2013 IEEE/RSJ International Conference on
Intelligent Robots and Systems (IROS)
November 3-7, 2013. Tokyo, Japan

A Series Elastic Actuator as a New Load-Sensitive Continuously Variable Transmission Mechanism for Control Actuation Systems*

Görkem Seçer and Efe Uzel

Abstract— **Electrical motors are often used to actuate mechanisms with varying speed and under varying load e.g., missile control actuation systems or robot extremities. Sizing of the motor is critical in these applications where space is limited. Varying speed and load cause poor utilization of the motor if the reduction ratio is constant. This requires employment of a larger motor than the smallest possible with ideal variable transmission. To realize better utilization of the motor, we propose a series elastic actuator that works as a load sensitive continuously variable transmission system. This is achieved by combining a pin in slot mechanism with an elastic output shaft. The proposed mechanism is modeled and simulated using example missile flight data and a typical robot finger loading scenario. Results are compared to the constant reduction ratio case, and advantages of the proposed mechanism are shown as decreased actuator size and power.**

I. Introduction

Guided missiles (GMs) use actuated aerodynamic control surfaces (CSs), or fins, to intercept their targets. CSs are rotated by control actuation systems (CASs) according to CS deflection commands generated by the autopilot system [1]. CAS is a servo system consisting of a controller, actuator, reduction mechanism and sensors.

In most CAS applications electromechanical actuators provide sufficient controllability and power density [2]. Hence, we limit the scope of this paper to electromechanical actuators (i.e., electric motors). Space in radial direction available for a CAS is usually limited in a GM, whereas in the axial direction, which is the missile's roll axis, it is more abundant [3]. With such geometric characteristics, placing the motor parallel to the missile's roll axis and outputting the torque in the radial direction through the transmission mechanism result in a compact CAS. Electrical motors nominally generate power at high speed and small torque values than CAS requires so a transmission mechanism with a high reduction ratio has to be used. This mechanism can be a constant reduction ratio type or variable reduction ratio type. Bevel and worm gears are two examples of constant ratio type mechanisms. However, they are not used in modern missile CASs since high reduction ratios cannot be achieved with bevel gears, and efficiency is low in worm gears compared to their modern replacements. The pin-in-slot mechanism is popular for the missile CAS transmission problem and mentioned in detail in Sec. III. Variable transmission mechanisms are favored in applications where the CAS is exposed to a variable load profile [4], which is the case for GM CASs, because of their contribution in efficiency by making utilization smaller motors possible.

Among the previously developed variable transmission mechanisms, two main types are stepped and continuous. Stepped types allow a selection a discrete set of predetermined reduction ratios and cause a loss of power transmission during shifting. These drawbacks make continuously variable transmissions (CVTs) more suitable for the applications in consideration.

Most commercial CVTs make use of the friction to change the reduction ratio. Efficiency becomes an issue for them due to power losses. Besides that, their weight and big size makes them infeasible for CAS applications. Regarding this, we leave friction CVTs out of consideration.

As explained in [5], the reduction ratio of a CVT can be changed either actively or passively. Active adjustment of the reduction ratio requires an additional actuator while passive adjustments can be achieved by an elastic element of deformation that results in automatic adaptation of the reduction ratio with respect to a load. The former technique increases weight and volume used by CAS though bringing flexibility in the reduction ratio change. Therefore, the latter technique is more suitable for applications where weight and space considerations are important. They are also called load-sensitive CVTs.

There are several disadvantages of the existing load- sensitive CVTs in the literature. The CVT proposed in [5] and [6] is unidirectional, which makes it impracticable for some applications. The reference [7] modifies a standard screw/nut mechanism resulting in a new load-sensitive CVT. In [8], a CVT comprising a pulley, rollers, levers, wires connecting them, and a spring is presented. However, designs of [7] and [8] are highly complex, and this lowers their feasibility. A new load-sensitive CVT is presented in Yamada's recent work [4] which is not very cost-effective as it is composed of two ball screws. Drawbacks of existing load-sensitive CVTs are the motivation behind this work.

In this paper, we present a novel load-sensitive CVT mechanism, which is also a series elastic actuator [9]. Our main contributions can be summarized as decreasing the size of CAS by being able to select a smaller motor and improved force control which actually comes with the series elasticity feature. Then, we demonstrate the performance and effectiveness of the proposed system via simulation results. Our design gets one step ahead of the previous work [4]-[8] with its simple design, mixed series elasticity and CVT feature, and bidirectional operation capability.

*Research supported by Roketsan Missiles Industries Incorporation.

G. Seçer is with Roketsan Missiles Industries Inc., Ankara, Turkey, also a Ph.D. student in the Department of Computer Engineering, Middle East Technical University, Ankara, Turkey (gsecer@roketsan.com.tr).

978-1-4673-6357-0/13/$31.00 ©2013 IEEE

Although intended for GM applications, the proposed mechanism may find applications in biomimetic robots too. For example, legs of walking robots need to support the weight of the robot while in contact with the ground but move quickly to complete the cycle otherwise. Another example could be robot fingers where high gripping forces are required along with quick travel before the contact. Both high speed at low or no load operation and high force capability are desired at joint actuators of such robots but it is difficult to obtain when the reduction ratio is constant [5]. Furthermore, series elastic actuators are also popular for these applications because of their enhanced force/torque control capability and low impedance [10],[11]. However, there is no such bidirectional mechanism having the characteristics of both series elastic actuation and load-sensitive CVT, to our knowledge. Therefore, we believe that the proposed mechanism bridges this gap.

The rest of this paper is organized as follows: In Sec. II, application characteristics and the need for variable transmission are detailed. In Sec. III, the proposed load-sensitive CVT mechanism is introduced, and its working principles are explained. In Sec. IV, simulation results are given to demonstrate the potential of the proposed mechanism. In the last section, important aspects of the work are highlighted, and possible future work is briefly presented.

II. Application Fundamentals

The CAS generally does not need to perform a complete rotation [3]. CS rotation is limited to $\pm\beta°$. The parameter β generally takes values between 15° and 40°. Its exact value is determined by the system solution.

Electrical motors have certain specifications such as nominal speed/torque, maximum speed/torque, and nominal power. They operate most efficiently and run for a long period of time without a failure at their nominal operating points [12]. Maximum/stall torque rating is much higher than the nominal; however the motor can only intermittently operate at that maximum level due to thermal limitations.

For the CAS, instantaneous required speed is the derivative of CS deflection commands, and the load is induced by aerodynamic forces. Given these, different torque and speed combinations may be demanded by the autopilot of the GM during operation so CAS applications require output torque and speed that vary in a range. A rough generalization can be made by assuming that CS velocity is inversely proportional to the load which acts in the opposite direction of CS deflection. Data for an example GM are shown in Fig. 1. Here, it can be observed that data lie under a constant power curve.

When there is a constant ratio transmission between the motor and the CS, the motor needs to be chosen such that it will meet both maximum torque and maximum speed requirements separately. As explained in the above paragraph, a motor that has a nominal torque rating greater than or equal to maximum required load torque has to be selected [4]. Additionally, the nominal speed of motors is close to their maximum speed. This results in a selection of a larger motor that has a power rating higher than what application requires. On the other hand, an ideal CVT satisfies that the reduction varies in such a way that different output torque and speed requirements are met while operating the motor at its nominal torque constantly during the operation. Thereby, a motor whose nominal power is equal to power required by the application can be selected, resulting in a smaller motor that operates more efficiently. Regarding the space limitation in GM CAS, this yields an important contribution.

In Fig. 1, torque/speed curves associated with motors selected for constant and variable reduction ratio cases are given for an example case. Specifically, the CAS must be capable of providing 7.5 Nm output torque at stall and 180°/s at nearly no-load. In the constant ratio case, a motor with stall torque of 7.5 Nm and no-load speed of 180°/s at the CAS output would be impracticable since it will burn out when it operates at its stall torque to overcome 7.5 Nm for a very short duration. Thus, a proper motor selection dictates that the motor's nominal torque should be greater than 7.5 Nm independent of how small the required speed is. This leads to an overpowered motor choice since the speed at the motor's nominal torque is usually very close to the no-load speed. For this example, an ideal CVT allows a 5.5 W motor compared to 21.8 W one which in the constant ratio case (165°/s nominal speed at the nominal torque of 7.5 Nm).

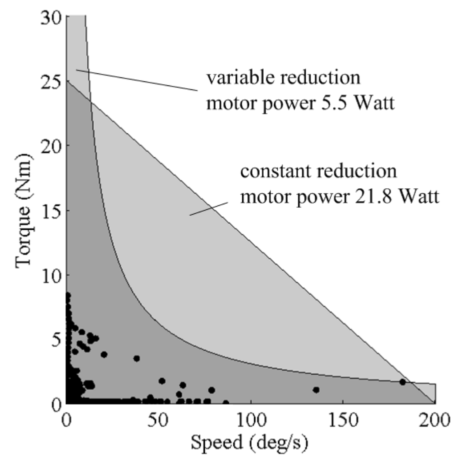

Figure 1. Example CS torque-speed values and required torque/speed curves of motors to drive this CS when there is constant and variable reduction.

III. Proposed Mechanism

The load-sensitive CVT to be introduced is obtained by modifying a well-known mechanism. This original mechanism is a popular one in CAS applications since some of its inherent qualities suit well with the geometrical requirements of the CASs, which are briefly explained in Section I. In the following subsections, firstly force/motion analysis of the original mechanism are given and discussed, and the proposed mechanism is presented then.

A. The original mechanism: Pin in slot joint

A pin in slot joint driven with a linear actuator, when used as a part of transmission, provides the desired right angle between the actuator and output as well as sufficient reduction capability. This actuator may be a linear motor or a combination of a rotary motor and a suitable transmission mechanism (e.g., a motor, gear head, and ball screw [13]). The force exerted on the pin is generated by the actuator and transmitted to the CS, which is mounted to the output shaft of the mechanism, through the slotted link. A schematic illustration of the mechanism is given in Fig. 2. Here, θ_{slot}, θ_{fin}, θ_{def}, and s denote the slot position with respect to the actuator, fin deflection with respect to missile axis, torsional deflection of output shaft, and linear actuation displacement, respectively. From geometry, angles are related as $\theta_{fin} = \theta_{slot} - \theta_{def}$. Based on this figure, we show steps for the mathematical derivation of the motion/torque transmission in the rest of this subsection.

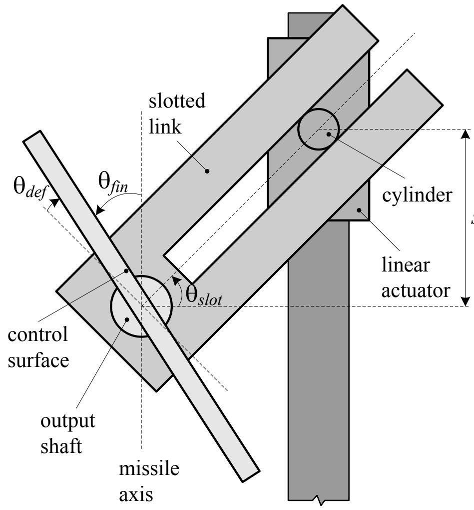

Figure 2. A schematic representation of pin in slot mechanism.

For neither ball screws nor linear electrical motor, self-locking is an issue. Their designs maintain backdrivability. Thus, disregarding issues related to self-locking, Fig. 3 presents free body diagrams of the slotted link and the pin where F_{la} denotes the force generated by the actuator, F denotes the contact force between the pin and slot, R_x and R_y denote reaction forces from the output shaft, R_{la} denotes the radial force on the linear actuator, and l_0 denotes the distance between axes of output shaft and the actuator.

Moment balance about the output shaft axis of the slotted link and force balance in y direction are written to derive the input-output force-torque relation of the mechanism. Moment and force balances are given in the equations below:

$$\begin{aligned} F\ l_0 / \cos\theta_{slot} - M_h &= 0 \\ F_{la} - F\cos\theta_{slot} &= 0 \end{aligned} \tag{1}$$

Solving (1), the following relation between F_{la} and M_h can be simply obtained as

$$M_h = \frac{F_{la} \cdot l_0}{\left(\cos\theta_{slot}\right)^2}. \tag{2}$$

Equation (2) presents a unique characteristic of the pin in slot mechanism (i.e., dependency of the reduction ratio on the slotted link angular position). It can be found as

$$r = M_h / \left(F_{la} \cdot l_0\right) = 1 / \cos(\theta_{slot})^2, \tag{3}$$

where r denotes the reduction ratio in question. According to this, r can be set to infinity. However, this theoretical upper bound cannot be realized since the mechanism approaches the singularity that occurs at $\theta_{slot} = 90°$. In this regard, we set the maximum permissible range for θ_{slot} to ±75° in this work to avoid the singularity. The effect of θ_{slot} on r is shown in Fig 4. As seen from this curve, r becomes heavily dependent on θ_{slot} as θ_{slot} increases.

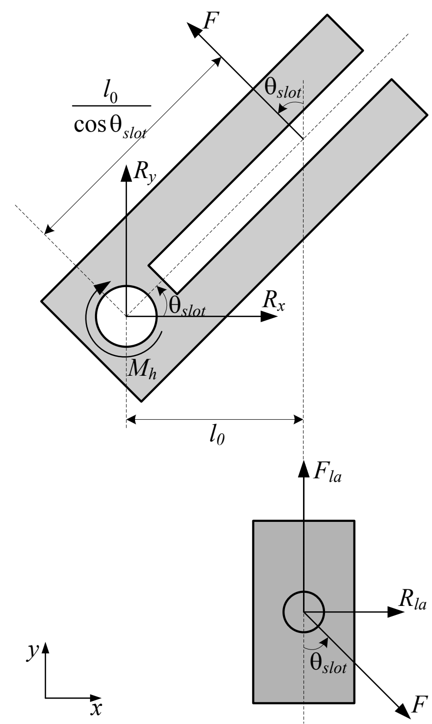

Figure 3. Free body diagrams of the slotted link and pin.

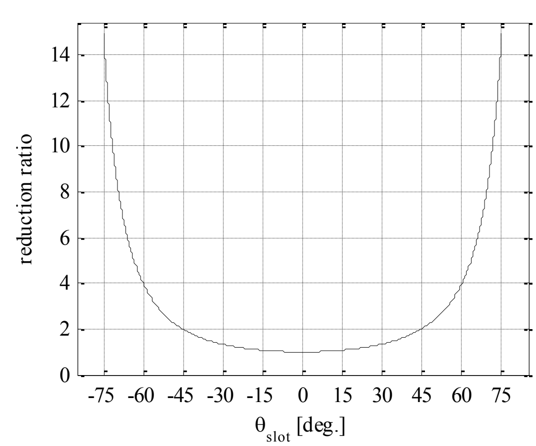

Figure 4. Variation of r with respect to θ_{slot}.

B. Elastic output shaft

As seen in Fig. 4, r increases with θ_{slot}. In this context, if θ_{slot} can be increased with the load, then a load-sensitive CVT is obtained. Assuming that the load acts in the opposite direction of θ_{slot}, we propose that this can be achieved by using an elastic output shaft. As θ_{def} depends on the load by Hooke's Law, resulting CVT will be load-sensitive. Assuming that inertial torque of the CS is negligible compared to the load, this can be mathematically expressed as

$$M_h = k \cdot \theta_{def} = \frac{F_{la} \cdot l_0}{\left(\cos\left(\theta_{fin} + \theta_{def}\right)\right)^2}, \quad (4)$$

where k represents torsional stiffness of the CS shaft and is dependent on shaft geometry and material [14]. With this utilization of the elastic output shaft, not only do we come up with a CVT, but the pin in slot-based CAS also becomes a series elastic actuator. This secondary feature is actually inevitable to transform the original mechanism into a CVT. Furthermore, the proposed mechanism differs from the notable ones of the existing CVTs such as [5, 6] in that it can work in both directions.

A torsionally elastic and high strength shaft is needed to effectively realize CVT. For high load sensitivity, θ_{def} should increase significantly with load. On the other hand, the shaft needs to withstand the output torque without yielding. Such a shaft is constructed in [14] by machining the shaft made of widely available materials to a special geometry.

IV. Simulations and Results

We consider two cases for simulation purposes: a general missile CAS example and a robotic finger application. Both cases require controlled motion of θ_{fin} according to commands. We utilize a gain-scheduled state-feedback controller to track θ_{fin} commands since the proposed system has nonlinear dynamics, and the nonlinearity increases with θ_{slot}. It is assumed that states $\dot{\theta}_{slot}$ and $\dot{\theta}_{fin}$ are numerically computed using states θ_{slot} and θ_{fin} measured by position sensors (e.g., absolute encoders). Thus, the controller ensures that control surfaces are positioned with respect to autopilot commands no matter what θ_{slot} is. However, for space considerations, we do not present the derivation of equations of motion and skip the design details of θ_{fin} controller in this paper. Interested reader can find a concise treatment of design of gain-scheduled state-feedback controllers in [15]. To compare the performance of the CVT, we run the same simulations for the original mechanism, which has the rigid shaft representing the constant r case, also. In addition to the system dynamics practical issues like friction and backlash are also modeled in the simulations.

For the first simulation case, a low-power missile CAS application is considered. For this scenario, the maximum load on the CAS is 1.5 Nm, and β is 15. Corresponding load, to be applied to the CAS, and θ_{fin} commands, to be realized by the CAS in this scenario, are generated with respect to data collected from actual missile test cases. Thus, they include effects of turbulent load conditions and highly dynamic autopilot needs. The parameter l_0 is chosen to be 0.023 m. The linear actuator considered here is composed of a rotary motor, a planetary gearbox, and a ball screw. Inertia of the motor, the gearbox ratio, and pitch of the ball screw are chosen to be 0.5 kg mm^2, 3.7:1, and 3 mm, respectively. These values are chosen regarding the mechanical power need of the application. The parameter J_{cs} is chosen to be 170.0 kg mm^2. Additionally, desired θ_{def} at the maximum load is determined as 45° in the absolute sense meaning that k is 1.91 Nm/rad. To ensure that a shaft that is elastic enough is physically possible, we have followed the method presented in [14] for design. For the defined requirements, the shaft with the following properties is found to be conceivable: The shaft should have plus shaped cross section with width 18mm, web thickness 1mm and effective length for torsion 83mm. A shaft with this geometry, if made from AL 7075 T6, is both strong enough to bear the torsion without yielding and elastic enough to assist CVT [16]. Simulation results are given in Figures 5, 6 and 7.

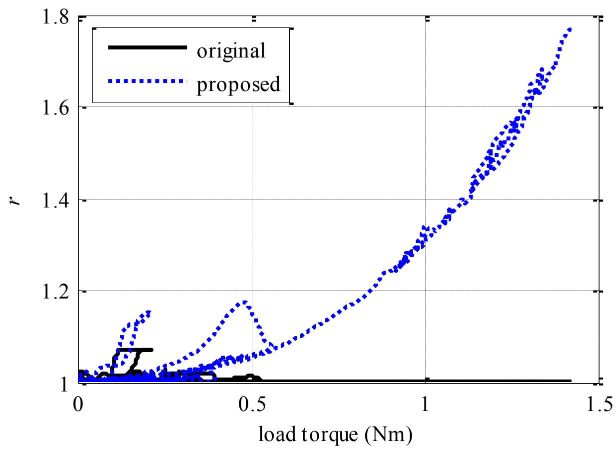

Figure 5. Case I. Reduction ratio of the CVT as a function of the load torque for the original and proposed CASs.

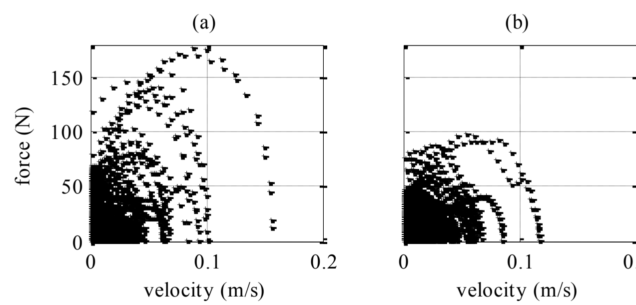

Figure 6. Case I. Resulting force/velocity pairs of the linear actuator for the (a) original and (b) proposed CASs.

In Fig. 5, it is observed that r increases with the load, as expected. Fig. 6 illustrates the required force/velocity operating points of the linear actuator. They must be analyzed carefully as the force/velocity pairs are an important measure for the selection of the motor. For a failure-free operation, the motor must be capable of operating continuously at every point. This means that the motor must be able to produce both the maximum force and velocity nominally as explained in Sec. II. These are presented in Table I. The difference in maximum force is consistent with the maximum reduction ratio of approximately 1.8 as seen in Fig. 5. On the other hand, maximum speed is decreased for the CVT as a result of deterioration in the bandwidth, details of which are discussed further. These results verify that we achieve our goal of selecting a smaller motor by means of the proposed load-sensitive CVT. As seen in Fig. 7, the CAS is able to follow θ_{fin} commands. Although the general performance is good, in short time intervals where commands change rapidly, we see that the performance degrades which can be seen in top inset of Fig. 7. As for the original mechanism, we do not present anything graphically but it responds to such sharp commands more quickly. Therefore, we conclude that the achievable bandwidth diminishes for the proposed CVT. If the achieved bandwidth is below the requirement of the CAS, the controller must be improved to satisfy the requirement.

TABLE I. ACTUATOR REQUIREMENTS FOR CASE I.

	Original (rigid shaft)	Proposed (elastic shaft)
Max.[a] force (N)	180	100
Max. speed (m/s)	0.15	0.12
Motor power (W)	27	15[b]

a. Max. is the abbreviation for maximum.

b. To avoid possible misinterpretation of decrease in motor power of the proposed mechanism, power of motors are computed using the speed of 0.15 m/s for both original and proposed mechanisms.

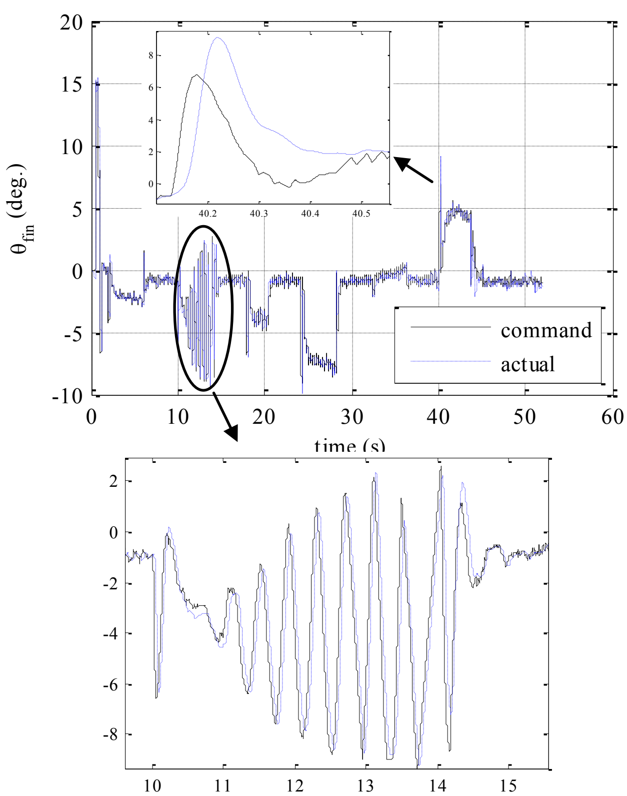

Figure 7. Case I. Overall command tracking performance of the proposed CAS.

For the second simulation case, a robotic finger joint is simulated based on data from [5]. The CAS performs 30° rotation under no load to reach a position where a load of maximum 100 N will be hanged on it. To simulate this load, a first-order filtered 4 Nm step load torque profile is applied to the CAS while it tries not to diverge from θ_{slot} = 30°. The parameter l_0 is chosen to be 0.030 m and J_{cs}=500.0 kg·mm^2. Other parameters are the same as the previous scenario. Additionally, desired θ_{def} at the maximum load is chosen as 45° in the absolute sense meaning that k is 5.10 Nm/rad. The shaft has the same material and geometry as the first simulation case except width increases to 48 mm. Simulation results are given in Figures 8, 9, and 10.

Fig. 8 shows the increase in the r with load torque. Greater reduction ratios are achieved as load torque increases for the proposed mechanism, whereas r is almost constant for the original. Forces of the both mechanisms are shown in Fig. 9. Since peak forces produced in the beginning of the simulation are associated with the intermittent operation of the motor, we do not take them into account for sizing. Thus, the steady-state forces are 100 N and 20N for the original and proposed mechanisms, respectively. It is not given explicitly in a figure here but maximum speeds achieved during the simulation are the same for this scenario since both original and proposed mechanisms follow deflection commands accurately. Therefore, a motor with 80% lower power than the original mechanism's can meet the power requirements of the proposed CAS.

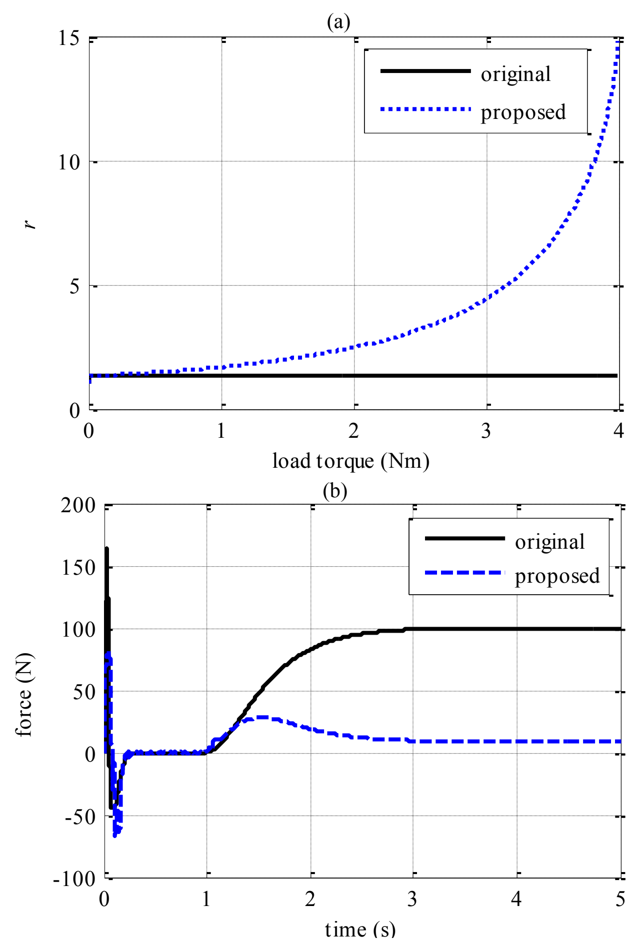

Figure 8. Case II. Reduction ratio (a) and force profile (b) of the original and proposed mechanisms.

In Fig. 10, θ_{fin} commands, response, and disturbance torque are illustrated. The CAS operates at no load for the first second. At the end of the first second, the object begins to apply a load, and actual θ_{fin} temporarily deviates from the command as a result. The magnitude of this deviation can be associated with disturbance rejection performance of the control loop. We believe that one can achieve a better deviation by a more sophisticated controller and improving its disturbance rejection performance (e.g., a disturbance observer and feedforward controller). Thus, we do not concern if the resulting tracking error is considerably large for a successful mission in this paper and leave it as a further work in order not to go beyond the scope of this paper.

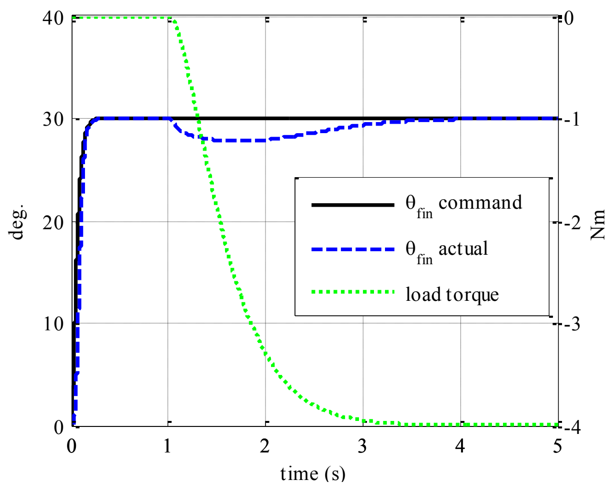

Figure 9. Case II. Command tracking performance of the proposed CAS and load torque profile.

V. Conclusion

In this paper, we have presented a novel actuation mechanism which is fundamentally a series elastic actuator working as a load-sensitive CVT. Physical attainability of the mechanical design have been considered especially for the torsionally elastic shaft. Advantages and effectiveness of the new mechanism are demonstrated by the simulations for an example missile CAS and a robot finger application.

The demonstrated advantages of the proposed mechanism are given as: 1) High output torque or speed values have been achieved separately with the same input force. 2) Use of a smaller excitation motor has been made possible for applications where load is inversely proportional with speed. 3) The proposed mechanism can produce a bidirectional force. 4) The need for an additional actuator has not arisen as reduction ratio is passively adjusted by the load. Although not investigated in this paper, there are additional advantages inherited from elastic actuator characteristic such as easier force control using shaft deflection information and low impedance. However, it has disadvantages of reduced bandwidth, higher nonlinearity in system dynamics, moderately increased mechanical complexity, and limited range of motion. By changing output shaft stiffness, achievable bandwidth and reduction ratio can be adjusted. Therefore, the proposed mechanism can be used in a variety of applications with a suitable compromise.

As a future work, a more advanced control strategy may be employed to minimize the reduction in bandwidth. Simulation results may be supported with experimental data, and new torsionally elastic shaft designs may be investigated to increase reduction ratio range for the CVT without increasing its size.

Acknowledgment

This study is conducted at Guidance and Control Design Department, Roketsan Missiles Industries. We would like to thank our department manager Mr. Bülent Semerci and team manager Dr. Burcu Dönmez for their encouragement and support.

References

[1] G. M. Siouris, *Missile Guidance and Control Systems*. Springer, 2004.

[2] P. Jackson, "Overview of Missile Flight Control Systems," *Johns Hopkins APL Technical Digest*, vol. 29, no. 1, pp. 9–24, 2010.

[3] G. D. Goldshine and G. T. Lacy, "High-response electromechanical control actuator," in *Proc.of the 3rd Aerospace Mechanisms Symposium*, Pasadena, CA, USA, May 1968, pp. 19–26.

[4] H. Yamada, "A Radial Crank-type continuously variable transmission driven by two ball screws," in *Proc. of the 2012 IEEE Int. Conf. on Robotics and Automation*, Saint Paul, MN, USA, May 2012, pp. 1982–1987.

[5] T. Takaki and T. Omata, "Load-sensitive continuously variable transmission for robot hands," in *Proc. of the 2004 IEEE Int. Conf. on Robotics and Automation*, New Orleans, LA, USA, Apr. 2004, pp. 3391–3396.

[6] T. Takaki and T. Omata, "100g-100N finger joint with load-sensitive continuously variable transmission," in *Proc. of the 2006 IEEE Int. Conf. on Robotics and Automation,* Orlanda, FL, USA, May 2006, pp. 976–981.

[7] S. Hirose, C. Tibbetts, and T. Hagiwara "Development of X-screw: a load sensitive actuator incorporating a variable transmission," in *Proc. of the 1999 IEEE Int. Conf. on Robotics and Automation*, Detroit, MI, USA, May 1999, pp. 193–199.

[8] J. Ingvast, "The PVT, an elastic conservative transmission," *The Int. J. of Robotics Research*, vol. 25, no. 10, pp. 1013–1032, Oct. 2006.

[9] G. Pratt and M. M. Williamson, "Series elastic actuators," in *Proc. of the 1995 IEEE/RSJ Int. Conf. on Intelligent Robots and Systems. Human Robot Interaction and Cooperative Robots*, Pittsburgh, PA, USA, Aug. 1995, vol. 1, pp. 399–406.

[10] J. E. Pratt and B. T. Krupp, "Series elastic actuators for legged robots," in *Proc. of the SPIE. Unmanned Ground Vehicle Technology VI*, Orlando, FL, USA, Apr. 2004, vol. 5422.

[11] D. W. Robinson, J. E. Pratt, D. J. Paluska, and G. A. Pratt, "Series elastic actuator development for a biomimetic walking robot," in *Proc. of the 1999 IEEE/ASME Int. Conf. on Advanced Intelligent Mechatronics*, Atlanta, GA, USA, Sep. 1999, pp. 561–568.

[12] U. Kafader, *The Selection of High-Precision Microdrives.* Sachseln, Switzerland: Maxon Academy, 2006

[13] M. J. Jang, K. C. Lin, and C. L. Chen, "Modeling and positioning control of a ball screw driven stage," in *Proc. of the 2004 IEEE Int. Conf. on Networking, Sensing and Control,* Taipei, Taiwan, Mar. 2004, vol. 2. pp. 943–948.

[14] M. M. Williamson, "Series Elastic Actuators," MS. Thesis, Dept. of Electrical Eng. and Computer Sci., Massachusetts Institute of Technology, Cambridge, MA, USA, 1995.

[15] J. W. Rugh, "Analytical framework for gain scheduling," *IEEE Control Systems,* vol. 11, no. 1. pp. 79–84, Jan. 1991.

[16] W. D. Jr. Callister, *Materials Science and Engineering: An Introduction*. 7th ed., New York: John Wiley & Sons Inc., 2007, Appendix B pp. A3-A30.