of the SEA is lighter than the other designs due to the frameless brushless motor and frames designed according to their dimensions. According to dynamic walking simulations, designed exoskeleton requires 164 Nm maximum output torque at knee joints and 94 Nm maximum output torque at the hip and knee joints of the exoskeleton. [34] The weight of the high torque SEA is 2.25 kg and the weight of the low torque SEA is 1.87 kg. The specifications of SEAs are given in Table 2. The side view cut CAD model of SEA is displayed in Figure 7. The realized SEA unit can be seen in Figure 2.1.

Table 2: Specification of SEAs

SEA Specification	Low Torque	High Torque
Frameless Motor Model	TBM 7615	TBM 7631
Maximum Cont. Torque (Nm)	99.6	169.0
Weight (kg)	1.87	2.25
Radius-Length (mm x mm)	47x147	47x168
Torque to Mass Ratio (Nm/kg)	50.5	72.9

**Figure 6:** Mechanical Illustration of SEA

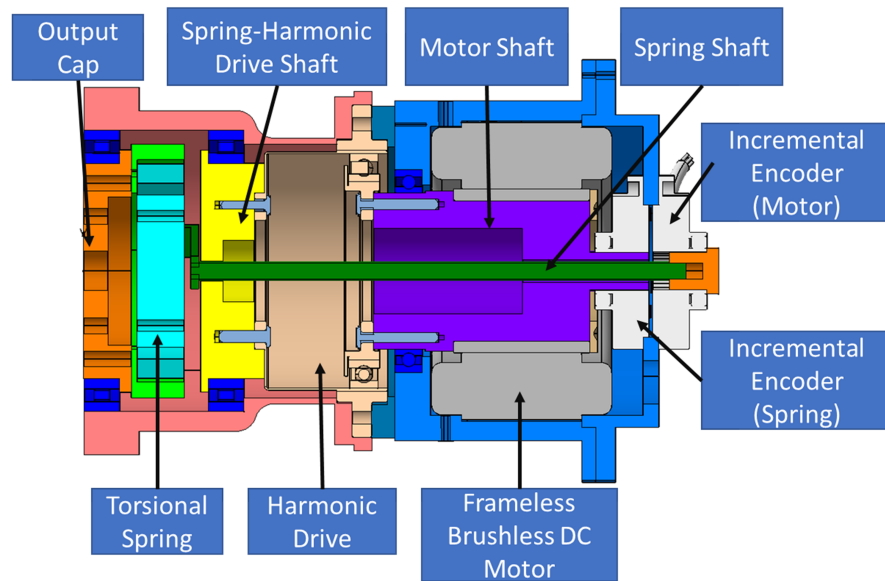


Figure 7: Cross sectional view of the finalized SEA design.

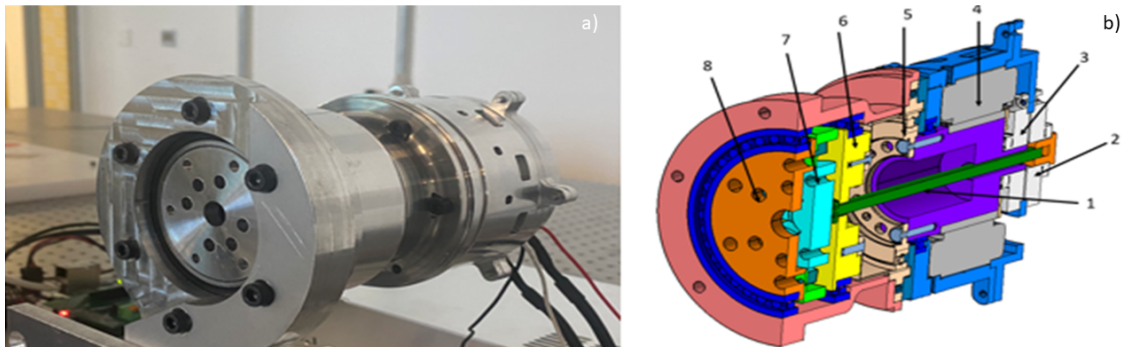


Figure 8: Realized SEA unit a) along with the cross-section of rotary SEA b). 1: spring shaft, 2: spring deflection encoder, 3: motor encoder, 4: frameless brushless DC motor, 5: Harmonic Drive, 6: Harmonic Drive output 7: torsion spring, 8: mechanical output

At the initial design of the SEA, the encoder measures torque from the deflection of the spring located on the torsional spring inside the SEA to provide modularity. Hence the rotational imbalance and concentricity of both encoder and torsional provide rotation of the encoder. Due to the rotation of the encoder, the measurement of the torque and control of the torque cannot be provided.

Initially, a brushless DC motor provided by FEMSAN was used to design the actuator frame. Later, another DC motor was used in the design to provide better motor parameters. The designed frame for the FEMSAN 18128-5 frameless brushless DC motor affected the modularity of the SEA and also affected the placement of the SEA on the powered exoskeleton. FEMSAN FC-8725 has been used instead of FEMSAN 18128-5 to provide more power and shorten the SEA. Continuous rated torque is the motor's continuous torque that it can output at rated speed, close to the max speed of the motor. This is a better judge of the motor's performance as it's the motor's torque at speed. Continuous stall torque is the motor's continuous torque at 0 speed. This is the locked rotor torque. Manufacturers test this torque by locking the rotor and then monitoring the motor temperature as current is powered into the motor. This motor has 1.6 Nm continuous torque output, much higher compared to previously chosen frameless brushless DC motor which had 1.2 Nm. However, to decrease the mass and provide the required torque to each joint of the exoskeleton, the frameless brushless DC motors was changed to Kollmorgen TBM 7631 and 7615.

To provide modular series elastic actuator assembly, the motor position encoder, and the spring deflection encoder was accumulated. One was fitted at the motor output, and the secondary encoder connected to the actuator output through a secondary shaft passing through the hollow motor shaft. The casing of the spring has been used for the transmission of the deflection of the spring to the shaft which goes through the motor shaft and absolute encoder to provide the measurement of the torque and position of the motor on the same side. The DC motor provides rotary motion and the continuous torque at the motor side of the

SEA. The strain wave generator or Harmonic Drive reduces speed of the output of the DC motor and increases the torque output. The spring attached at the end of the Harmonic Drive renders deflection on the elastic element . The deflection of the elastic element is transmitted by the secondary shaft which is attached to the outer cover fixed to the elastic element. The transmitted deflection of the spring is measured by the 21-bit encoder and the position of the DC motor is measured by the secondary 21-bit encoder attached to the DC motor shaft. Details of the torque tracking performance and the experimental procedures will be given in the following sections.

At the first glimpse of the carried out experiments the torsional spring can transmit its deflection through the designed spring shaft. Hence during the spring calibration experiments, the behavior of the springs changes due to the nature of the harmonic drive. The elliptic motion of the harmonic drive tends to pull the harmonic module inwards and the screws connecting the spring to the output of the harmonic drive cause preload and under the high impact of external forces the torsional spring fails at the feders located at the center of the spring. This problem later was solved by placing spacers between harmonic drive output and the torsional spring. Also, the tightness of the screws is another parameter that must be added to the assembling process. Based on these failed design iterations, further experiments on the other springs and SEA units are conducted.

The initially designed SEA has been shown in Figure 9, the developed mechanism of the torsional spring along with its realized form is demonstrated in Figure 10, the CAD view of the SEA is shown in Figure 11, the sectional view of the SEA is shown in Figure 12.

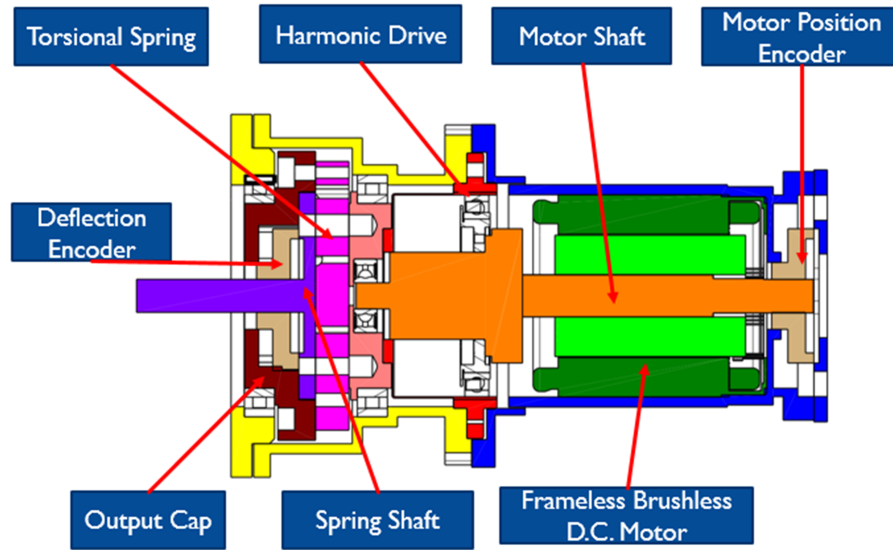


Figure 9: Cut section view of the first designed SEA

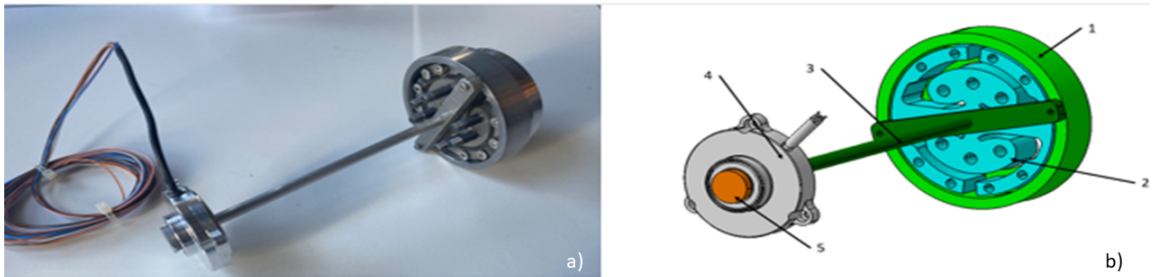


Figure 10: Realized spring deflection assembly a) Isometric view of spring deflection assembly b). 1: torsional spring cover, 2: torsion spring, 3: deflection shaft, 4: deflection encoder, 5: encoder cap [35]



Figure 11: FEMSAN SEA Module

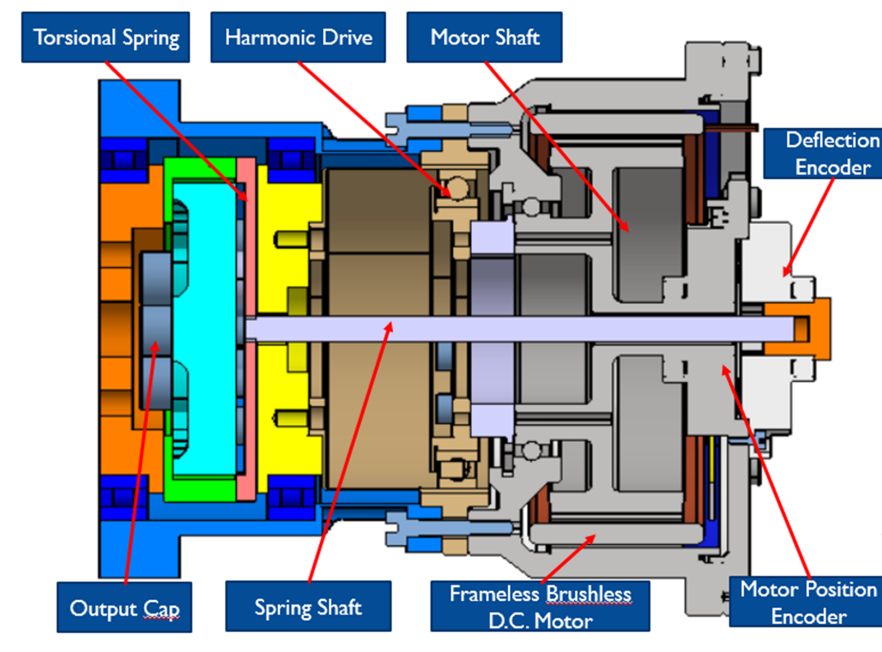


Figure 12: Sideview of FEMSAN SEA Module

2.2 Electrical Hardware of SEA

To operate the series elastic actuator mechanically an electrical setup has to be built. To provide power a 48 V power supply has been used on the system. This power supply provides enough current to drive the frameless brushless dc motor via EPOS 4 motor driver. After commutating each DC motor at EPOS Studio with the ethernet cable connecting the EPOS 4 motor driver to the PC.

The commutation of the DC motors is done by supplying 30 percent of cyclic torque to the DC motor and then the SSI position raw value is collected and used to find the SSI position offset of the DC motor. Then submitting the SSI position offset to the commutation value at BLDC mode of DC motor. Furthermore, with the submitted SSI offset value position, velocity and current tuning are conducted. The motor position values are collected via FENAC AS40HB3K 21-bit absolute encoders cabled to the Maxon EPOS 4 motor driver.

To read the spring deflection value at the link side of the series elastic actuator a secondary encoder has to be placed into SEA. The secondary encoder is placed at the rear end of the SEA and it can read deflection values via the EtherCAT shield which is connected to the EPOS 4 motor driver with an ethernet cable. These values are read by connecting this electrical hardware to NVIDIA Jetson AGX Xavier which can write and read data with high sample rates. The electrical hardware scheme is given in Figure 13.

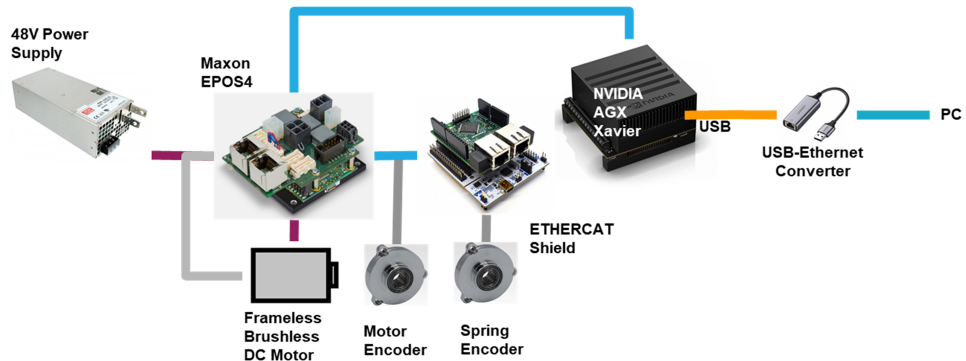


Figure 13: Electrical Hardware of SEA

2.3 Design of the Torsional Spring

The Elastic element is one of the main component of the series elastic actuator unit, the torsional spring requires long fatigue life cycle, stay in the elastic region under the specific load and safe. In addition the stiffness constant should be kept linear and durable under external forces. During the design pure bending based spring structure due to transmission of deflection compared to other elastic elements [36].

The spring stiffness aimed for a minimum 2° and a maximum 5° degree of deflection under maximum torque load of 160 Nm provided by the final design of the unit. On the other hand the position control bandwidth will be limited by the softer springs due to their influence in robot become more dominant even though they provide better torque measurement resolution. In this regard using stiffer spring could be more useful to provide larger position bandwidth but it may be limited. As a result, the deflection under the 160 Nm torque load determined as 2° when the resolution-accuracy trade-off considered [37]. The CAD drawing of the spring is given in Figure 14. Realized torsional spring is shown in Figure 15. and the parameters of designed spring were given in Table 3.

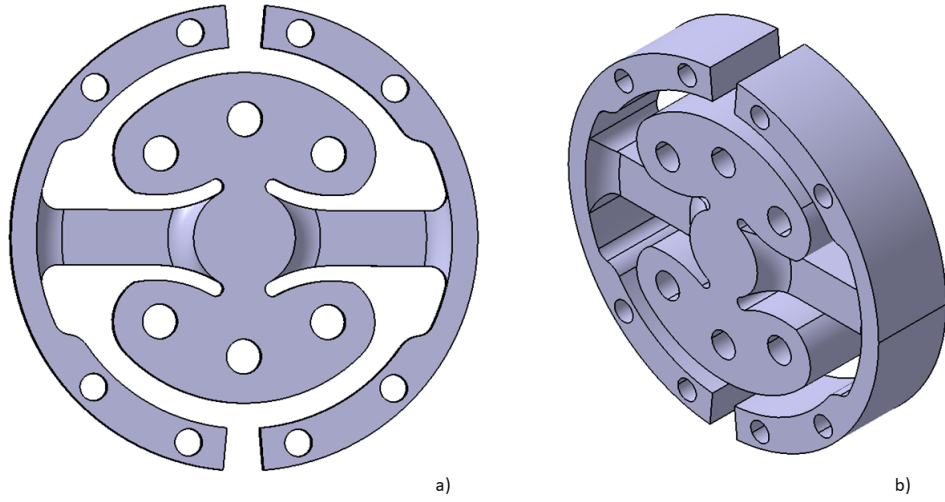


Figure 14: Frontal CAD View of Designed Torsional Spring a) with Isometric View b)



Figure 15: Realized Torsional Spring

Table 3: Parameters of The Spring

SEA Specification	Low Torque	High Torque
Spring Diameter (mm)	9	9
Outer Diameter (mm)	55	55
Feder Thickness (mm)	6.5	6.5
Total Thickness (mm)	8	12
Torsion Bar Angle (degree)	45	45
Angle of Deflection (degree)	1.88°	2°

CHAPTER III

SIMULATION RESULTS

3.1 Structural Analysis of the SEA Frames

To determine the safety and stresses acting on the SEA frames a structural analysis has conducted at each frame. The reason behind to conducting analysis individually due to output moment acting on the frames are different.

First, a structural analysis conducted on motor frame. The radial force acting on the bearings of the motor shaft has found 8.2 N. The radial force transmitted as bearing contact force via ANSYS Workbench. Then the motor frame fixed at the holes which is connecting the motor frame to harmonic frame also it is fixed where it is connecting the motor frame to the motor cap. Maximum principal stress acting on the motor found 0.15 MPa. This result may be seem oversafe design while using Aluminum 7075 series. Hence the design limitations caused by the frameless brushless DC Motors dimensions led the design at this safety. The maximum deformation of the motor frame can be seen in Figure 16. also the maximum principle stress of the motor frame demonstrated in Figure 17.

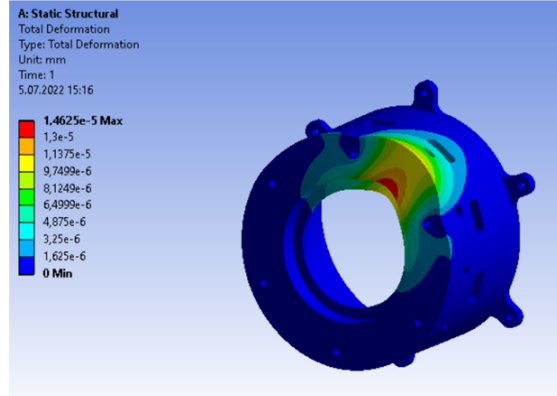


Figure 16: Total Deformation of Motor Frame of SEA

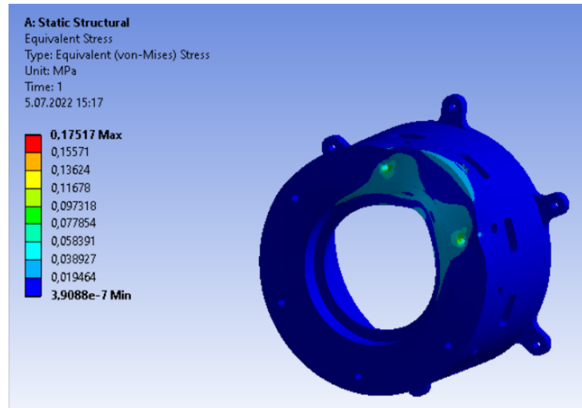


Figure 17: Maximum Principle Stress of Motor Frame of SEA

Then, the structural analysis is conducted at harmonic frame differs from motor with the 100:1 gear ratio of the Harmonic Drive. The radial force acting on bearings of the harmonic drive spring connecting shaft and output shaft calculated

as 392.5 N. The harmonic drive frame fixed from the holes where SEA connected to the motor frame also it is fixed at where it is fixed to the exoskeleton link. The maximum principal stress resulted with 30.89 MPa and the safety number of the harmonic drive calculated as 18. The maximum principle stress of the harmonic drive frame shown in Figure 18. Moreover, the total deformation of harmonic drive frame is given in Figure 19.

The frame design analysis conducted to prove and observe the safety of the chassis parts. The main loads occurred on the parts rendered by the radial and axial loads transmitted through the bearings of the rotary parts such as motor shaft and strain wave gearbox. Through the simulations the loads based on the walking torque data of the previously designed exoskeleton.[34]. As a result the deformation and the behavior of the stress acts higher where the push off motion occurs.

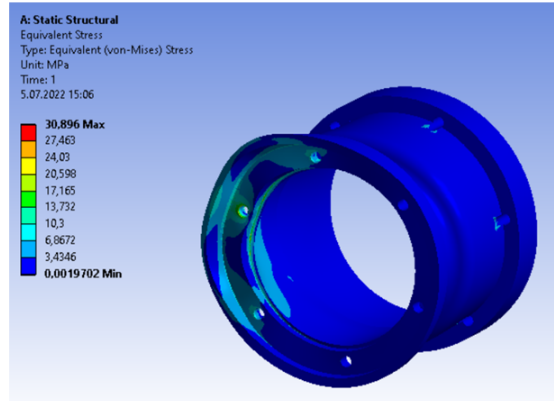


Figure 18: Maximum Principle Stress of Harmonic Drive Frame of SEA

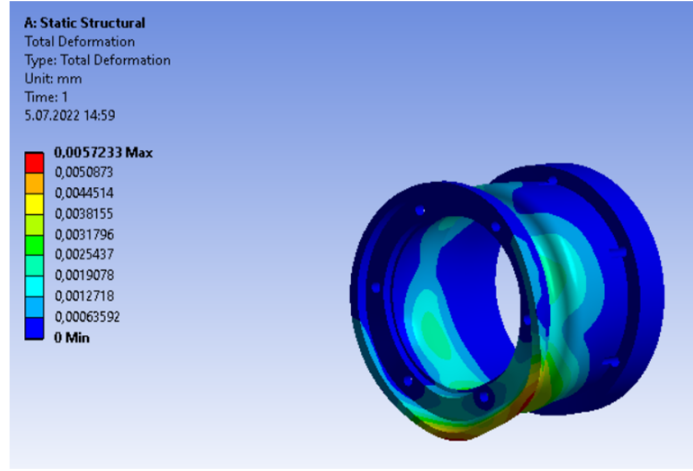


Figure 19: Total Deformation of Harmonic Drive Frame of SEA

3.2 Dynamic Analysis of the SEA Elements

To provide equivalent stresses and deformations occurred due to radial and axial forces resulted from the rotational motion on the rotary elements of the SEA a dynamic analysis conducted via ANSYS. During the walking motion of the powered exoskeleton angular velocity data of each joint have been converted in to motor velocity in terms of rpm. These data have been imported to ANSYS Simulation Software to provide dynamic analysis. Calculated total deformation and equivalent stress values used to observe the safety of the rotary elements.

The dynamic analysis conducted 2 main rotary element which are motor shaft which transmits the rotary motion to strain wave gearbox and the other element was output cap which provides angular motion to exoskeleton joints and transmits the torque. The dynamic analysis conducted for 10 seconds with a 1 kHz sampling rate and 1001 steps of rpm data have been used for each joint. The equivalent stress of the output cap is shown in Figure 20 and the total deformation of the output cap is demonstrated in Figure 21.

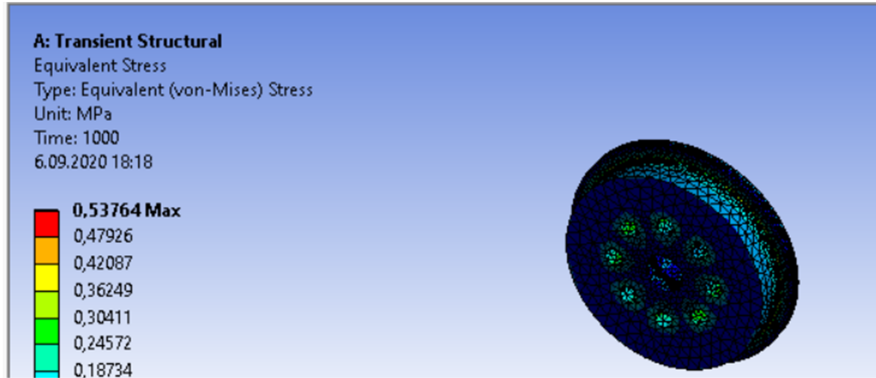


Figure 20: Equivalent Stress of Output Cap under Dynamic Analysis

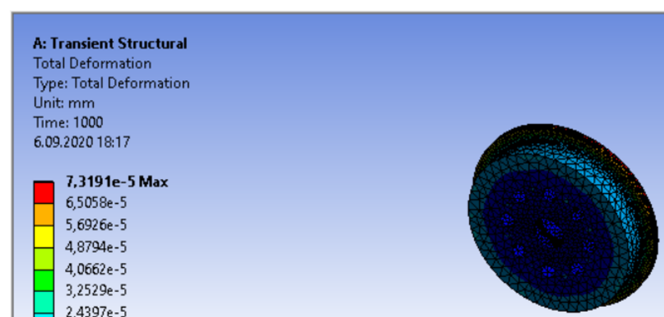
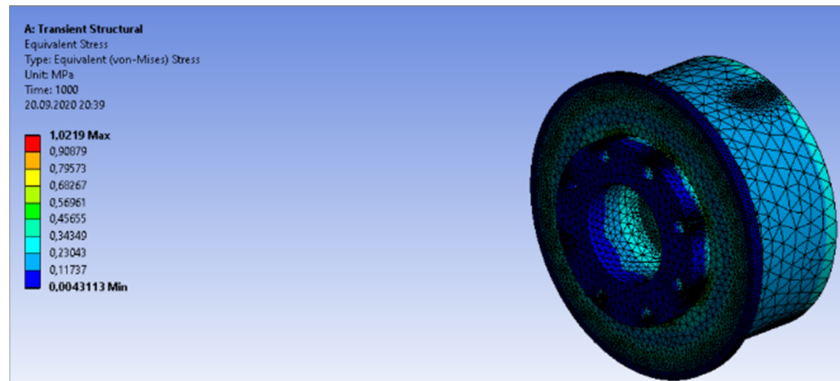


Figure 21: Total Deformation of Output Cap under Dynamic Analysis

Table 4: Dynamic Analysis Results of Rotary Elements

SEA Specification	Motor Shaft	Output Cap
Maximum Principle Stress (MPa)	120	160
Total Displacement (mm)	3.2	2.75

In addition, the equivalent stress of the hollow motor shaft is shown in Figure 22 and the total deformation of the motor shaft is demonstrated in Figure 23. Moreover, the results of the each element are given in Table 4. The flow chart followed during the dynamic analysis of the rotary components is illustrated in Figure 24.

**Figure 22:** Equivalent Stress of Motor Shaft under Dynamic Analysis

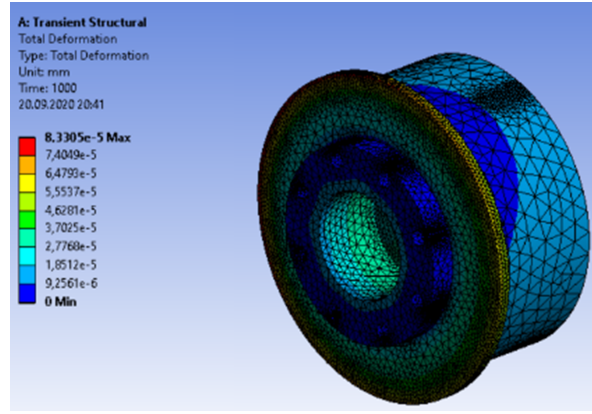


Figure 23: Total Deformation of Motor Shaft under Dynamic Analysis

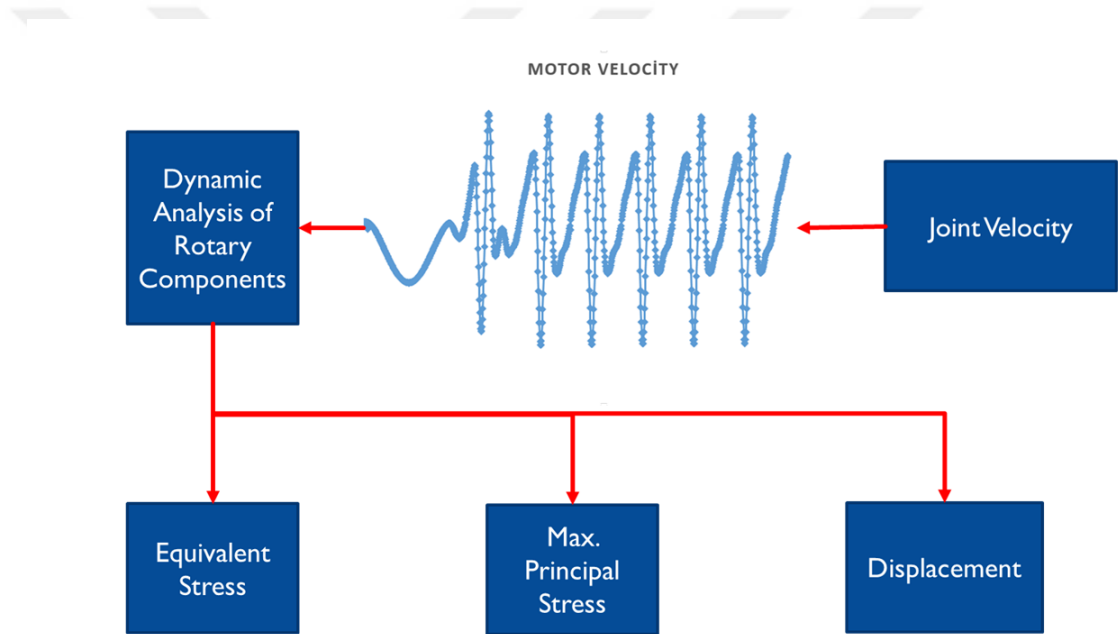


Figure 24: Flow Chart of Dynamic Analysis of Rotary Elements

3.3 Structural Analysis of the Torsional Spring

The shape of the torsional spring based on the previous works topology optimization conducted by Yildirim et al. According to their study the critical locations has been determined on the preliminary design by ranking the locations with respect to the values of the topology variable. The determined target of deflection was 2 degrees after 154 iterations conducted by FEA analysis 44 percent of the

material was removed from the preliminary geometry. The kidney shape of the spring comes from the topology optimization from the previous works of Yıldırım et al. This shape provides mobility and the transmission of the torque applied. In this sense the transmitted torque buckles the feders of the spring.[21]

To validate the designed torsional spring deflection resulted 2° of deflection under 164 Nm torque load a structural analysis has been conducted via ANSYS Simulation Software. The softness and hardness of the spring quantified by its spring coefficient based on applied maximum torque with respect to the deformation occurred under maximum torque. The elastic element stiffness related with the position control bandwidth. The stiffness of the elastic element related with the resolution of the angle measurement of the spring. The soft spring provides good position measurement quality hence the dynamic behavior of the soft spring limits the position control bandwidth due to its sensitivity under the applied load. The resolution of the spring is the deflection occurred at the rated torque which is the stiffness of the spring.

The soft springs used in SEAs limits the position control bandwidth. The dynamic nature of the soft spring become more dominant to the dynamics of the exoskeleton robot and limits the position control of the link side of SEA.

The material used during the simulation and manufacturing of the spring was Aluminum 7075 which has properties of high tensile strength, quality of manufacturability and vast usage of spring manufacturing area.

The torsional spring located between the output cap and the strain wave gearbox will be under cyclic loading during the walking motion of the powered exoskeleton. Therefore, the material chosen should have long fatigue life cycle the selected material meets with these desired boundary conditions.

During the simulation the spring is fixed from the holes located at the outer diameter and the 164 Nm load was given to the holes located at the inner diameter of the spring. The 164 Nm load was divided into 6 holes which resulted with 27.33 Nm applied at each hole. The given torques at each hole were accumulated at the

center point of the spring on the simulation.

To measure the deflection of the spring a feature of ANSYS was used which is called “Flexible Probe” to measure the deformation inserted to the structural system located at the rotating surface due to torsional bending of the spring. Then the calculated deformation converted into deflection in terms of degree by implementing “Command Tool” to “Flexible Probe”. The simulation results indicates that the deflection of the spring is 2.02° under 164 Nm torque load. In addition during deflection the spring should remain in the elastic region to prove that the principle stress should not exceed the tensile strength of Aluminum 7075.

The results shows that the maximum principle stress had found 243 MPa which is under 503 MPa tensile strength of Aluminum 7075. Same procedure has been followed for the SEA module which has a torque output of 94 Nm in order to achieve 2° degrees of rotation. Using the same spring yields a stiffer spring constant and 94 Nm torque output cannot provide 2 degrees of rotation. Eventually due to the nature of the angle of twist motion and equations a thinner spring with the same feder thickness hence the spring thickness has been decreased by 4 mm. Following the same structural analysis method the spring provides 1.88° of rotation. The simulation results validates the spring can remain in the elastic region and it is able to deflect with respect to desired value. The constraint equations is the term of the ANSYS report after the structural analysis of the torsional spring is conducted. The applied circular supports at the outer diameter and the moment applied to the holes at the inner diameter defined as the constraint equations. The nodes are shown in constraint equation in red and accumulated at center node of the elastic element to calculate rotational element at z-direction. The constraint equation illustrated in Figure 25, the maximum principal stress distribution is shown in Figure 26 and the results of each solution is given in Table 5.

Table 5: Sinusoidal reference tracking results of the actuator controlled using Cascaded PID Controller

	Velocity RMSE (10^{-3} rad/s)		Torque RMSE (mNm)	
	with DoB	w/out DoB	with DoB	w/out DoB
Cascaded PID Controller				
0.5 Hz	0.261	0.566	3.62	4.72
1 Hz	0.267	0.601	5.52	6.24
2 Hz	0.470	0.621	9.19	11.81

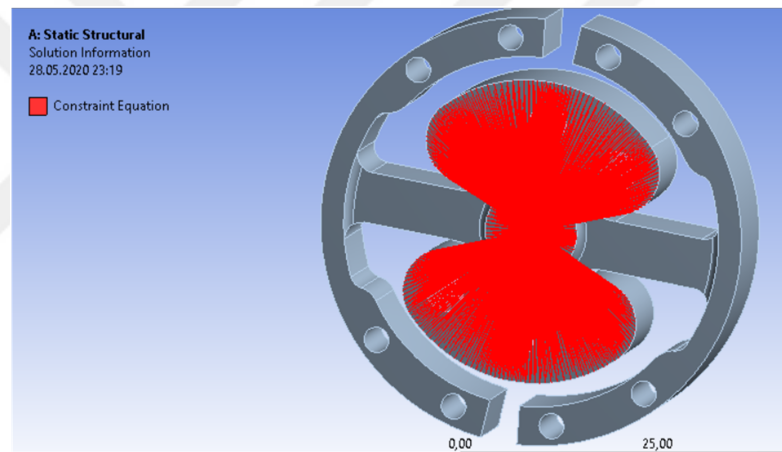


Figure 25: Constraint Equation of Spring under Structural Analysis

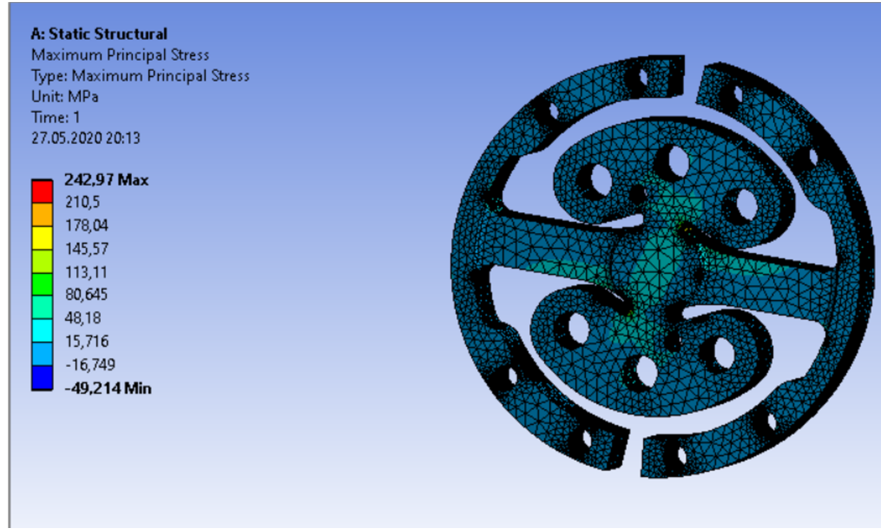


Figure 26: Maximum Principle Stress of Torsional Spring

3.4 Fatigue Life Simulations of the Torsional Spring

Since the stiffness coefficient and elastic/plastic limits of the spring are determined. A fatigue life analysis assessed for the torsional spring the flow chart of the followed methodology depicted in Figure 27. In this analysis experimental torque data provided from the real-life supported walking experiments have been used [15] rather than considering constant-valued torques [38]. During the fatigue-life cycle analysis the S-N curve and the fatigue strength of 159 MPa under $5 \cdot 10^8$ completely reversed stress cycles [39] were implemented to the simulation.

Furthermore, the applied torque was 164 Nm at the each hole at kidney shaped torsional spring it is divided into 6 holes. Furthermore during scaling the completely reversed walking torque data its must be scaled down to unit torque as maximum value in order to assign 164 Nm which is the maximum continuous torque provided by the SEA unit. To provide completely reversed cycle of torque value 25 seconds of data have been used. Then simulation input becomes 25 s of

duration of simulation with 1 kHz sampling rate 25001 data have been used.

The result of the fatigue life cycle analysis obtained as $6.6 \cdot 10^6$ cycles. Since one cycle of the torque data is 25 seconds the result becomes 45833 hours of operation. Therefore the fatigue life of the torsional spring becomes 1900 days after conversion which yields to 5 years of fatigue life cycle. The fatigue life distribution of the torsional spring is shown in Figure 28.

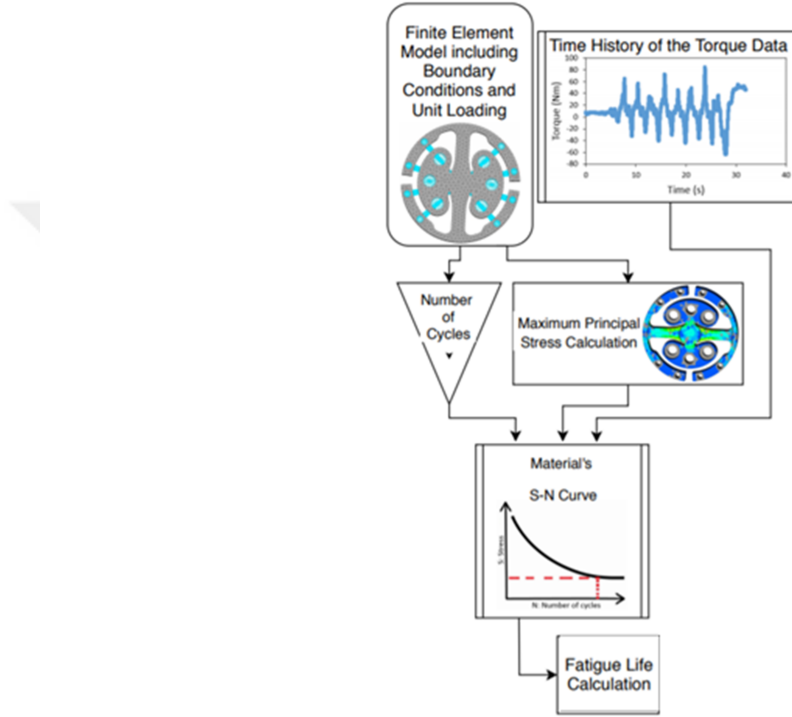


Figure 27: Fatigue Analysis Flow Chart of Torsional Spring

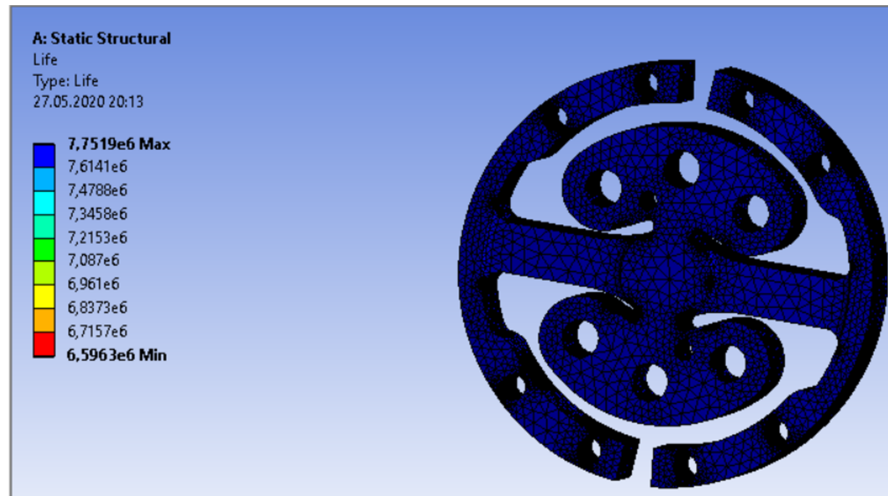


Figure 28: Results of Fatigue - Life Cycle Analysis in terms of Cycle

CHAPTER IV

TORQUE CONTROL STRATEGY OF SEA

In order to improve the actuators performance researchers still proposes undergoing research to solve the robust and high fidelity control of SEAs. Main focus of this controllers contains disturbance observer to diminish the nonlinear disturbances acting on the actuator. While using simple PID controller on SEA is resulted with under performing results, Cascaded PID controllers are generally used in SEA implementations. In order to increase the performance of the controller, Cascaded PID was analyzed under two cases which is with and without DoB.

4.1 Cascaded PID Controller

As its name cascaded controller refers to multiple control loops which the output generates a reference for the other loop. It is stated that Cascaded PID controllers increase the stability and decrease the disturbances occurring at the system in addition, they widely used in torque-force control of SEA.[19, 25]

Heike et al. proposed an analysis about the improvement of the stability while applying cascaded PID on the system. It is achieved that with applying high-frequency inner velocity loop, outer loop torque control provides better tracking results. [30] The control diagram of cascaded PID controller has been illustrated in Figure 29. In this diagram, the outer loop feedback control the output torque of the actuator to the PID controller. The output rendered by PID controller generates the reference motor velocity and with the PI controller provides the velocity control at the inner loop with respect to the calculated velocity reference.

The measurement of the torque starts with giving reference torque input to the SEA the measured deflection of the spring multiplied by spring stiffness in control algorithm. The deflection of the spring is calculated the difference between the

spring position and the motor position divided by 100. The position of the spring is measured by the 21-bit absolute encoder by inner shaft goes through the motor shaft and fixed with a cap to measure the deflection at the left hand side of the SEA. The position of the motor shaft is also measured by the 21-bit absolute encoder. The difference between the input torque and the measured torque is resulted with torque error goes through PID and resulted as velocity reference. The motor velocity is also measured by the encoders deducted by the obtained reference velocity. Resulted velocity error goes through the inner loop PD which is resulted as torque input to the dynamic system. In this sense, torque measurement provided by the elastic element is achieved.

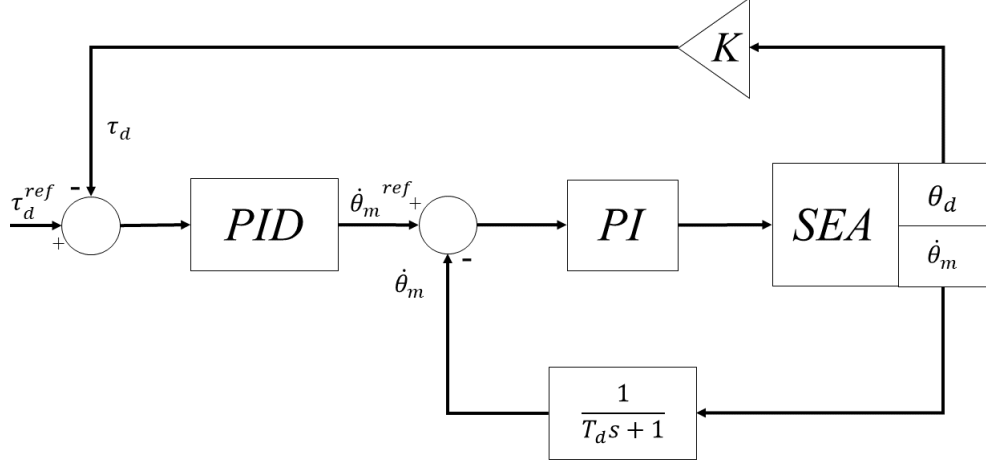


Figure 29: Cascaded PID Control Diagram

4.2 Cascaded PID Controller with DoB

Series elastic actuators are high susceptible to nonlinear disturbances such as viscous friction of the DC motor and friction of the strain wave gearbox. DoB implemented control methods improve the robustness of the system [32].

Since the frequency of the inner velocity loop affect the overall performance directly, main goal of the inner loop is achieving robust velocity control in order to increase the torque tracking performance of the whole system. Adding disturbance

observer at motor side can compensate the disturbances generated from the friction and external disturbances acting on the load side of the SEA. The control diagram of the cascaded controller with a DoB is illustrated in Figure 30.

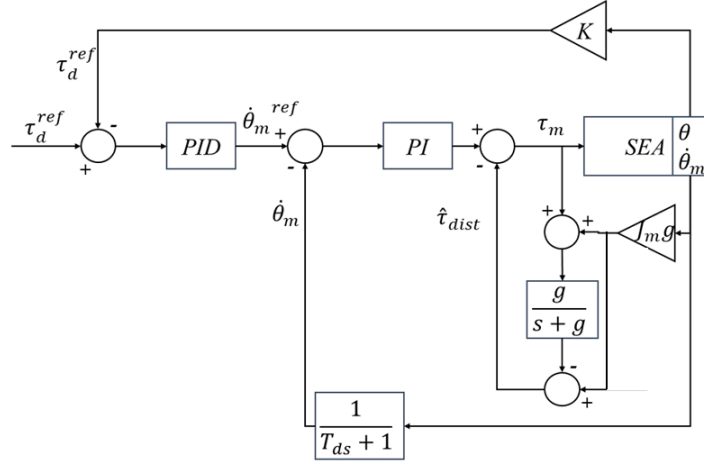


Figure 30: Cascaded PID wit DoB Control Diagram

CHAPTER V

EXPERIMENTAL RESULTS

5.1 Identification of System Parameters

In this experiment, the viscous friction coefficient is found from the relationship between the measured velocity and current of the DC motor under chirp input. By providing several chirp inputs the data of velocity and current is collected. The average of the linearized data of the transfer function of the DC motor has taken which is given in Equation (1). b is considered as the viscous friction coefficient. The inertia of the motor shaft J_m is known.

Average of the viscous friction data is taken positive and negative side simultaneously. The chirp signal input renders positive and negative velocity data by rotating the motor shaft the collected negative and positive velocity with respected to the rating current linearized during this system identification. In this sense the dead zone of the data which renders 0 velocity is omitted and the data at the negative side and positive side of the coordinate frame linearized separately by using the equation of motion of the DC motor. By taking the average linearized function for the each negative and positive data seperately positive and negative viscous friction can be found.

$$G(s) = \frac{1}{J_ms + b\dot{\theta}} \quad (1)$$

To define the viscous friction parameters an identification process has to be conducted. To find negative stiction and positive stiction values of the rotor of the motora, a ramp signal input was given to the frameless brushless DC motor. During this process, a PD position controller has been used. 10 data sets were collected, where velocity and the current relationship were used. The dead zone found at the velocity-current graph was ignored since it did not correlate with the

negative and positive stiction.

The magnetic effect between the stator and the rotor differs during the rotational motion. In order to calculate the viscous friction of the DC motor several chirp signals are given to the rotor as an input. Then the measured velocity and the current of the DC motor. The main deterministic error in this experiment is the coefficient of the positive and viscous friction which can be lowered by increasing the iteration amount of experiment and obtain the optimal value of the friction coefficient. To find the positive and negative stiction values, the average of the data taken for the positive and negative rotation separately then the averaged data was linearized to calculate the stiction values. Obtained data and velocity-current graph can be seen in Figure 31.

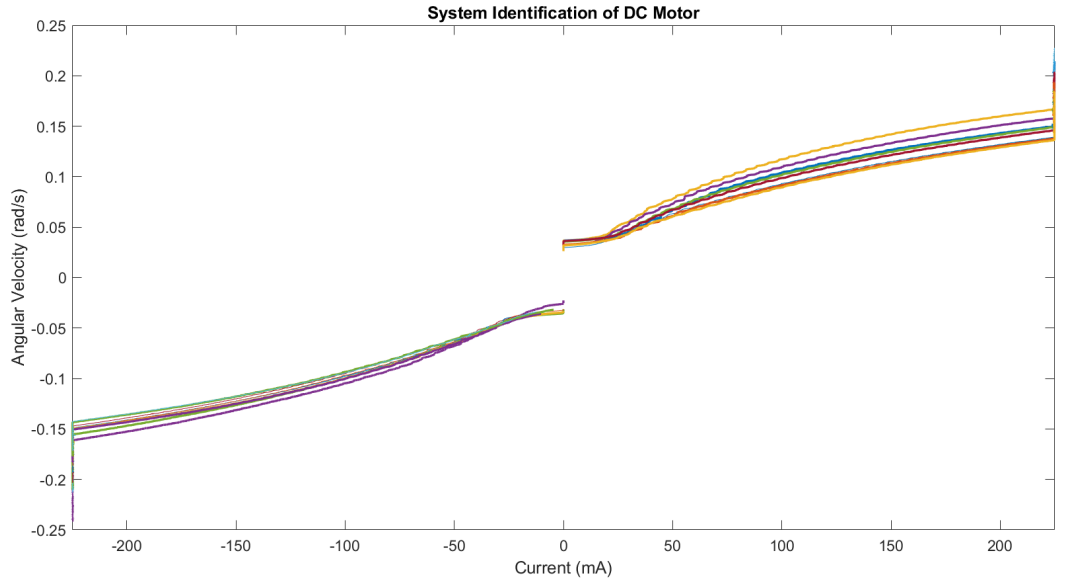


Figure 31: Current-Velocity Graph of DC Motor System Identification

5.2 Calibration of Torsional Spring

The stiffness coefficient of the torsional spring was determined using the relation between the spring torque and the angular displacement. The resolution or the stiffness of the spring can be calculated by measuring the output torque of the SEA with a 6-axis force/torque sensor and the deflection of the spring is measured

with a 21-bit encoder and the obtained values are used to calculate stiffness of the spring. To provide environmental stiffness value which is needed for the control algorithm a torsion-based test unit is assembled. The spring calibration test unit consists of a series elastic actuator, a secondary output shaft to connect the output of the series elastic actuator, and the shaft designed to transmit the moment. The designed shaft is connected to the input of the force/torque sensor (OptoForce HEX-70-CE-2000N) which is fixed to the frame of the calibration unit. The CAD view cross-section of the torsional spring calibration unit is shown in Figure 32 and the realized unit can be seen in Figure 38.

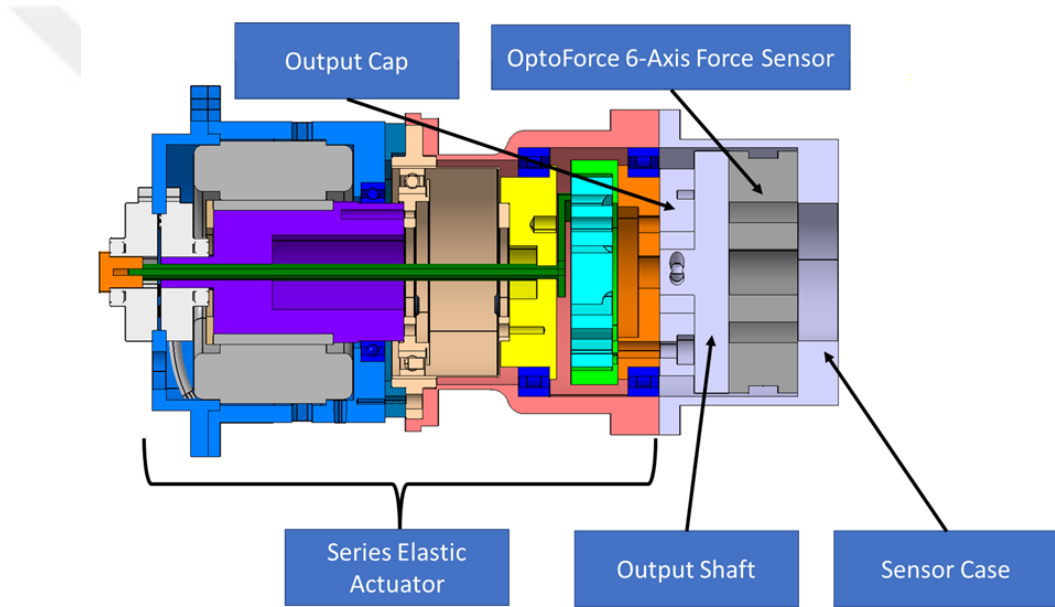


Figure 32: CAD Cross-Section view of Spring Calibration Unit

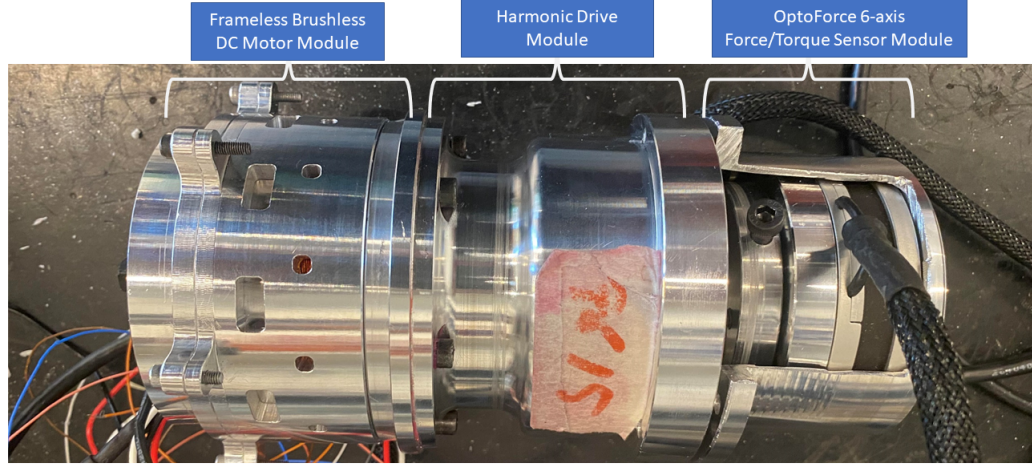


Figure 33: Realized Spring Calibration Unit. F/T sensor is fixed between the output shaft and the motor frame and prevents rotation. Output shaft is connected to the F/T sensor so that the stall torque can be measured.

To implement torque control on the series elastic actuator, the stiffness coefficient of the torsional spring must be known. To measure the stall torque of the actuator a force/torque sensor (OptoForce HEX-70- CE-2000N) was used. The sensor was statically loaded by fixing it to the output of the series elastic actuator. The angular displacement of the torsional spring was measured with the 21-bit absolute encoder (FENAC AS40HB3K).

The torque measured by the 6-axis force/torque sensor which is fixed to the end of the frame. The followed methodology is providing buckling at the end of the SEA which is the torque sensor. The buckling is rendered by the secondary output cap attached to the end of the SEA and this secondary output cap is fixed to the shaft with the horizontally opened holes and screwed to the shaft which is attached to the torque sensor. The main equations used to calculate the torque is given in Equation 2.

$$\tau = K\theta \quad (2)$$

where K is the spring stiffness constant, θ is the spring deflection and τ is the output torque.

The relation between torque output and the angular displacement was expected to be linear, as already presented in [18] with a simulation study. The angular displacement vs. torque graph is shown in Figure 39. The stiffness coefficient is used to calculate the desired torque at the output. A linear fit is made on the collected data. The graph shows the positive twisted spring deformation, but the same data is observed for the negative side. The linear fit has an offset due to the non-linearity of the spring structure. The main deterministic error in this experiment is the stiffness coefficient of the elastic element. Whether the stiffness is incorrect the error of torque measurement remains constant and cannot be decreased.

Near zero angular twist, a dead zone is assumed in the spring linearization, but in practice this offset is neglected. The experimental spring coefficient was found to be $K=3919 \text{ Nm/rad}$, which is in argument with the analysis results.

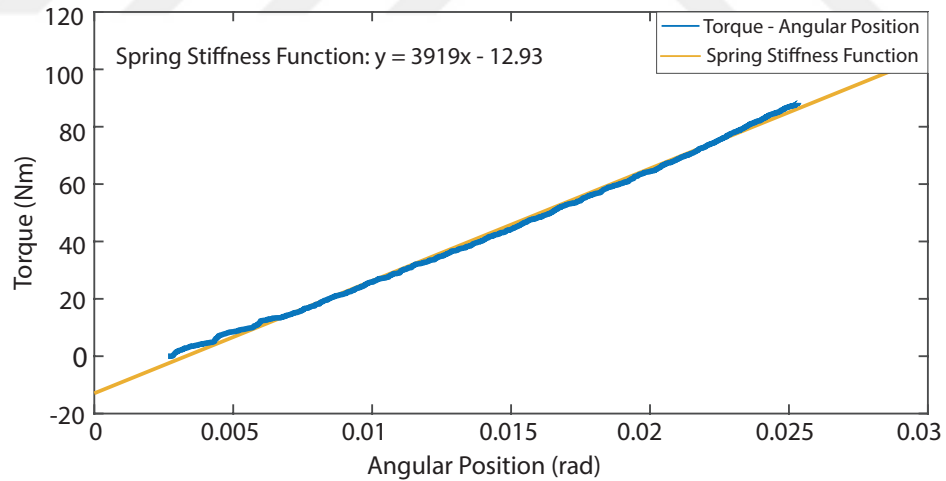


Figure 34: Angular Position-Torque Graph for Spring Stiffness Calibration

5.3 Comparison of Cascaded PID Controller with and without DoB

The torque control of the series elastic actuator is made using a cascaded PID controller by feeding the position and the velocity of the motor to the controller.

The main reason to apply a cascaded PID controller to the SEA is to get rid of the disturbances without using complex algorithms. The outer loop of the controller, which is a PID controller, gets the spring encoder data and the inner loop of the controller, which is a PI controller, gets the motor encoder data [40].

To evaluate the performance of the cascaded PID controller, two cases have been implemented. The first case is a cascaded PID controller with a disturbance observer (DoB) to reject the external disturbances. The second case is the same controller without added DoB. Disturbance observer is an inner-loop output-feedback controller whose role is to reject external disturbances and to make the outer-loop baseline controller robust against plant's uncertainties. Therefore, the closed-loop system with the DOB approximates the nominal closed-loop by the baseline controller and the nominal plant model with no disturbances. Sinusoidal torque signals were given to the system as reference inputs with different frequencies, 0.5 Hz, 1 Hz, and 2 Hz, with an amplitude of 10 Nm. The main aim was to distinguish the behavior of the system at different operating frequencies and the effect of DoB on angular velocity control.

Since velocity control of the inner loop provides an improvement on torque tracking performance of the link side of the motor with higher frequencies. The velocity tracking performance of the cascaded PID is observed with and without DoB and their velocity tracking performance and error plot are demonstrated in Figure 40, Figure 41 and Figure 37 .

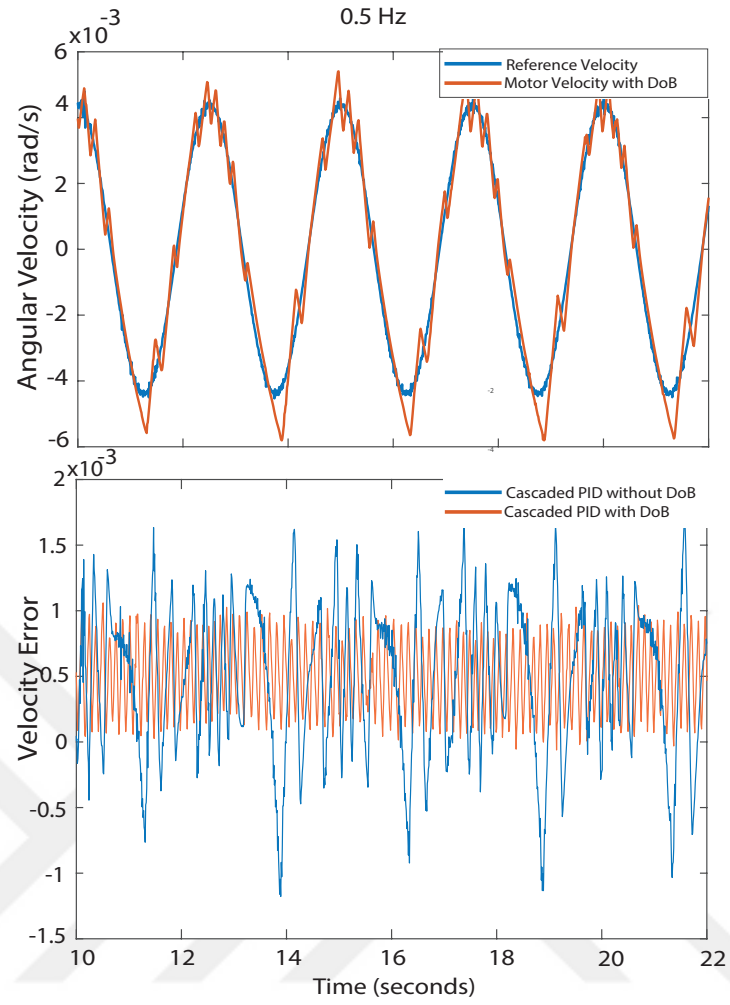


Figure 35: *Top:* Velocity tracking performance of the DoB added control system for 0.5 Hz sinusoidal reference. *Bottom:* Velocity difference between reference and motor velocity with and without DoB added to the control system.

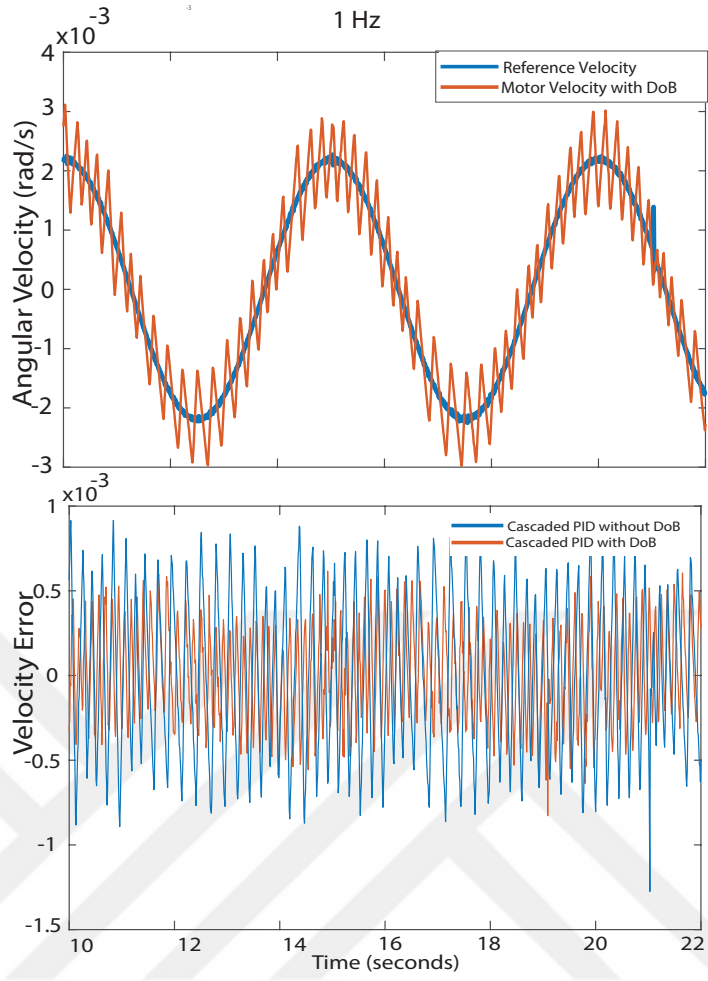


Figure 36: *Top:* Velocity tracking performance of the DoB added control system for 1 Hz sinusoidal reference. *Bottom:* Velocity difference between reference and motor velocity with and without DoB added to the control system.

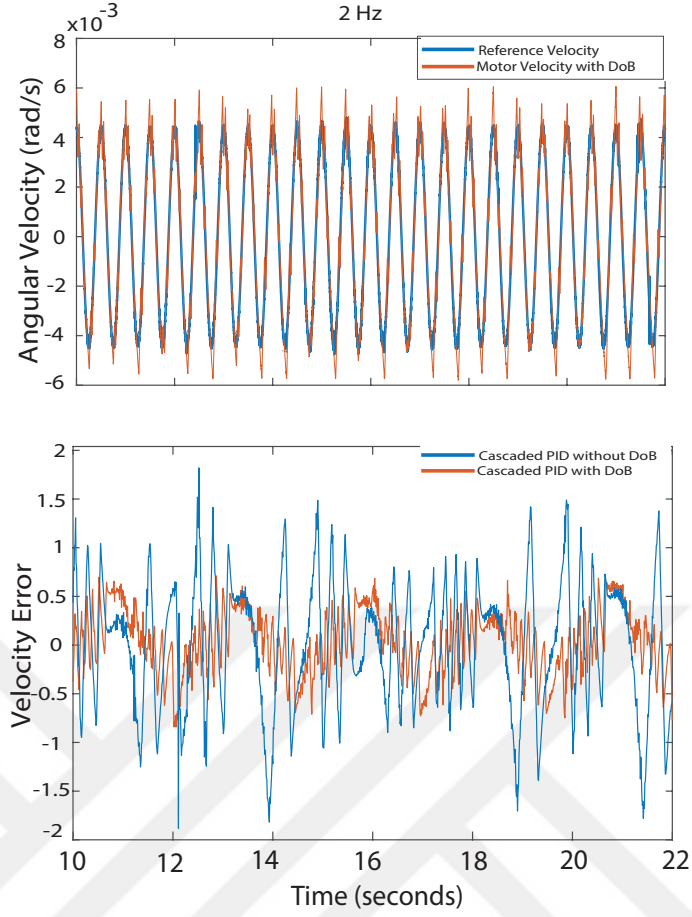


Figure 37: *Top:* Velocity tracking performance of the DoB added control system for 2 Hz sinusoidal reference. *Bottom:* Velocity difference between reference and motor velocity with and without DoB added to the control system.

RMS torque errors of controller with and without DoB added and RMS errors of velocity of actuator are given in Table 5. As expected, as the frequency increases, the RMS error of the controllers also increases. However, the results obtained demonstrate that the actuators are suitable to be used in the exoskeletal systems. The error can be lowered by conducting further system identification on DC motor to obtain viscous friction coefficients and through the experiments the torque error can be reduced. In addition, conducting spring stiffness calibration experiments at different conditions and environments can diminish the error between experimental stiffness and the actual stiffness of the spring. The output torque along with reference signals and the error plots with and without DoB

shown in Figure 38, Figure 39 and Figure 40.

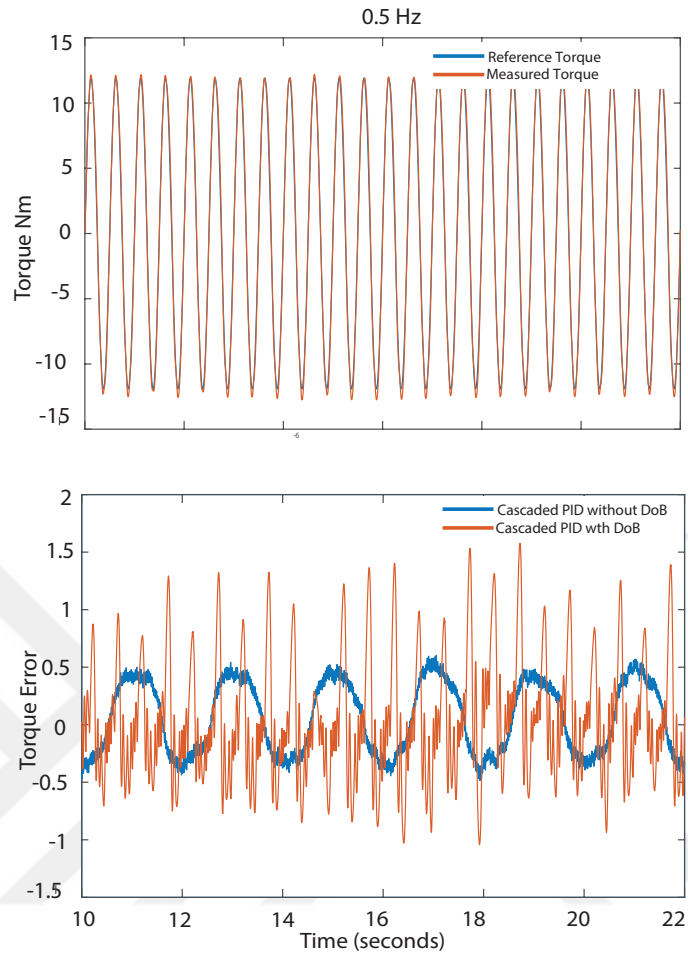


Figure 38: *Top:* Torque tracking performance of the DoB added control system for 0.5 Hz sinusoidal reference. *Bottom:* Torque difference between reference and output torques with and without DoB added to the control system.

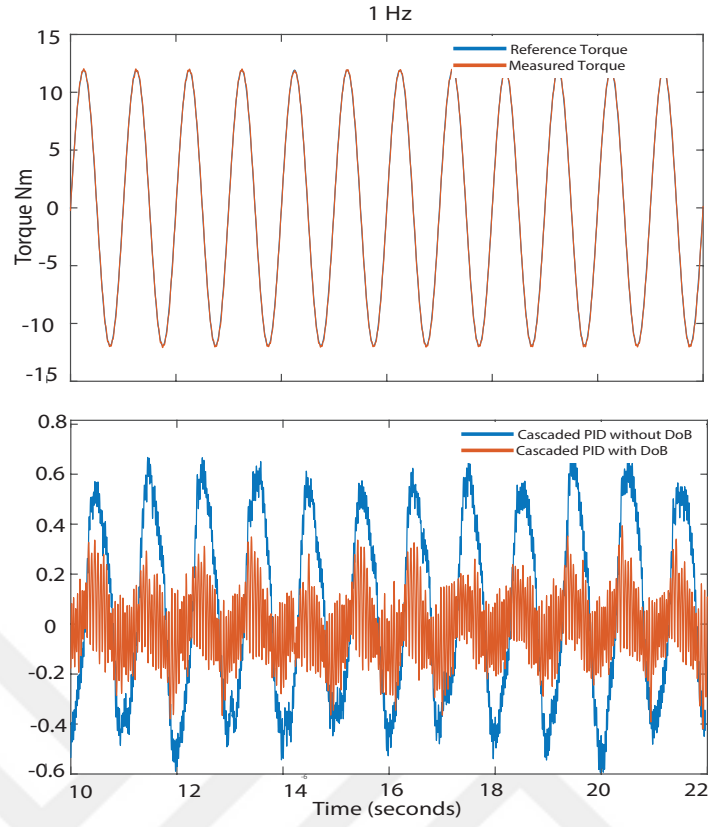


Figure 39: *Top:* Torque tracking performance of the DoB added control system for 1 Hz sinusoidal reference. *Bottom:* Torque difference between reference and output torques with and without DoB added to the control system.

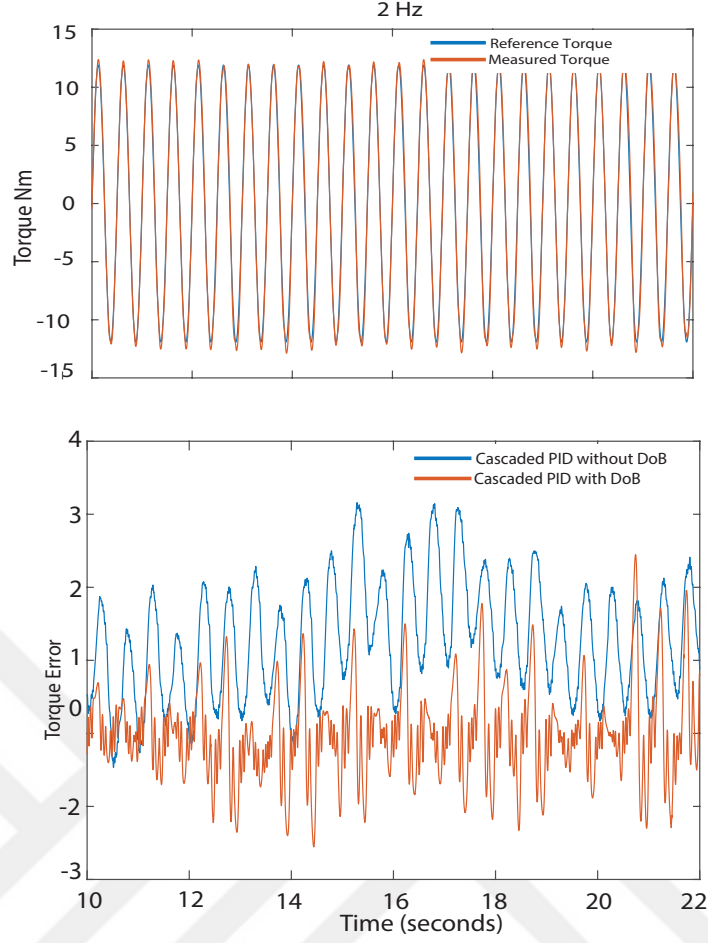


Figure 40: *Top:* Torque tracking performance of the DoB added control system for 2 Hz sinusoidal reference. *Bottom:* Torque difference between reference and output torques with and without DoB added to the control system.

5.4 Zero Torque Control of SEA

To observe the force sensing capabilities of the series elastic actuator zero torque control (ZTC) has been applied. The main goal of this experiment is simply to observe the force sensing performance of the spring which is located at the link side of the SEA also useful for further control algorithms. To determine the compliance of the SEA.

During the control process cascaded PID with zero input is applied, also gravity compensation and friction compensation has been added to the control algorithm.

A link of the exoskeleton has been attached to the output of the SEA to apply

external force to the tip of the link. Moreover, it is observed that ZTC is providing a frictionless motion to the link side of the SEA under a small force applied. In addition, the gravity compensation also provides the neglect of the mass of the link during the ZTC to diminish excess forces occurring at the output of the SEA during ZTC. The graph of link angle, torque, and current of the SEA has been illustrated in Figure 41.

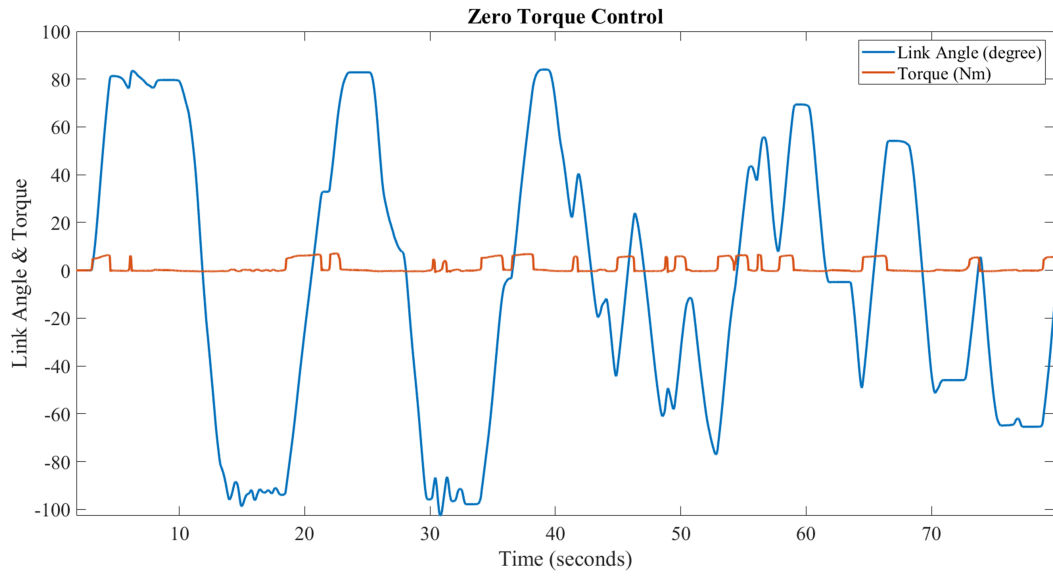


Figure 41: Demonstration of zero torque control. Output torque is kept close to zero although the output shaft rotates.

CHAPTER VI

CONCLUSION

In this thesis, a new SEA design where the deflection of its torsional spring is measured with the specially designed double shaft mechanism is presented, and also the torque control of the SEA was achieved at the desired level with measurements. With this proposed double shaft system, the assembly and disassembly process has been made modular by dividing the SEA into sub-systems such as the actuator-motor module and the reducer-spring module. The proposed series elastic actuator is lighter, shorter, and a high torque-to-mass ratio is achieved. Compared with the previously designed SEAs such as Sergi et al., which has 165 mm in length and mass of 1.8 kg, with torque to mass ratio of 13.72 Nm/kg [33], and SEA design by Lagoda et al., which has length of 185 mm, weight of 3.175 kg, and the torque to mass ratio of 31.5 Nm/kg [1], our design provides higher torque to mass ratios, which are 50.5 Nm/kg for the small actuator and 72.9 Nm/kg for the larger actuator and shorter lengths of 147 and 168 mm, respectively.

Accurate tracking performance was achieved using the motor and the spring encoders even though torque reference input was given. The torque tracking performance of the torsional spring was tested by applying cascaded PID controller with and without DoB to the control system. The tracking error results indicate that applying disturbance observer to the system decrease the error of the angular velocity and of the torque. The successful application of torque tracking at different frequencies shows that the actuator has a control bandwidth at the desired level.

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