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A Low-Friction Passive Fluid Transmission and Fluid-Tendon Soft Actuator

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Abstract— **We present a passive fluid transmission based on antagonist pairs of rolling diaphragm cylinders. The transmission fluid working volume is completely sealed, forming a closed, passive system, ensuring input-output symmetry and complete backdrivability. Rolling diaphragm-sealed cylinders provide leak-free operation without the stiction of a traditional sliding seal. Fluid pressure preloading allows for bidirectional operation and also serves to preload the gears or belts in the linear-to-rotary output coupler, eliminating system backlash end-to-end. A prototype transmission is built and tested for stiffness, bandwidth, and frictional properties using either air or water as working fluids. Torque transmission is smooth over the entire stroke and stiction is measured to be one percent of full-range torque or less. We also present a tendon-coupled design where the rolling diaphragm is inverted from its normal orientation; this design does not require shaft support bushings, tolerates misalignment, and can be made out of substantially soft materials. Actuator units and a passive transmission are demonstrated using this new soft cylinder design.**

I. INTRODUCTION

Electric motors are efficient and simple to control, but suffer from low torque density. Hydraulic actuators have much higher torque density [1], but the mass of the required valves, pumps, and accessories limit system-wide torque density. Hydraulic actuators and their flexible supply hoses constitute a fluid transmission system with high end-effector torque density, which allows for smaller limb mass and inertia and high-speed operation, in spite of the large overall system mass. If the transmission elements of a hydraulic system are combined with proximally located electric motors, we may be able to benefit from the advantages of both systems.

Valve-controlled hydraulic actuators and highly geared motors both suffer from high mechanical output impedance (i.e. they have high output friction, stiffness, and reflected inertia) and methods to reduce the overall effective joint impedance are often sought, by using either lightly geared motors [2], or closed-loop force feedback control [3]. Because of their low torque density, lightly geared motors are often too heavy to place inside serial-link robot limbs at distal joints. If flexible, efficient, and low profile mechanical transmissions are available, these motors can be placed inside the body of the robot via the transmission, reducing limb inertia and allowing high speed motion, high overall efficiency, and easy backdrivability. These configurations are especially attractive for robots designed to interact directly with humans, where limb lightness is an important safety factor, and high passive backdrivability is desirable for safe and simple force-mediated interaction.

Multi-link articulated cable drives running over low-friction pulleys and capstans offer perhaps the highest efficiency and smoothest operation among existing mechanical transmissions [2], [4]. However, such systems are complex and there is a practical limit to the number of cables that can be routed through the proximal joint, such as the shoulder. This type of transmission is highly integrated into the design of the robot limb. Transmissions that are mechanically separate from the arm, allow flexible routing, and have low intrinsic bending stiffness, can greatly simplify the design of the arm and allow for more joints to be routed to the base.

Bowden cables (e.g. bicycle cable brakes) use a tension cable running inside a flexible compression housing [5]. They are flexible and provide high work density per cycle, but suffer from high static friction, wear, and nonlinear behavior [6]. In particular, the friction in a Bowden cable increases exponentially with the total bend angle. Using fluid actuators in a passive hydrostatic configuration with either low-friction linear cylinders [7] or reversible rotary fluid pumps [8], has been studied as an alternative. Closed loop control is required to combat fluid leakage or bypass flow to maintain input-output synchronization, but these fluid-based transmissions offer greater routing flexibility than cable drives and Bowden cable transmissions.

We propose a passive, fluid-based transmission, offering the torque density of fluid actuators, without the complexity and high impedance of servo valves and pumps. Our approach is to use rolling diaphragm cylinders to form the closed volumes of fluid necessary. Rolling diaphragms are tube-like reinforced rubber seals that roll from bore to piston instead of sliding. These cylinders suffer from hysteresis due to the bending and unbending of the rubber, but avoid the leakage and high static friction that have challenged previous efforts to develop high-performance hydrostatic transmissions [7], [9].

In this paper, we present the rolling diaphragm concept and propose a passive hydrostatic transmission which leverages its unique properties. We have also designed a new soft fluid actuator based on a reverse-acting rolling diaphragm. The behavior and performance of this new concept is presented qualitatively in the video accompanying this paper. We have constructed a prototype transmission using off-the-shelf components; it is tested passively by measuring work loops under manual manipulation and tested under motor-drive

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Fig. 1. (color) This diaphragm is 23 mm in diameter at the bead-end and 22 mm tall (Control Air 346-700-002). It is made of nitrile rubber, reinforced with woven polyester fabric. The diaphragm is also shown after a piston is pressed into the top and the diaphragm is inverted back into itself.

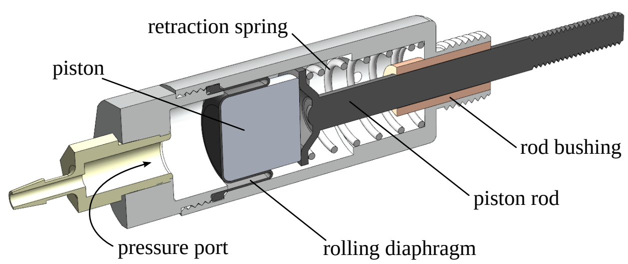

Fig. 2. (color) A rolling diaphragm cylinder is substantially similar to a traditional pneumatic or hydraulic cylinder, except for the diaphragm and a reduced size piston to allow for diaphragm convolution. These cylinders are usually single-action, an internal spring providing a retraction force.

for dynamic performance and haptic qualities, including force bandwidth and step response. We analyze these results to determine hysteresis, static friction, and stiffness when operated with both air and water. The accompanying video shows the transmission under motorized operation and during manual manipulation.

II. ROLLING DIAPHRAGM CYLINDERS

Hydraulic and pneumatic cylinders present an unavoidable tradeoff between seal friction and fluid leakage. For highly dynamic applications, pneumatics are preferred for their high speed and the acceptability of leaking air in exchange for low friction. Most fluid-actuator systems use a high pressure source and valves; these systems are completely non-backdrivable. In [7] a backdrivable passive hydraulic transmission was built with the goal of reducing seal friction, particularly seal stiction, as much as possible. Two cylinders were connected by flexible reinforced tubing to form a symmetric constant volume closed system. Special cylinders with precision cylinder bores and pistons were used without any seals, relying only on close tolerances. Leakage was very high and a make-up pump was used to maintain constant transmission volume. In actual use, a low pressure leakage capture-and-return system would be required, with position measurement at the input and output and closed-loop feedback to maintain constant fluid volume.

Rolling diaphragms are an alternative method of sealing a pneumatic or hydraulic cylinder [10]. A diaphragm shaped like a “top hat” is inverted upon itself and a piston fitted inside. Figure 1 shows the diaphragm used throughout this paper before installation and installed over a piston. Figure 2 shows the basic design of a single-acting rolling diaphragm cylinder. As the piston moves in and out the diaphragm rolls off the wall onto the piston and vice versa, much as a sock or sleeve may be pulled back on itself. Because there is no sliding motion, static friction is very low. If the walls of the diaphragm are straight or nearly straight, then there is no appreciable spring-rate throughout the stroke. The diaphragm is a fabric-reinforced elastomer and hysteresis from bending and unbending of the diaphragm at the convolution is expected.

In general, rolling diaphragms are low pressure seals, typically limited to 10 bar (150 psi). Off-the-shelf diaphragms rated up to 17 bar (250 psi) are available, but this is an order-of-magnitude lower than typical hydraulic system pressures of 100 to 200 bar (1500 to 3000 psi). Rolling diaphragms are manufactured by compression molding thin sheets of compounded rubber backed by high-tenacity finely-woven fabric. This process allows diaphragms to be economically made in parallel with multi-cavity molds. However, the fabric can only be drawn to a depth of about one bore diameter without excessive distortion and creasing of the fabric. Thus, commercially available rolling diaphragm cylinders have short strokes when compared to sliding-seal cylinders of equal bore. This restriction may explain their limited use in robotics [11], [12]. Long-stroke diaphragms made by individual layup or other methods have been proposed [13] but are not commercially available.

Rolling diaphragm cylinders are used for very sensitive dynamic applications such as web tensioning in roll-to-roll material processing and for automation environments that require absolute cleanliness and zero lubricants, such as pharmaceutical production and clean-room operations. The rolling diaphragm concept is also ubiquitous for air springs used in heavy vehicle air brakes and high-performance vibration isolation tables.

III. TRANSMISSION DESIGN

Rolling diaphragms cannot support reverse pressures because the convolution will invert and the diaphragm will jam; hydraulic cylinders are normally limited to no more than one atmosphere of reverse pressure by fluid cavitation. It is necessary to pre-pressurize the cylinders to one-half their maximum operating pressure by some external means to allow bidirectional operation. In [7] the (non-diaphragm) cylinders were preloaded by constant-force springs mounted to a linear guide; using a constant-force spring prevents the introduction of a spring rate to the transmission. This solution was workable, but non-ideal properties of the available constant force springs and integration with the linear guide added friction and complexity.

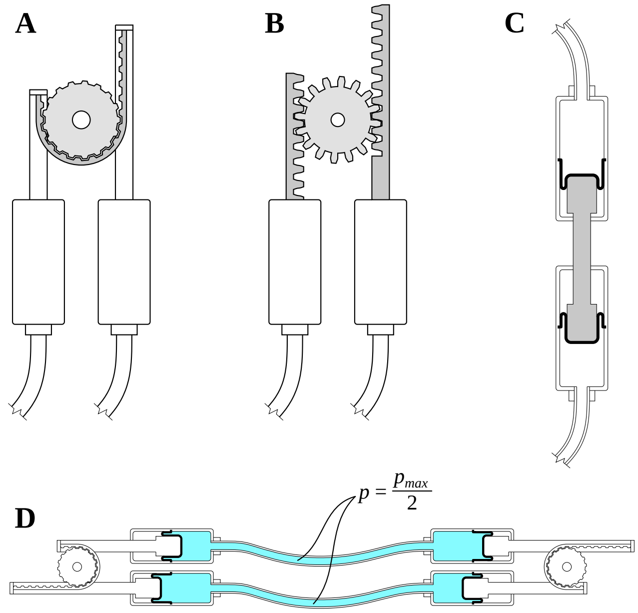

Fig. 3. (color) Cylinder pairs preloaded and balanced against one another via (**A**) timing belts, (**B**) rack and pinion gearing, or (**C**) direct opposition. Fluid-pressure preloading to one-half the maximum operating pressure eliminates backlash in the timing belt and gear balancers. These cylinder pairs are then connected (**D**) to form a closed-volume passive transmission.

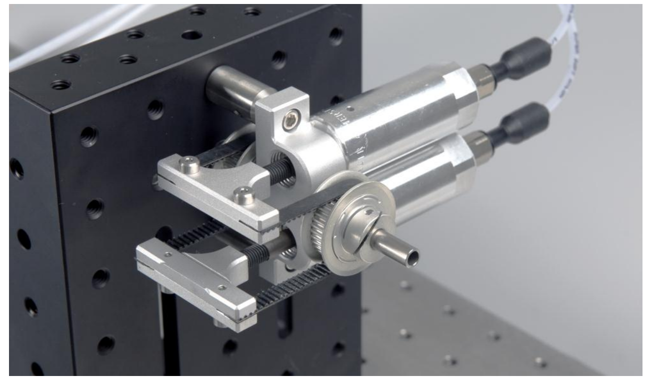

Fig. 4. (color) At each transmission end, a pair of cylinders are balanced against one another using timing belts and pulleys in the arrangement shown. The cylinders attach to a common mounting block which houses ball bearings for the output shaft.

A. Antagonist Passive Transmission

Our approach is to use pairs of cylinders pre-loaded against one another. This configuration ostensibly doubles the mass of the transmission, but fluid operation already affords us excellent force density and forgoing a tricky constant-force spring simplifies the profile and mechanics of the system. Figure 3 shows three proposed ways of balancing pairs of cylinders against one another. A complete four-cylinder transmission setup is also illustrated. The lines are pre-pressurized to half the maximum system pressure, preventing the diaphragms from ever being reverse-biased. In the case of options (**A**) and (**B**), this fluid pre-pressurization serves to form an anti-backlash configuration for the linear-to-rotary output coupler. For the rack-and-pinion design, it is possible to add an additional gear stage using symmetric half-width gears for each piston, mating with a full-width output gear, maintaining the anti-backlash configuration through the added gear stage. The third design (**C**) forms, essentially, a double-acting rolling diaphragm cylinder[1].

We constructed a prototype transmission using the timing belt design. One of the two identical ends of the transmission is shown in figure 4. We use pairs of 6 mm wide timing belts (2 mm pitch) to balance the cylinders pairs. This setup uses the smallest rolling diaphragm cylinders commercially available as stock items: Control Air 349-180-009 (alternatively Bellofram 908-034-000). The return springs were removed from the stock actuators. Stroke is approximately 17 mm and the effective piston area is 248 mm^2. These cylinders use a diaphragm rated to 8.6 bar (125 psi); maximum force is 214 N (48 lbf). To maximize angular range of motion given the short stroke, the smallest practical timing belt pulley pitch diameter, 25.5 mm, was chosen without having to angle the cylinders. This gives a maximum torque of 2.7 N-m over a 75° range of motion. A smaller pulley (and thus a larger range of motion) will require angling the cylinders to maintain belt alignment axial with the rod. Custom-made long-stroke diaphragms would also increase the range of motion and increase the overall work density per cycle of the transmission.

Each cylinder has a mass of 70 grams when filled with water. Two cylinders, tubing fittings, the mounting block, timing belts and pulleys, shaft, and bearings together have a mass of 246 grams.

B. Soft Actuator Version

The cross-section of a standard rolling diaphragm cylinder is shown in figure 5-**A**. Fluid pressure against the piston is balanced by an external compressive force transmitted through the cylinder rod. To prevent buckling, a guide bushing is required. It acts only to stabilize the rod, so if the cylinder components are well aligned axially, and the external load is applied axially, there should be little side load on the bushing and negligible static friction. Still, a stable configuration where the fluid pressure is balanced by an external tension will, in addition to eliminating the bushing, allow for tendon-type coupling, which is desirable for many configurations.

In figure 5-**B**, the orientation of the diaphragm has been reversed and the piston pulls on a cable tendon, rather than pushing on a rod. Now under tension, the cylinder is self-aligning and stable without a bushing. However, operation is now conducted by vacuum pressure; air operation will be hopelessly compliant, and liquid operation is limited to small negative pressures by fluid cavitation.

[1]Note that commercially available double-action rolling diaphragm cylinders actually use a shaft seal to allow the shared shaft to stick out of the end of the cylinder, compromising the low-stiction and fully-sealed advantages.

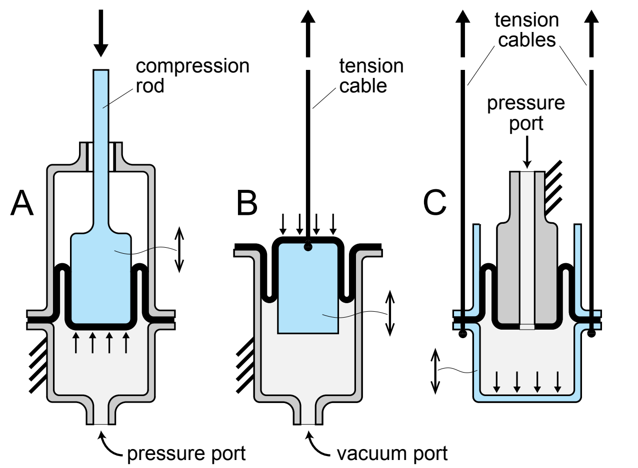

Fig. 5. (color) Three possible cylinder configurations; the moving portion is shown in light blue and the grounded portion in gray. (**A**) Classic rolling diaphragm cylinder; the fluid pressure is balanced by a compressive external force through the cylinder rod, which is stabilized against buckling by a guide bushing. (**B**) Here, the diaphragm is inverted from its normal orientation and vacuum pressure is balanced against an external force through a tension cable. This configuration is stable and self-aligning, so a guide bushing is not strictly required. (**C**) This configuration combines tension-balancing with pressure (rather than vacuum) operation. The piston is grounded and the cylinder is in motion.

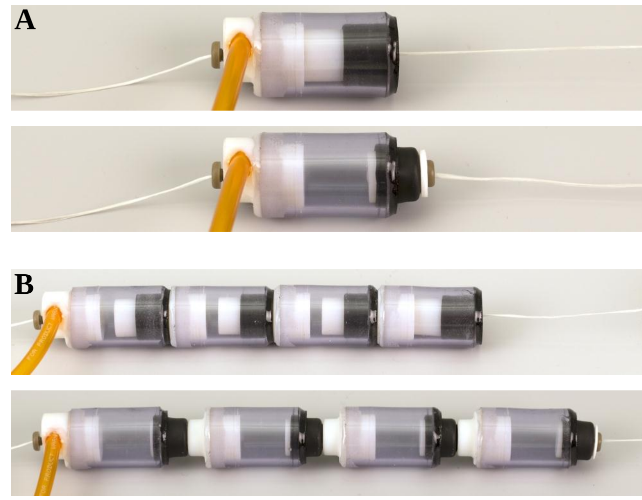

Fig. 6. (color) Soft, vacuum operated rolling diaphragm cylinders (**A**), using the diaphragms in Figure 1. The mass of each cylinder unit is 25 grams. Connected cylinders (**B**) have pistons modified with a flow-through passage and seals to connect to the plug-end of the next cylinder. Each cylinder has a stroke of 18 mm. Vacuum operation and water-filled operation are shown in the video.

We constructed cylinders of this configuration using soft materials; they are shown in figure 6. The cylinder housing is medium soft vinyl tubing (Shore A85), the pistons and plug-ends are 3D printed acrylate plastic, and synthetic tension cords are anchored to through-drilled PEEK machine screws.

The principle disadvantage of this design is the limited pressure available with vacuum operation. This cylinder, under full vacuum, can only exert 24 N (5 lbf) of force. With a mass of only 25 grams this design is competitive with linear electromagnetic actuators, but a positive pressure-acting cylinder exerting the same force could be made much, much smaller. Connecting several cylinders together as "muscle units" allows increased stroke, although force stays constant. Because the actuators are soft and self aligning, they may find use in human soft-exoskeleton applications [14]. The video shows passive cylinder operation under vacuum as well as water-filled operation in a passive transmission configuration.

Tension-balanced and bushing-free operation combined with positive pressure (instead of vacuum) operation is achieved with the configuration shown in figure 5-**C**. Here, the *piston* is grounded and the cylinder is in motion. The external load is transmitted through a pair of tension cables anchored to the cylinder. Pairs of these cylinders in an antagonist configuration may drive a rotary joint via tendons, or the cylinders may be arranged head-to-head to form a double-acting cylinder. This arrangement would be equivalent to the one in figure 3-**C**, but with the shared double-ended piston grounded, and the moving cylinders connected with tie rods. This configuration may be preferred when the lowest possible friction and high torque density are both required.

IV. OPERATION AND PERFORMANCE

The two pairs of cylinders constitute input and output actuators, connected to form a passive fluid transmission; in our experiments they were connected by flexible tubing. For air operation we used 2.4 mm ID, 3.2 mm OD, flexible nylon tubing, 60 cm long, as shown in Figure 7. To maximize transmission stiffness for water operation we used rigid nylon tubing, 4.3 mm ID, 6.4 mm OD, 122 cm long, spiral wound for flexibility; this tubing is shown in the video segment on water operation. With a Bowden cable it is not possible to spiral the cable for added flexibility due to the exponentially increasing capstan-type friction. This tubing has a larger ID to combat the greater viscosity of water. A detailed discussion of fluid viscous friction and added mass in passive hydraulic transmissions is found in [7].

The connecting tubing forms two isolated and closed volumes. During fill and pre-pressurization an isolation valve, as shown in Figure 7, is opened to equalize fluid pressure on both sides of the transmission and then sealed for operation. For air operation, a bicycle pump is used for filling and pressurization; for water operation, additional rolling diaphragm cylinders fitted with screws to push against the piston rods serve to fill the system and adjust the preload pressure.

A. Transmission Stiffness

To measure transmission stiffness for air operation, the input-side shaft was blocked and a calibrated torque sensor fitted between the output shaft and a lightweight handle, serving as a manipulandum for a human operator. The handle was slowly rotated manually through a total stroke amplitude of approximately 60° with a frequency between 0.1 and 0.2 Hz, keeping fluid velocity low to minimize fluid

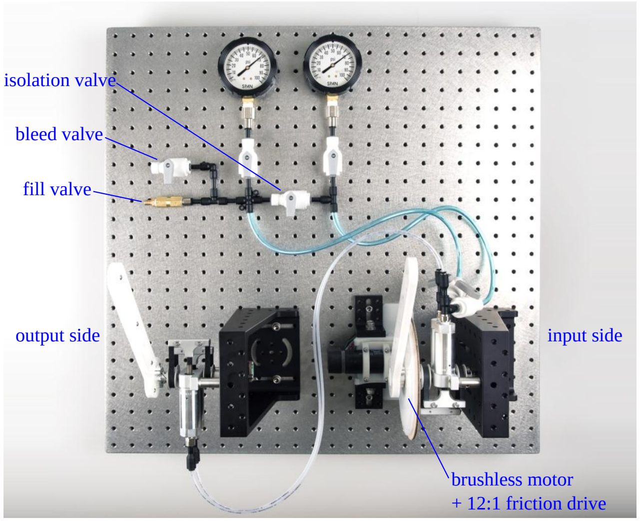

Fig. 7. (color) Benchtop transmission testing setup, plumbed for air operation. In addition to the primary isolation valve, each line on the input side has a local isolation valve to minimize transmission volume and compliance during operation. During water operation a rolling diaphragm cylinder-based reservoir is used to fill and pressurize the system. The motor and friction drive gear reduction used for powered tests is somewhat underpowered for this transmission; to equal the continuous torque rating of the transmission would require a total gear ratio of 42:1. The bare motor, without encoder or gear reduction, weights 260 grams.

damping. Both the input and output shafts are fitted with 10,000 count (after quadrature) optical encoders. Figure 8 plots work loops measured at 2 bar and 6 bar. Additional work loops were made with the isolation valve open, where transmission stiffness should be zero. These loops were full stroke, pushing until the cylinders bottomed out to measure the total range of motion. All work loops are clockwise, indicating energy loss, and are smooth, with no cogging evident from the timing belts.

The air tubing internal volume is about 70% of the theoretical cylinder volume. Because initial stiffness is inversely proportional to internal air volume, the achievable stiffness might be doubled over the result here by using smaller bore tubing and with more careful attention to the internal geometry of the valves.

Stiffness of the transmission when filled with water is 21 N-m/rad, as shown in Figure 9, also measured by manually backdriving the blocked transmission, with a torque-sensor-fitted handle.

A Maxon EC32 brushless motor with a 12:1 friction-drive gear reduction was fitted to the input side of the transmission and used to render a linear spring. This gear ratio was selected to maintain backdrivability and be feasible to make with a single-stage. It provides sufficient torque for testing but cannot deliver the transmission's full rated torque. Figure 10 shows a work loop when the handle is operated against the motor on the input-side shaft directly and then on the output shaft; in both cases the motor rendered a 1 N-m/rad virtual spring. Dissipation in the friction drive is negligible compared to the transmission. Performance is

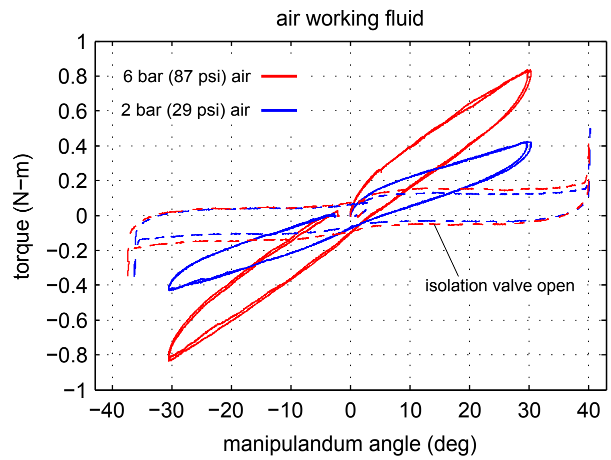

Fig. 8. (color) Air operation. The input shaft is blocked and a handle with a torque sensor is manually operated to determine system stiffness. Operation with the isolation valve open demonstrates the lack of significant transmission spring rate. For pressurized operation, work loops from two independent trials are overlain.

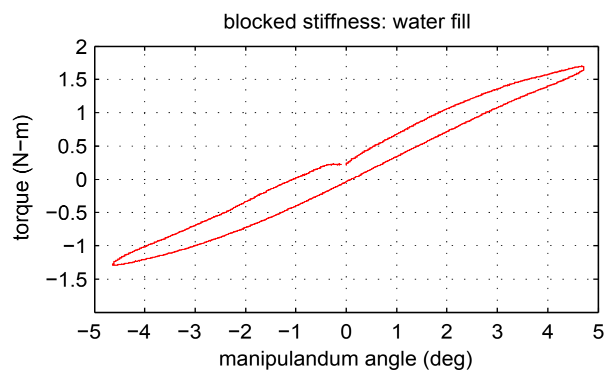

Fig. 9. (color) Water operation, 2.75 bar (40 psi) preload pressure. The setup is the same as for air stiffness tests. System stiffness with water operation measures approximately 21 N-m/rad.

very similar to the air operation tests, but in this case both transmission shafts are in motion during the work loop and so the overall hysteresis and friction is higher. In all motorized tests, encoder feedback comes only from the transmission input shaft; neither the output shaft angle or line pressures are used to improve the rendering. The low stiction and smooth behavior of the transmission should allow for much higher performance with a more sophisticated controller and additional feedback.

Compliance in the transmission during water operation is a combination of breathing of the tubing, stretching of the diaphragms, and compliance in the timing belts. Separating out each effect would require careful tests with variation of the parameters of each component; the intent here is to illustrate characteristic performance under realistic conditions.

B. Dynamic Performance

Force bandwidth was measured by commanding sinusoidal torques to the motor at the input shaft and attaching the

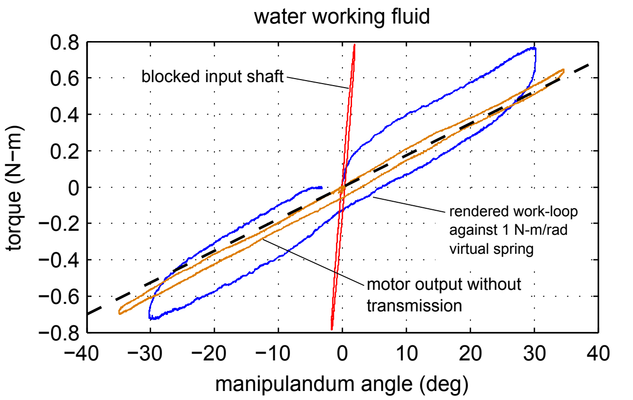

Fig. 10. (color) Water operation, 2.75 bar (40 psi) preload pressure. Result of haptic rendering of an ideal 1 N-m/rad linear spring (dashed line). A measured work loop, hand traversed, at both the motor friction drive output shaft without the transmission (orange) and with the transmission fitted, measured at the output shaft (blue). Maximum system stiffness is shown for reference (red).

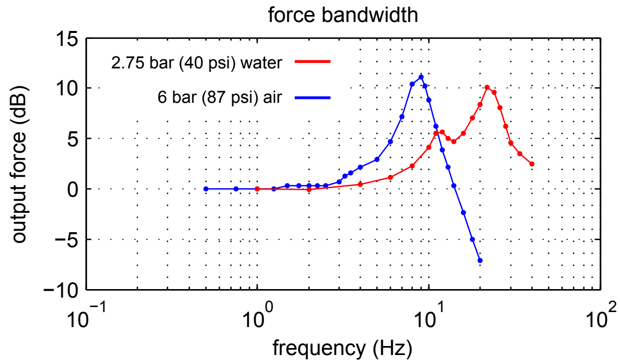

Fig. 11. (color) Force bandwidth is measured by fitting a torque sensor to the transmission output which is in turn grounded. Sinusoidal torques at individual frequencies are commanded to the motor. At DC, water-mode achieves 82% force transfer, air-mode 67%, comparing measured transmission output shaft torque to measured friction drive output shaft torque.

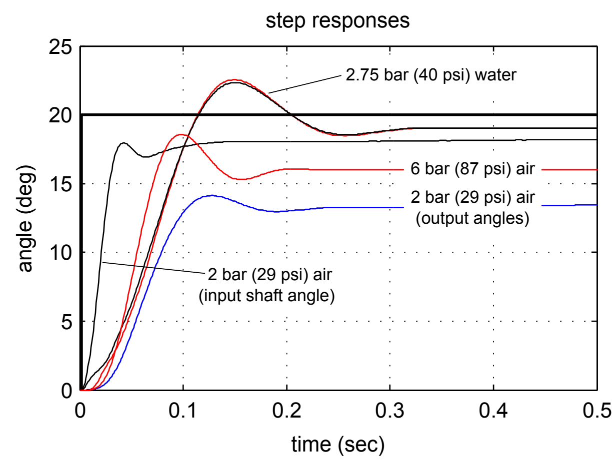

Fig. 12. (color) For this test, the manipulandum handle was fitted to the transmission output shaft; its measured moment of inertia is $9.8{\times}10^{\ 4}$ kg-m^2.

torque sensor between the output shaft and ground. The current amplitude was 3 Amps, corresponding to a measured static output torque, after the friction drive, of 0.375 N-m. Results for both air and water are shown in Figure 11. At 1 Hz, force transmission efficiency was 82% for water fill and 67% for air fill. The second resonance peak during water operation may result from fluid acceleration, which acts as a second effective mass in the system, but detailed modeling and further analysis is required. Extracting this effect would also require separate characterization of the friction drive without the fluid transmission.

We also measured displacement step responses to a commanded 20° step input, as shown in Figure 12. For this test, simple PD feedback was applied to the input shaft encoder reading. All tests used the same gain values. In this test, the added inertia of the water is evident; the motor is torque saturated for most of the response time so the water-filled step response is acceleration limited. Note that the water-filled transmission maintains almost exact synchronization between the input and output. Steady-state errors result from a lack of integral feedback. In this test, the manipulandum handle is mounted to the output shaft, with a measured mass moment of inertia of $9.8{\times}10^{-4}$ kg-m^2.

The video shows a high-speed video recording of the transmission driven by an impulse, and the resulting time response. The video also shows passive operation with a 500 gram load attached to the output handle.

C. Friction, Hysteresis, and Backlash

Hysteresis is clearly evident in the work loops shown (and noticeable when manipulating the transmission by hand). The work loops testing transmission stiffness at 6 bar air pressure exhibit a total hysteresis torque amplitude of 230 mN-m, or 4% of full-range rated torque ($\pm$2.7 N-m). At 2 bar, hysteresis is 170 mN-m (3%). When rendering the linear spring for water-filled operation, where all four diaphragms are in motion, hysteresis was 330 mN-m (6%) .

Stiction, which appears in the work loop plots as a vertical drop in torque with no change in angle, is low in all tests. Work loops collected during air operation show a stiction of 20 to 35 mN-m. For water operation during haptic rendering, stiction was 30 to 50 mN-m. Overall, maximum stiction was observed to be less than 1% of full-range torque.

Hysteresis is caused by the rolling and unrolling of the diaphragm and, to a lesser extent, flexing of the timing belt. Improving hysteresis directly will likely require the use of a diaphragm elastomer with an intrinsically lower mechanical loss tangent and higher resilience, or thinner diaphragms. Because the observed hysteresis was very smooth and repeatable, we are confident that additional feedback and control will reduce its impact. Future work should include servo-driven (dynamometer) work loop testing to accurately determine the effect of velocity on hysteresis.

Figure 13 shows transmission operation, water-filled to 2.75 bar (40 psi), manually driven over most of the transmis-

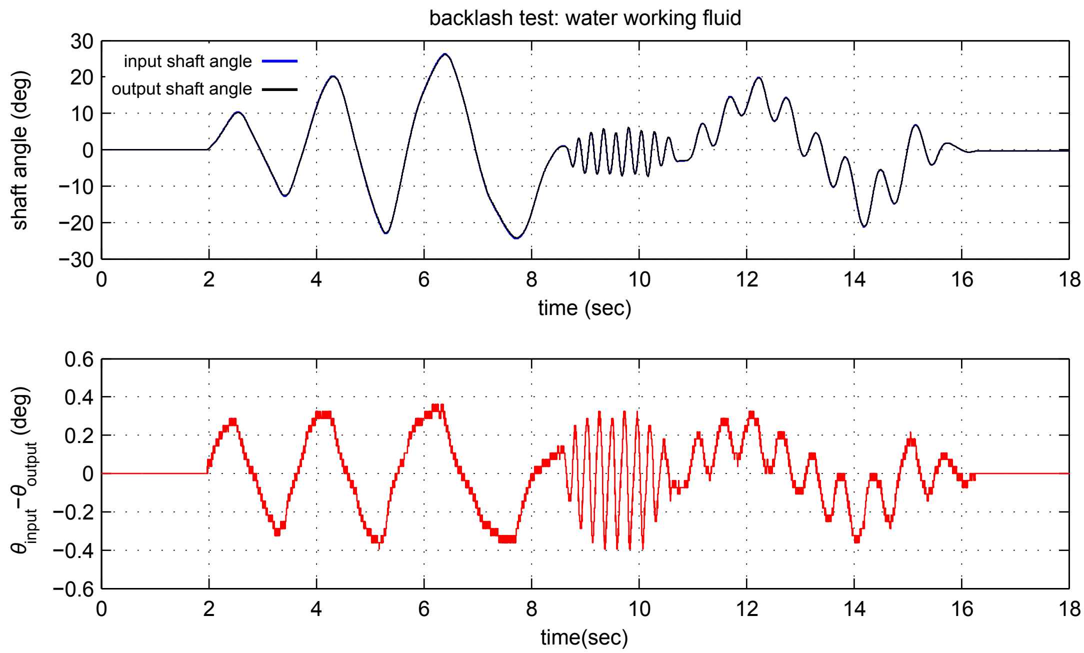

Fig. 13. (color) Water operation, 2.75 bar (40 psi). The input was manipulated manually over various ranges and speeds to determine tracking errors and backlash.

sion stroke. Synchronization between the input and output shaft is maintained to less than 0.5 degrees. Note that the tracking error is in phase with the input angle. This, and the smooth error signal, indicate that transmission backlash, vis-a-vis lost motion, is effectively zero, and the tracking error results only from the small, but non-zero, transmission compliance.

D. Limitations, Variations, and Improvements

This constant-volume transmission technique is attractive because of its simplicity and passivity. However, long-term operation relies upon continued maintenance of a constant volume state. The choice of materials is then especially important to minimize absorption of liquid and/or permeability of air. Thicker diaphragms are more impermeable, but will suffer greater hysteresis. Because the transmission is under constant pressure, even at rest, material creep, particularly in the tubing or hose, can lead to volume change. If many transmissions are used in a complex system, it should still be relatively straightforward to periodically open the isolation valves and reset the pressure as needed. The impact of creep and absorption and the difficulty of mitigating their effects will become clearer with increasing use of this transmission method.

Ultimately, use of this transmission in motor-driven systems is only beneficial if the output actuator has a higher torque density than the motor it displaces. Overall torque density is limited by the input motor, so using this transmission will actually decrease overall torque density, but limb inertia will be significantly reduced. This first-generation transmission design, using off-the-shelf components, is heavy and not optimized for high torque density. When filled with water, the output actuator weighs 246 grams. As a point of reference, the EC32 2-pole brushless motor used in testing the transmission weighs 260 grams, and requires a 42:1 gear ratio to equal the torque rating of the transmission rotary actuator, so this transmission already exceeds the torque density of lightly geared motors. For further comparison, a Dynamixel MX-28 servo can produce up to 3.1 N-m of torque and weights only 72 grams, but with a transmission ratio of 193:1, it is restricted to slow speeds and exhibits very high output impedance.

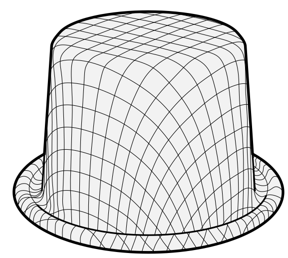

Fig. 14. Commercially available rolling diaphragms are produced by compression molding thin sheets of rubber backed by a finely woven high-tenacity fabric. Even with a high-drape weave, the diaphragm may only be drawn so far before fabric distortion becomes excessive and pleating occurs. This illustration shows the typical deformation patterns of the fibers after molding.

With future refinement, we believe the torque density of this transmission can be improved by an order of magnitude. Optimization of the cylinders and actuator structure can reduce mass, perhaps by a factor of three. The second generation transmission, currently under development, has a maximum operating pressure of 17 bar (250 psi), twice the current design, and uses a diaphragm with a larger stroke-to-bore ratio. As shown in figure 14, distortion of the fabric fibers during molding limits draw depth, but also results in a diaphragm that cannot maintain a cylindrical shape under pressure; it is not self-supporting, and must be placed inside a rigid cylinder to constrain its motion under pressure. If the diaphragms were instead molded individually, with customized fiber orientation, then under pressure the diaphragm could be self-supporting, and the mass of the rigid cylinders could be eliminated. Custom reinforcing fiber orientation would also allow for longer stroke diaphragms.

V. CONCLUSIONS AND FUTURE WORK

We have presented a unique fluid-based transmission using pre-loaded antagonist rolling-diaphragm cylinders. It exhibits stiction below one percent of full-range torque, complete backdrivability, and freedom from backlash. Using bulky off-the-shelf components, it is already mass-competitive with electric motors with moderate gear ratios, and is expected to exceed the torque density of harmonic drives and high-impedance servomotors with continued development.

The primary motivation and application for this transmission design is removing heavy electric motors from distal robot joints without suffering the high and variable static friction of Bowden cables or the complexity of ball-bearing-routed cable drives. At the joint, we gain the favorable torque density of a fluid actuator, but the properties of this transmission allow us to maintain backdrivability, low stiction, and freedom from backlash.

Because the transmission is completely passive, it is also possible to couple non-actuated joints together. The flexibility and low friction of the transmission lines allows joints with a greater physical separation to be coupled than is possible with Bowden cables. Differential coupling of multiple joints is straightforward. This capability may be useful for robots with passive-dynamic structures [15] or bilateral coupling for body-powered prosthetics. The transmission may also be applied to traditional powered prosthetic applications.

This transmission appears ideally suited for more advanced control methods, particularly the easy introduction of force feedback through microchip-based pressure sensors. Highly geared motors can drive the transmission, if desired, as a series elastic actuator using fluid pressure feedback.

This fluid-based transmission could also be used for building an MRI-compatible robot, especially if maintaining backdrivability is desired. The transmission is stiff enough under liquid operation that direct drive by a human operator would be possible, either in a completely passive mode like a classic cable-drive master-slave manipulator, or with motor augmentation for active hysteresis compensation and hand-tremor reduction. The high stiffness and high force bandwidth allow haptic-quality direct force feedback to the operator. Ongoing work aims to increase stiffness and reduce intrinsic hysteresis in support of better haptic feedback for remote manipulation applications.

Longer-stroke diaphragms would instantly multiply the work per cycle of the transmission with little increase in system mass, and also reduce the impact of diaphragm and timing belt hysteresis. Rolling diaphragms might be considered as a replacement for traditional McKibben air muscles, offering higher strain and spring-rate-free operation. Longer-stroke diaphragms, in this application, would allow a more favorable actuator aspect ratio, particularly for the linear, double-acting cylinder configuration. Custom-made diaphragms with optimized fiber reinforcement should enable higher pressure operation, and the elimination of the rigid cylinder housing.

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