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Abstract

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A Highly Stiffness-Adjustable Robot Leg for Enhancing Locomotive Performance

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Abstract

Of the recent developments in legged robotics, compliance control of legs has potential to make a huge impact in the transition from simple mechanisms to life-like machines. Given the nature of legged locomotion in humans and animals involves such compliance control, this is a step forward in terms of the artificial embodiment of the biological traits that make these living beings superior to the robots we have today. Taking a literal Rolling Spring Loaded Inverted Pendulum (R-SLIP) morphology, this work proposes a magnetorheological-fluid-centric (MRF) variable stiffness leg, aiming to further investigate this area and take advantage of the improved energy efficiency and gait stability made possible through leg stiffness control. With the potential for adaptive tunability of leg stiffness through semi-active control, the design is shown to be capable of a maximum stiffness increase of 257%, with behaviour predicted by a force model describing the mechanism. In subsequent dynamic locomotion testing was conducted and the testing results demonstrate that the new leg is able to reduce cost-of-transport (CoT) and verified the potential of the MRF leg on improving the locomotive performance.

Keywords: Locomotion, Robot Leg, Magnetorheological Fluid, Variable Stiffness

1. Introduction

With a likeness to running animals, starting with Raibert's hoppers [1], compliant-legged robots have been developed to capture the behaviour of their biological counterparts. Just as many animals do, while these legged machines locomote, they convert gravitational potential and kinetic energy into strain energy through either relatively simple mechanisms, or otherwise more complex structures, with this energy being subsequently returned at the latter-half of each step. To describe this dynamic behaviour, the Spring Loaded Inverted Pendulum (SLIP) model, proposed by Blickhan [2] is often used in its original or a modified form. Serving not only as a design tool, this model has highlighted the important link between ground stiffness and leg stiffness [3], considering animals or legged robots as simplified spring-mass systems able to locomote with resonant 'stride' frequencies for maximized energy efficiency. Provided leg stiffness can be tuned to optimally suit given terrain conditions, such systems would yield versatile energy efficient locomotion [4]. Further, as evidenced in human and animal locomotion [3, 5, 6], given non-ideal terrain conditions would otherwise serve as limitations, modulation of leg stiffness can be used to circumvent disturbances and maintain body stability.

While previous leg designs presented in literature were typically of passive compliance, such as those in SCOUT [7], Whigs [8], and Tekken [9], in recent years leg stiffness control has become a heated research area. An example of note, designed by Galloway et al. is a C-shaped leg with structural stiffness variation using a worm-gear driven slider mechanism [10]. Another design using a worm-gear drive is L-MESTRAN, a variable stiffness hopper, designed by Vu et al. [11]. With a spiral-shaped foot, the robot platform Dynarobin developed by Mutka et al. [12] also offers variable compliance through adjustment of the foot-ground contact, i.e. beam-deflection point. An important feature of these and similar designs is the rate at which they can be tuned to reach a desired stiffness. While the last design mentioned here is suited to rapid stiffness change between steps, longer tuning times of the previous two may limit abilities in rejecting disturbances such as those encountered when transitioning between ground materials. Amongst other aspects, this tuning time requirement shows exciting new prospects may exist for magnetorheological (MR) materials in legged robot locomotion.

MR-fluid (MRF) is a smart material that possesses the ability to have its rheological properties altered very rapidly when a magnetic field is applied. Known as the MR effect, magnetic flux through the MRF causes micro-scale ferromagnetic particles suspended in a

non-magnetic host liquid to align with magnetic field lines forming chain-like structures [13]. Typically, this material is used in a post-yield regime in which constant yielding leads the MR effect to cause greater energy dissipation, i.e. increased damping [14, 15]. While designs utilizing MRF for as a variable stiffness element in the (low-strain) pre-yield regime are possible [16, 17], this is typically less practical than using MR-elastomer (MRE) which provides variable stiffness under higher strain amplitudes. This point notwithstanding, both materials can enable rapid leg stiffness adjustment in the order of milliseconds [14] through smart design, enabling semi-active control with minimal tuning time.

Regarding typical uses, MR materials have seen widespread use in the fields of noise suppression, dynamic control, and vibration [18, 19]. While active forcing elements or otherwise passive structures have more extensive and common use in these areas, there is a compromise between the versatility of active elements and the often more reliable, energy efficient passive alternatives. For this reason, semi-active materials such as MRF with controllable mechanical properties may be suitable alternatives. This reasoning is often cited as an important consideration leading to the now common linear MR dampers and shock absorbers, typically designed for vehicle suspension [20-22]. Recently MR materials have also made their debut in legged robotics, with an MR-damper-based leg, adaptively tuneable to suit different running conditions [23]. Additionally, a novel design placing an MRE spring into a C-shaped leg joint with a plus-or-minus stiffness of 48% variation is presented in [24].

Following recent developments, this work utilizes a variable-stiffness-enabled MR damper design similar to the linear shock absorber presented in [22], albeit with a rotary design. Like our previously designed MRE leg, this design also uses a literal R-SLIP (Rolling Spring Loaded Inverted Pendulum) [25] morphology, with the joint mechanism on the arc of the C-shaped leg near the hip, replacing the torsional spring of the model. Through control of the electromagnetic coils of the MR damper in this dual-spring design, a secondary torsional spring is engaged, providing simple current control of leg stiffness. The advantages of this new design over our previous design are much higher stiffness variation range and lighter leg weight. Owing to its unique properties, the proposed design has large potential to improve locomotive performance in legged robots through disturbance rejection and resonant running, improving energy efficiency, i.e. reducing cost-of-transport (CoT).

The remainder of this article is organized as follows. Section 2 outlines the design and working mechanism of the leg and leg joint. Section 3 details the experimental static force characterization and model fitting of the leg. With the static performance of the leg

summarized, Section 4 then includes dynamic locomotion testing of a pair of the legs. This work is lastly summarized and concluded in Section 5, with possible future work indicated.

2. Design and working mechanism of the MRF-based Leg

2.1 Leg Structure

Illustrated in Fig. 1 is the R-SLIP-based design concept for the leg, suited to a torque-actuated robot platform. Apparent in this figure is the C-shaped leg structure, including the variable stiffness joint mechanism at a position 60° from the hip on the arc of the 60 mm leg radius, in a position theorized to provide optimal stability for such legs [26]. The upper leg section is coupled to the lower through an always-active ‘soft’ spring of stiffness 1.56 N·m/rad, pre-tensioned by 90° rotation. A rotary MR damper housed in the lower leg section has its output shaft linked to the upper leg section through a thereby controllable ‘stiff’ secondary spring of stiffness 4.06 N·m/rad. In the extreme ‘free-spinning’ case for the MR damper, that is, when damping is low due to zero coil current, the leg will have a ‘soft’ single spring stiffness. At the other extreme under a large input current, the damping torque will be large, ‘locking’ the damper and engaging the secondary spring, giving a far stiffer parallel arrangement.

Other features include a locking clamp, constraining the MR damper assembly to the lower leg section, also setting the zero-position of the upper. Additionally, a silicone rubber foot-pad is used to improve ground friction as the leg sections shown are 3D printed high density ABS plastic. To make the design more robust however, other non-metal parts of the assembly are printed in high density nylon. All inclusive, the leg has a total weight of 657 g.

Fig. 1 MRF-based variable stiffness leg structure; (a) side view, (b) back view

Regarding some of the equations describing the static forces generated in the leg, the R-SLIP nature of the design allows the equations presented in [25] to be applied. With the parameters of the R-SLIP model outlined in Fig. 2, taking guidance from [24], the following relationship can be used:

$$k = \frac{F}{(l_0 - l_b)}, \quad (1)$$

where k is vertical leg stiffness, F is the force exerted on the ground by the leg: the ‘leg force’, $l_0 = 2r$ is the free-length of the virtual linear spring between the hip (loading point of the upper leg segment) and ground contact point, and l_b is the loaded spring length. To determine l_b for a given contact angle, it may be found using:

$$l_b = \sqrt{l^2 + l_a^2 - 2(l)(l_a)\cos(\phi)}, \quad (2)$$

where l is the length of the rigid link between the torsional joint and the hip, $l_a = \sqrt{l_0^2 - l^2}$ is the chord drawn from the joint to the contact point, and ϕ is the angle subtended by the hip and the contact point, given by:

$$\phi = \phi_0 - \psi = \frac{\pi}{2} - \psi. \quad (3)$$

The initial angle formed ϕ_0 is equal to $\pi/2$ for a 0° contact angle, and ψ is the joint deflection angle which loads the torsional joint. More applicable to linear force testing, the deflection angle of the joint can also be described in terms of vertical leg displacement y :

$$\psi = \phi_0 - \frac{\xi}{2} - \sin^{-1} \left(\frac{y}{l_a} + \sin \left(\phi_0 - \frac{\xi}{2} \right) \right), \quad (4)$$

where ξ is the position angle of the joint, spanning CCW from the hip.

As the key result of these formulae, the effective torsional stiffness of the joint can be transposed into a linear leg stiffness through:

$$k = \frac{l_b \left(\frac{\pi}{2} - \phi \right)}{(l_0 - l_b)(l)(l_a) \sin(\phi)} \times k_t, \quad (5)$$

where k_t is the effective torsional stiffness of the joint.

Fig. 2 R-SLIP static model parameters

From the result of equation 5, with platform parameters known, a leg stiffness based on the mice-to-horse locomotion scaling work of Heglund et al. in [27] may be selected, or in this case a design stiffness range for the variable stiffness joint may be determined. As is now typical of similar design approaches, in this work a 10% leg compression level [25, 28] was considered. Based on the 5.9 kg two-legged torque actuated platform used in this work, a range of 0.5 kN/m to 2.0 kN/m was chosen.

2.2 Design of the Rotary MR Damper

Illustrated through the CAD model of Fig. 3 is the rotary MR damper which enables the control input to this leg. As shown, the damper is quite compact with a maximum diameter of 47 mm and a body length of 38 mm. The MRF (MRF-140CG, produced by LORD Corp.) in the joint surrounds a rotor adjacent to two electromagnetic coils wired in series. These coils of 120 turns each provide magnetic flux through the magnetic circuit made of high-permeability low carbon steel when current is supplied. The MRF in the 0.5 mm wide radial gaps inside and outside of the rotor drum is then subject to an increased yield stress and viscosity, increasing the braking torque between the aluminium (Al) shaft and outer-body of the damper. The reason for the use of this high density MRF (MRF-140CG) was to maximise output torque; to verify the capacity of this material to increase its shear stress under an applied magnetic field, rheometry was performed (with an Anton Paar MCR301/MRD180 rheometer), resulting in the data presented in Fig. 4. From the figure, the hysteretic nature of the material can be observed. Regarding the behaviour of the ‘T-shaped’ drum, this design essentially acts as a normal drum-type damper, however with a larger active area of MRF and less unnecessary geometry, reducing weight. Furthermore, this design trades the requirement of the large radial area needed in disc brakes for a more manageable increase in axial length of the damper, more suitable for compact robot legs.

Fig. 3 Compact rotary MR damper

As for the considerations that went into the design of the damper, a 2D axisymmetric field study was developed in FEM software COMSOL Multiphysics ver. 5.1. Within this

simulation, the MRF was defined by the B-H curve LORD Corp. provided in the material data sheet for MRF-140CG [29], and the steel of the magnetic circuit was set to possess a relative permeability μ_r of 300. All other materials as detailed in Fig. 3 were defined by the built-in properties of the software package. Through iteration, the geometry of the joint was revised to meet an optimal trade-off between weight and magnetic flux penetrating the MRF. Plotted in Fig. 5(a) is the simulated mean flux through the inner and outer gaps of the damper under the application of a 0 to 1 A current range, with these regions highlighted in the model shown in Fig. 5(b). Within this result, it is seen that under a 1 A current, the maximum flux through the MRF achieved is 0.54 T in the inner gap, along with 0.28 T in the outer gap. As the magnetic flux through the damper tends to saturate beyond 1 A of current and the current carrying capability of the coil with 0.5mm wire diameter is 1A, the maximum operating current of the damper will be set as 1 A. Regarding the seemingly large deviation amongst these flux densities of the two gaps, this is primarily due to the inverse proportionality between flux density and area, with the area increasing radially outwards.

Fig. 4 Hysteretic behaviour of LORD MRF-140CG (10% strain, 5 Hz)

Fig. 5 Magnetic field FEM analysis: (a) mean flux through MRF, (b) joint model

2.3 Working Mechanism

As previously mentioned, the leg design of this work utilizes two springs, the softer of the two with a relatively low torsional stiffness of $k_l = 1.5 \text{ N}\cdot\text{m}/\text{rad}$, and the other with a higher stiffness $k_h = 4 \text{ N}\cdot\text{m}/\text{rad}$. Described in the diagram of this mechanism, Fig. 6, while the softer spring represented by k_l always couples the upper and lower leg sections, the stiffer spring k_h acts through the MR damper. It can be inferred that depending on the relative angular displacement about the damper, i.e. $(\psi_2 - \psi_1)$, will govern whether the output torque T is produced primarily by k_l , or comes from the parallel arrangement with k_h and the MR damper in series. If it is assumed that internal friction and viscous damping in the mechanism are negligible, the result simplifies and two extremes may be considered: if no current is input to the MR damper, $\psi_1 = 0$, and the joint stiffness will be equal to k_l ; if a large current is supplied to the damper, causing the internal shaft to lock to the outer body, $\psi_1 = \psi_2$ and the joint stiffness is the sum $(k_l + k_h)$.

Fig. 6 Variable stiffness MR damper mechanism

To formalize the behaviour of the mechanism, describing the leg force in terms of displacement and input current to the damper, a theoretical analysis on its force displacement relationship has been conducted in conjunction with modification of previously provided R-SLIP equations. From the FEM result provided in Fig. 5(a), the mean flux through the inner and outer gaps of the MR damper, B_i and B_o respectively, can be approximated by the following quadratic equations within a 0 to 1 A input current I range, starting with the inner flux:

$$B_i(I) = -0.0691I^2 + 0.5950I \quad [\text{T}], \quad (7)$$

and then the outer flux:

$$B_o(I) = -0.0358I^2 + 0.3116I \quad [\text{T}]. \quad (8)$$

As provided by LORD Corp., the B-H and shear yield stress relationships for the MRF used (MRF-140CG) can be found in [29]. Through curve fitting, the magnetic field strength H can be described accurately in terms of the flux density B by the following cubic:

$$H(B) = 60.7433B^3 + 105.8471B^2 + 33.5748B \quad [\text{kA/m}]. \quad (9)$$

To then relate the magnetic field strength to the yield behaviour of the material, the shear yield stress τ_y can be expressed in terms of H with a quadratic fit:

$$\tau_y(H) = -1.3835H^2 + 567.2490H + 603.5291 \quad [\text{Pa}]. \quad (10)$$

Lastly, rotary MR damper design equations can be considered to convert this MRF yield stress into a static braking torque, modifying the drum-type brake equations provided in [30]. Considering a Bingham plastic model, the braking torque of the damper is the sum of two components, a Coulomb friction (pre-yield induced) torque T_τ , and a viscous rate-dependent damping T_η , i.e. $T_{MR \text{ damper}} = T_\tau + T_\eta$. Considering the angular velocity of the damper as zero, since we are interested in the pre-yield torque of the damper, also assuming T_η to be relatively small in the post-yield state, we may neglect the viscous-damping term and describe the joint torque through the following:

$$\begin{aligned} T_{MR,y}(\tau_y) &= 2\pi(\tau_{y,i}(H_i) \times w_i \times r_i^2 + \tau_{y,o}(H_o) \times w_o \times r_o^2) \\ &= 2\pi(\tau_{y,i}(H_i) \times 0.01 \times 0.015^2 \\ &\quad + \tau_{y,o}(H_o) \times 0.014 \times 0.02^2) \quad [\text{N} \cdot \text{m}], \end{aligned} \quad (11)$$

where w and r are the axial lengths and radii, respectively, of the inner and outer drum areas of the rotary damper. Also, note the two yield stresses of this equation are different; given the MRF flux is described by two equations, equations 7 and 8, two results must be obtained for equation 9 and consequently equation 10. Employing the set of equations 7 through 11 then, the yield torque of the MR damper can be estimated as a function of input current to the electromagnetic coils. It should also be noted that this torque model only considers unidirectional loading as the C-shaped nature of the leg results in ground contact only during the compression phase when locomoting. The leg is subsequently returned to its initial state in the aerial phase, aided by the ‘always active’ soft spring, during which the typical hysteretic behaviour of MRF occurs. Consequently, a simplified torque-displacement, and then force-displacement model for the leg can be established.

Now, with a means of determining the yield torque of the damper, conditions for its behaviour can be set based on whether the MRF has yielded or not, determining if the entire mechanism is in the stiff-state or soft-state. As it is the force relationship of the leg while locomoting we are interested in, the torque output of the mechanism described in Fig. 6 can be transposed into a leg force F using equations 1 through 5, giving the combined result:

$$F = \lambda \times T, \quad (12)$$

where λ can be considered as the reciprocal of the moment arm through which the force rotates the joint mechanism housed in the leg. This can be determined from:

$$\lambda = \frac{\sqrt{l^2 + l_a^2 - 2(l)(l_a) \cos(\phi_0 - \psi_2)}}{(l)(l_a) \sin(\phi_0 - \psi_2)}, \quad (13)$$

again, remembering ψ_2 is the angular deflection of the mechanism which, as per equation 4, is a function of leg geometry and vertical displacement.

With a relationship describing the leg force in terms of vertical deflection and input current, the following conditions may be used:

$$\text{If: } k_h \psi_2 < T_{MR,y}$$

$$\psi_1 = \psi_2, \quad (14)$$

and:

$$F = \lambda \times T = \lambda \times [(k_l + k_h) \psi_2]. \quad (15)$$

$$\text{If: } k_h \psi_2 \geq T_{MR,y}$$

$$\psi_1 = 0, \quad (16)$$

and:

$$F = \lambda \times T = \lambda \times [k_l \psi_2], \quad (17)$$

taking the yield-point of the damper as a reference for these two equations.

Following the above equations, the force displacement relationships under different currents can be calculated and are illustrated in Fig. 8.

3. Leg Characterization

The experimental test setup for the static force and stiffness characterization of the leg prototype is shown in Fig. 7. In this arrangement, rather than a robot platform, the leg is coupled to the top clamp of an MTS Landmark hydraulic test system (Load Frame Model: 370.02, MTS Systems Corp.), with the lower section of the leg (rubber foot-pad removed) free to slip on a low friction base affixed to a load transducer. In a similar fashion to the procedure followed in [24], through linear ramping of the servo-hydraulic actuator of the top clamp, the system supplied uniaxial loading to the vertically restrained leg set to a 0° initial contact angle. The time, force, and displacement data is simultaneously logged to a computer via a data acquisition (DAQ) board. To provide current to the coils of the MR damper inside the leg, a DC power supply (CPX400A, Aim-TTi Ltd.) was used.

Fig. 7 Static force testing experimental setup

Plotted in Fig. 8 is the experimental test result along with the theoretical calculation result, whereby the leg undergoes deflection with an 8 mm vertical displacement with currents

varied from 0 to 1 A between tests. As can be seen, under zero input current, the leg possesses a constant soft stiffness across the displacement range tested, with a peak force of 4.05 N. As current is increased, shown in the 0.25 A and 0.50 A test runs the leg is more stiff at lower displacement levels, and at some point, i.e. the damper's yield point, the slope decreases, again giving the soft stiffness state. As the turning-point at which the damper yields gradually increases with joint input current, it can be seen in the 1.00 A curve that the damper no longer yields under an 8 mm displacement level, giving a peak force of 14.14 N, representing an increase of 249.14% from 0 A. Providing validation for the theoretical calculation of the working mechanism, this behaviour is consistent with the description of the two stiffness states the leg provides, with conditions depending on the yield torque and angular displacement of the MR damper. Regarding the parameters used, some have already been incorporated into the equations of the model and are therefore implicit, however some others are included for reference in Table 1.

Fig. 8 Experimentally measured and model calculated leg force under varied current

Table 1 Leg force model parameters

Parameter	l	l_a	ξ	ϕ_0	k_l	k_h
Value	0.06 m	0.1039 m	$\pi/3$ rad	$\pi/2$ rad	1.56 N·m/rad	4.06 N·m/rad

Focusing more specifically on the effective leg stiffness of the leg as tested, this is included in Fig. 9 for the 0 to 1 A range tested, with numerical values provided in Table 2. As can be seen, the leg stiffness in the softest state under zero input current is 0.506 kN/m, consistent

with the low-end of the design stiffness level (0.5 kN/m). At the maximum, 1 A current level, with subtle saturation evident, the stiffness increased to 1.811 kN/m, just shy of the 2.0 kN/m design target, which is reasonable. The result presented here represents a maximum stiffness increase of 257.40%, which should allow the leg to adequately alter the dynamic performance of a robot platform for a modest range of test conditions.

Fig. 9 Relationship between current and leg effective linear stiffness

Table 2 Effective linear stiffness under different currents

	Applied current (A)				Increase from 0→1 A (%)
	0	0.25	0.50	1.00	
k_{eff} (N/m)	506.81	949.53	1348.30	1811.33	257.40

4. Locomotion Testing

In order to characterize the dynamic performance of the variable stiffness leg, a test rig was fabricated to facilitate simple two-dimensional running, with a two-legged test platform that is restrained to move only through the sagittal plane, translating both horizontally and vertically, restricting lateral motion and all rotational degrees of freedom. The test rig, shown in the CAD model of Fig. 10, has an aluminium extrusion frame, with a 20 mm ply base, providing an adequately rigid structure. The 5.9 kg robot platform follows vertical and horizontal rails upon which a guide carriage is mounted to restrict rotation of the platform

and house a laser used to measure vertical displacement (Keyence LB-11/LB-60; 80 mm to 120 mm range).

Fig. 10 Locomotion test rig model (dims in mm)

To measure horizontal displacement and calculate average forwarding velocity v_{avg} of the robot platform, a SICK DT60-P211B laser with a 0 to 5.3 m effective range is equipped to the rig. To actuate the legs of the platform, two AC synchronous motors (Panasonic 1.3 N·m MSMJO42G1U, with MBDKT2510CA1 200 V driver) are coupled to 10:1 planetary reduction gearboxes, giving an output torque of 13 N·m to the legs. These motors are also able to measure torque and can be used to determine average locomotive power consumption P_{avg} , such that the energy efficiency of locomotion, cost-of-transport CoT [23], can be found using:

$$CoT = \frac{P_{avg}}{mgv_{avg}}, \quad (18)$$

where $m = 7.2$ kg is the total platform mass and $g = 9.81$ m/s² is gravity.

To energize the MRF joints of the leg, a DC power supply is used, with varied currents set between tests. Seen in the test setup shown in Fig. 11, regarding the data acquisition and control, the system uses an NI myRIO-1990, programmed through NI LabVIEW, with a host computer to prepare tests and display live measurements.

Fig. 11 Experimental setup for legged locomotion testing

In the tests carried out here, a gait pattern typical of hexapods is employed, wherein for a designated contact range (set to 90°) the legs will rotate at a set-speed while they contact the ground, after which it is assumed they have lost contact, so they will then rapidly rotate (at 45 rad/s) until they return to take another step [31]. Additionally, to improve stability, the legs walk with a 45° phase difference (half of the contact range). During these tests, the platform is set to run a 1 m length, with measurements taken in a from 0.4 m to 0.7 m, considered as a safe range away from starting or stopping disturbances. This was done over a range of contact velocities for the legs, ranging from 0.5 rad/s to 15 rad/s, yielding the included values of CoT in Fig. 12 for the minimum possible stiffness of the legs (with 0 A input current), and the maximum stiffness (2 A, to ensure magnetic saturation). Each test was conducted numerous times to yield a more reliable result.

A key observation made here is that in the low speed range, i.e. speeds lower than about 5 rad/s, generally the cost-of-transport is lower for a *lower* leg stiffness, illustrating improved energy efficiency under this condition. The largest change in this speed range occurs at 1.5 rad/s with a mean reduction of 23.48% from a CoT of 6.09 to 4.66. Towards higher speeds above 8 rad/s, the trend is opposite, with a *higher* leg stiffness reducing the CoT . The maximum reduction here occurs at 10 rad/s from a CoT of 4.51 to 3.29, representing a 27.05% mean change. While this result is not necessarily suggestive of the theorized resonant running, what it does show in line with theory is that modulating leg stiffness in legged robots can improve energy efficiency while locomoting. Although more adept control strategies may yield better results, even a simple series of conditions based on leg speed could give this

platform improved gait performance, as evidenced by this result. Specifically, the MRF leg can perform soft stiffness when the running speed is less than about 5 rad/s and perform high stiffness when the speed is higher than 5 rad/s. With this control method the proposed MRF leg can achieve much lower CoT s than both constant stiff or soft legs.

Fig. 12 CoT of test platform equipped with MRF-based legs

5. Conclusions

Inspired by biological studies and the impeccable performance of legged creatures, the variable stiffness MRF-based legs presented in this study have proven to be an effective and novel solution to improve locomotive performance with regards to energy efficiency, evidenced through locomotion testing. Additionally, through experimental force characterization, the legs were shown to be capable of a maximum stiffness increase of 257.40%, with the result giving validation to the proposed force-displacement model describing the legs. As for future efforts in this area, the test rig will employ other ground materials to investigate leg performance while traversing more complex terrains.

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