Evaluation of a Parallel Actuation Approach for MR-Compatible Haptics

Chembian Parthiban* Christopher Esser† Michael R. Zinn‡
University of Wisconsin–Madison, Madison, WI, USA

ABSTRACT
In recent years, the use of haptics within the MRI environment has increased, particularly for fMRI procedures. However, the demanding MRI environment has made the development of high performance devices very difficult, as most common actuation approaches are not well suited to the high-strength magnetic fields present in the MR-bore. An alternative actuation concept has been proposed using the motion of parallel ultrasonic motors. In this paper we investigate the use of this new approach for MR-compatible haptic devices, including the development of a suitable control approach, experimental validation, and an analytical investigation of achievable virtual admittances. A prototype of this device is constructed and evaluated. Results demonstrate the viability of the proposed approach for MR-compatible applications.

KEYWORDS: MR-compatible haptics, medical robotics, actuators, MR-compatible actuation.

1 INTRODUCTION
The use of haptic systems in a Magnetic Resonance Imaging environment has many applications, particularly in the area of fMRI procedures where neurological studies using functional MRI requiring haptic feedback within the MR-bore [1-7].

However, working in an MRI environment is very demanding due to the limited space of the MR-bore and the severe material constraints due to the MR magnetic and RF environment. These constraints are particularly challenging in regards to actuation development. The large magnetic fields and sensitivity to RF noise preclude the use of ferromagnetic materials in any part of the device, all but ruling out the use of electromagnetic actuators as well as many commercially available actuators.

1.1 Previous Actuation Methods
To overcome these limitations, a number of researchers have developed MR-compatible actuation methods using a variety of approaches including pneumatic, hydraulic, and ultrasonic components [1-5, 8-13]. While these actuation approaches have met with considerable success, their use in applications that require actuation with high-bandwidth linear characteristics, such as in a haptic interface, has been challenging.

Pneumatic actuation methods have primarily focused on stepper-motor implementations [1, 3, 8, 9, 12, 14]. In these cases, pneumatic valves are located outside the bore for MR-compatibility. Air-line compliance is generally not an issue as the actuator is operated open-loop, in that desired positions are achieved through the command of specified actuator steps. This configuration functions well for position control applications but does not lend itself to haptic applications where high-bandwidth linear actuator characteristics are required.

Hydraulic actuation has been implemented in a number of prototype systems [1, 10, 15, 16]. This approach works well for low frequency applications. However, due to fluid line compliance and valve nonlinearities, this approach does not lend itself to applications requiring high-bandwidth control.

Finally, ultrasonic motors (USMs) have been extensively used in MR-compatible robotic applications. Ultrasonic motors create rotational motion by applying high-frequency, high-voltage signals to a ceramic stator. The stator is pressed against a ceramic rotor and vibrates in response to this signal, creating what is known as a traveling wave. This wave causes the rotor to advance with a velocity that is determined by the amplitude and frequency of the traveling wave. Because of these different operating principles, USMs do not rely on magnets or ferromagnetic material for motive torque. For this reason, they are well suited for MR-applications [2, 17, 18]. However, while previous applications using USMs have met with success, system performance has been limited by the non-linear characteristics of the ultrasonic motors [2, 18].

© 20xx IEEE. Personal use of this material is permitted. Permission from IEEE must be obtained for all other uses, in any current or future media, including reprinting/republishing this material for advertising or promotional purposes, creating new collective works, for resale or redistribution to servers or lists, or reuse of any copyrighted component of this work in other works.

* e-mail: parthiban@wisc.edu
† e-mail: cesser85@gmail.com
‡ e-mail: mzinn@wisc.edu

Fig. 1. Measured ultrasonic motor (USM) output velocity as a function of desired input velocity (proportional to input voltage frequency). Note nonlinear velocity deadband about zero velocity. Model tested: Shinsei USR60E53N [20].

1.2 Ultrasonic Motor Limitation
Ultrasonic motors can provide a linear, one-to-one relationship between desired and actual output velocity if a suitable velocity control approach is implemented [19]. However, due to the USM’s physical limitations, it generally cannot be driven below a minimum threshold velocity, typically 10% of the maximum output velocity [20]. This creates a nonlinear velocity-dead zone about zero velocity (see Fig. 1). In addition, there is commonly a time delay associated with velocity reversals [20]. These characteristics make high-bandwidth control, essential for admittance-based haptics devices, very challenging.
2 Proposed Actuation Concept

To overcome the limitations of USMs, a new actuation concept has been proposed [21]. In this paper we investigate the use of this new approach for MR-compatible haptic devices, including the development of a suitable control approach, experimental validation, and an analytical investigation of achievable virtual admittances.

The proposed approach combines the output motion of two, parallel USMs through the use of a differential mechanism. While the details of the approach are given in [21], an overview is provided here for completeness. The differential mechanism acts as a motion summer, where the output is a linear combination of the two parallel USM’s motion. Unlike previous investigations of USM use in MR-compatible haptics[18], the velocity of each USM can be constrained to the linear region of operation, avoiding the dead band non-linearity and delay associated with velocity reversals. A functional overview of the proposed approach is shown in Fig. 2. Output velocity is regulated by independently adjusting the relative input velocities of the two input USMs.

\[ \dot{\omega}_o = C_1 \dot{\omega}_1 + C_2 \dot{\omega}_2 \]

where \( C_1 = \frac{\partial \theta_i}{\partial \dot{\theta}_1} \); \( C_2 = \frac{\partial \theta_i}{\partial \dot{\theta}_2} \)

The weighting coefficients \( C_1 \) and \( C_2 \), in the case of a common differential drive mechanism, are constant. A possible gear arrangement for a differential using a planetary gear is shown in Fig. 3(a).

![Fig. 3. Possible differential gear arrangements for use in the proposed actuation approach. (a) single output, dual (parallel) inputs, (b) two outputs, three inputs, (c) n outputs, n+1 inputs (with common input shaft with constant velocity).](image_url)

![Fig. 4. Output velocity, \( \omega_{out} \), as a function of input velocities, \( \omega_1 \) and \( \omega_2 \).](image_url)

In general, the resulting output velocity, \( \omega_{out} \), as a function of input velocities, \( \omega_1 \) and \( \omega_2 \), and the weighting coefficients is shown in Fig. 4. To avoid the nonlinearities associated with velocity reversals, minimum velocity limits are enforced for each USM. In addition, the maximum velocity of each USM introduces an additional constraint. As such, the output velocity, \( \omega_{out} \), is limited to the regions identified in Fig. 4. As seen from Fig.
4, bi-directional motion is possible when operating in quadrants II and IV while quadrants I and III support unidirectional motion only. The maximum obtainable output velocities are located at the extreme corners of quadrants I and III. Tasks requiring linear, bi-directional output would, by necessity, operate within quadrants II or IV. When higher output velocities are desired, mode transitions between quadrants are possible assuming that the delay and nonlinearity that occurs during the transition is acceptable. In most applications, where the USM actuators are identical, the weighting coefficients would be equal in magnitude. In situations where different actuators are used or where desired positive and negative output velocities are not equal, unequal weighting coefficients may be desirable.

2.2 Velocity Partitioning

Redundant actuation requires purposeful partitioning of input velocities. As seen in Fig. 4, for a given desired output velocity, \( \omega_o \), the two input velocities are constrained along a line defined by equation (1). While there are numerous possible schemes to partition the input velocities such that a desired output velocity is achieved, we will discuss two schemes which have been implemented on the hardware discussed in Section III.

The first partitioning scheme considered takes a simplified approach, where the input velocities fall along curve (1) shown in Fig. 5. Considering equation (1) and assuming that the minimum and maximum velocities of the two input motors are equal, the required input velocities as a function of desired output velocity is given as:

\[
\omega_1 = \left(\frac{C_2}{C_1 + C_2}\right)(\omega_{\min} + \omega_{\max}) + \left(\frac{1}{C_1 + C_2}\right)\omega_o
\]

\[
\omega_2 = \left(\frac{-C_1}{C_1 + C_2}\right)(\omega_{\min} + \omega_{\max}) + \left(\frac{1}{C_1 + C_2}\right)\omega_o
\]

If the weighting coefficients are equal (\( C_1 = C_2 = C \)), equation (2) reduces to:

\[
\omega_1 = \frac{1}{2}(\omega_{\min} + \omega_{\max}) + \left(\frac{1}{2C}\right)\omega_o
\]

\[
\omega_2 = -\frac{1}{2}(\omega_{\min} + \omega_{\max}) + \left(\frac{1}{2C}\right)\omega_o
\]

The second partitioning scheme seeks to minimize the Euclidian norm of the input actuator velocities, \( ||\omega|| = \sqrt{\omega_1^2 + \omega_2^2} \). This approach is motivated by the desire to minimize power consumption, as power output is proportional to velocity, as well as maximize available actuator torque – as the output torque of the USMs are greatest at lower velocities. This velocity partitioning profile is described by equation (4) and is shown as curve (2) in Fig. 5.

\[
\begin{align*}
\text{if } \omega_o > 0 & \quad \omega_1 = \frac{C_1}{C_1 - C_2} \omega_{\min} + \frac{1}{C_1} \omega_o \\
\text{if } \omega_o < 0 & \quad \omega_1 = \omega_{\min} \\
\end{align*}
\]

\[
\begin{align*}
\omega_2 = -\omega_{\min} & \quad \omega_2 = -\frac{C_1}{C_2} \omega_{\min} + \frac{1}{C_2} \omega_o \\
\end{align*}
\]

As seen in Fig. 5, the velocity norm is minimized at the intersection of the isocline associated with \( \omega_o \) and the minimum velocity constraints of the input motors.

When considering which partitioning scheme to adopt, it is important to consider the limitations of the input actuators. As mentioned above, minimizing the norm of input velocities will ensure that the torque-velocity space of the parallel actuation is as large as possible, particularly at lower output velocities. However, the acceleration capability of the input actuators may also limit performance.

3 HARDWARE IMPLEMENTATION

To validate our approach, a one-degree-of-freedom MR-compatible, admittance-based haptic device prototype, based on our parallel actuation approach, was constructed. The goal was to evaluate the performance of the actuation technique and demonstrate its use in a simulated MR-clinical environment. As such, the testbed was designed to emulate a simple hand-motion task as might be required during a fMRI investigation. The developed testbed is shown in Fig. 6. The testbed consists of an input knob supported on a linear slide. An integrated Futek LSB200 force sensor [26] provides force measurements. The linear slide is driven through a rack and pinion which in turn is driven by the parallel actuation prototype.

3.1 Differential Mechanism

We chose to implement the differential mechanism, which combines the input USM motion, using a planetary gear train (see Fig. 8 and 3(a)). The two input ultrasonic motors drive the sun gear and ring gear. The planet gears, coupled to the output shaft, provided the actuator output. This configuration was chosen because it possessed simple kinematics and was easily fabricated. An additional gear stage, using brass gears to maximize the stiffness of the system while maintaining MR compatibility, was added between the planetary gear and each USM to equalize their contribution to the parallel actuator output velocity. The system
was designed such that the output velocity was a simple summation of the input USM actuator velocities.

3.2 Actuators and Processing

The ultrasonic motors selected for the prototype system were from Shinsei Corporation (Tokyo, Japan). The ultrasonic motor (model USR60E) and its drive (model D6060E) came as a matched pair. The Shinsei USM was constructed entirely of non-ferromagnetic materials and was equipped with a 1000 counts/revolution encoder. The control software was written using Matlab xPC. The xPC target was equipped with standard I/O cards for analog outputs, analog inputs, and encoder counters.

3.3 Input Knob Linear Stage

The output of the planetary gear is connected to a rack and pinion that drives the linear stage of the device. The gear ratio from rack to pinion was chosen so that maximum rotational velocity of the output (~15 rad/sec) would map to a linear velocity of approximately 150 mm/sec. This maximum velocity is reasonable for haptics applications. The total linear translation of the sled is approximately 100 mm. The linear position of the slide was measured using rotary encoders mounted on both the output pinion shaft and the output shafts of both input USMs. The device hardware is shown in Fig. 6.

The linear stage served as a mount for the input knob and force sensor. These were supported by a matched pair of linear flexures, which served to isolate the force sensor from all forces not applied axially. This was important so that erroneous forces were not fed back into a force controller, which would cause abnormal behavior.

Extreme care was taken to ensure all components were MR compatible. Aluminum and brass were used as structural materials, including use in the drive train, as their higher stiffness as compared to plastic components provided the stiffness required. All bearings used were constructed from glass impregnated ceramics.

RF noise was also a major concern in the design of the prototype. The high frequency signals that drive the USMs create RF noise that needed to be blocked from entering the MR room. A faraday cage was constructed to house the actuator electronics. The Faraday cage was grounded to the MR-scanner room's Faraday cage through a low impedance ground.

To assess initial MR-compatibility a set of simple imaging tests were performed. Specifically, MR-images were acquired using a T1-weighted spoiled gradient echo imaging pulse sequence on a 1.5 T scanner (SignaHDx, General Electric, Milwaukee, WI). Images of a spherical phantom placed in close proximity of the haptic device prototype where taken under various conditions (see Fig. 7). The system was compared to images taken with the haptic device prototype in both the powered and unpowered state. As seen from the Fig. 7, the MR-images are virtually identical, demonstrating that the USM-actuated model subsystem was compatible with concurrent imaging and robotic actuation. The signal to noise ratio, measured as the mean of the signal divided by the standard deviation of the noise, was approximately 159 (44.0 dB) for the powered system while the phantom alone, with no device or controller present, was approximately 171 (44.7 dB), resulting in a difference of less than 0.7 dB. While the initial MR-compatibility testing shows promise, a more rigorous testing process is currently under development – to include T2 weighted fields, using EPI sequences acquired with and without the device present.

![Fig. 6. Overview of parallel actuation test-bed. Parallel USM (slave) system with rack and pinion output. Linear slide drives an input knob in-line with an MR compatible force sensor.](image)

Fig. 7. MR-images of a spherical phantom placed adjacent to the parallel actuator prototype in the unpowered and powered state.

4 EXPERIMENTAL RESULTS

An experimental setup was developed using the constructed prototype to identify the characteristics and evaluate the performance of the parallel actuator mechanism as a haptic device. The first set of experiments involved evaluating the position control of the device and deriving an equivalent model of the closed loop position control followed by a series of experiments to model and to predict the device haptic performance.

4.1 Position Control

The hardware described in Section III was used (see Fig. 6) to implement the parallel actuation approach, and a position controller of the type shown in Fig. 8 was implemented.

To avoid a non-collocated actuator-sensor arrangement, and thus avoid the stability issues associated with a non-collocated approach [22, 23], the control structure in Fig. 8 uses direct measurements of each USM shaft rotation. An estimate of the
combined actuator output motion, $\theta^*$, is obtained by combining the measured USM rotations using the summation mechanism kinematics.

![Diagram of parallel actuator position control structure.](image)

Fig. 8. Overview of parallel actuator position control structure. Controller includes proportional control in the outer position loop with inner PI control loop on USM motor velocity.

While a complete description of the USM dynamics is complex, for low inertial loading we can assume a 1st order model relating desired USM input velocity to USM output velocity [19]. The addition of an inner-loop velocity PI controller results in an overall 2nd order system relating desired velocity, $\omega_i$ ($i = 1, 2$) to output velocity, $\omega_o$ as shown in Fig. 8. The high output impedance of the USM, in combination with the high-gain velocity loop, allows us to ignore loading effects for the application in question. While there are many possible position control compensation filters that could be used, the dynamics of the open-loop system, including the inner velocity controller, allow for the use of a simple proportional controller as long as the compensated system's cross-over frequency is set below the inner velocity controller's closed loop bandwidth (~50 Hz). In this case, the open-loop compensated system will approximate a pure integrator and will possess a phase margin approaching 90 degrees – providing ample stability robustness.

The first set of position control experiments evaluated the step response of the closed-loop parallel actuator system. As seen in Fig. 9(a), the closed-loop system exhibits a 2nd order dominant root response with an undamped natural frequency of approximately 50 Hz and overall time constant of 3.75 milliseconds, resulting in a closed-loop bandwidth of approximately 42 Hertz. The position controller is robust to gain variations and, in particular, behaves well under USM velocity saturation conditions. As seen in Fig. 9(b), a large step input exhibits velocity saturation but remains stable with little or no over-shoot.

As discussed and from the experimental step response we can see that the dominant response is of a second order system and an equivalent model described by the below transfer function for the closed loop position controller was developed by matching the closed loop bandwidth of the system.

$$\frac{\theta_d}{\alpha} = \frac{7.198 \times 10^4}{s^2 + 187.8s + 7.198 \times 10^4} \quad (6)$$

### 4.2 Haptic Performance

To evaluate the haptic specifications of the actuation approach a series of experiments were performed to evaluate the performance limits of the device and to develop a model describing the haptic interaction. A summary of the device specifications are given in Table 1. The maximum force output is a function of a number of factors including the maximum output torque of the actuators, the reduction ratio of the planetary gear and the overall efficiency of the gear train. However, for the prototype developed, the maximum force is limited by the load rating of the force sensor as well as the load capacity of the linear guiding flexures. The maximum output velocity is limited by the maximum output velocity of the USM in combination with the differential gear design.

![Graph of position control step response.](image)

Fig. 9. Position control step response of the closed-loop parallel actuator testbed. (a) response to small magnitude input step command, (b) response to large magnitude input step command.

<table>
<thead>
<tr>
<th>Specifications</th>
<th>Device Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Force$^a$</td>
<td>80 N / 45 N</td>
</tr>
<tr>
<td>Maximum Velocity</td>
<td>133 mm/s</td>
</tr>
<tr>
<td>Minimum Tip Inertia</td>
<td>200 grams</td>
</tr>
<tr>
<td>Closed-loop position</td>
<td>~40 Hz</td>
</tr>
<tr>
<td>Control bandwidth</td>
<td></td>
</tr>
</tbody>
</table>

$^a$Maximum force reported corresponds to torque limits of the USM and the load limits of the force sensor

The ability of the system to render virtual admittances is primarily limited by the closed-loop bandwidth of the closed-loop position controller as well limiting factors such as actuator torque and velocity saturation and unmodeled dynamics. To explore this further, the minimum tip inertia of the prototype was experimentally determined through simple user experiments. In the user experiments, the virtual inertia, $m_v$, was adjusted while the virtual stiffness, $k_v$, and damping, $c_v$, were set to zero. While somewhat subjective in nature, these user experiments demonstrated that the prototype device could consistently render a mass of 200 grams or greater without exhibiting oscillations.

### 4.3 Frequency Response Experiments

In addition to the simple user experiments, the frequency response of the system was experimentally investigated to explore the limits of the overall approach. The objective of this testing was to understand the system dynamics and develop a model of overall system where by the performance limits could be studied.
The assumed dynamics model of the overall haptic system is shown in Fig. 10(a). The system block diagram in Fig. 10(a), illustrates the admittance control approach adopted, including the parallel actuator position controller discussed earlier. In addition, the system block diagram includes the desired virtual admittance (a function of the virtual environment) as well as the impedance of the human user.

\[
X_m(s) \rightarrow T_{h,m}(s) \rightarrow F_b(s) \rightarrow F_h(s) \rightarrow X_o(s) \rightarrow X_d(s) \rightarrow \text{Force Sensor} \rightarrow F_m(s) \rightarrow \text{Closed Loop Position Controller} \rightarrow \text{Human Impedance} \rightarrow \text{Virtual Admittance} \rightarrow X_o(s) \rightarrow X_m(s) \rightarrow F_b(s) \rightarrow F_h(s) \rightarrow X_o(s) \rightarrow X_d(s) \rightarrow \text{Force Sensor} \rightarrow F_m(s)
\]

(a)

Fig. 10. System block diagrams of haptic system: (a) assumed system model, (b) open-loop frequency response experiment signal flow.

The frequency response experiments sought to measure the open-loop response of the system shown in Fig. 10(a), excluding the desired virtual admittance and assuming that the desired input motion, \(x_d\), is zero. A block diagram of the open-loop system is shown in Fig. 10(b). The experiments were performed on the system described in Section 3 using two different user subjects as well as low mass foam and unloaded conditions (see Figure 11).

One important assumption of the assumed model is the high output impedance of the USM parallel actuation. In Figure 12, the frequency response as measured from the desired actuator input displacement, \(x_d\), to measured actuator output position, \(x_m\), clearly shows that both the magnitude and phase response for the open-loop system under various loading conditions is virtually identical. If the actuation was affected by the impedance of the load, the frequency response would show deviations in magnitude and/or phase. We are not able to view the fall off at the cut off frequency as we restricted the frequency range to 30 Hz to prevent damage to the hardware.

![Overlaid frequency response plot for different load conditions including no load, low-mass foam, and two humans. Note, the high output impedance and high closed loop bandwidth results in a tight agreement between the various loading cases, making it difficult to differentiate the various outputs.](image)

![Assumed human impedance model](image)

The frequency response would show deviations in magnitude and/or phase. We are not able to view the fall off at the cut off frequency as we restricted the frequency range to 30 Hz to prevent damage to the hardware.

\[
T_{h,o}(s) = \frac{F_h(s)}{X_o(s)} = \frac{(m_h s^2 + b_2 s + k_2)(b_1 s + k_1)}{m_h s^2 + (b_2 + b_1)s + (k_1 + k_2)} \quad (7)
\]

\[
T_{h,m}(s) = \frac{F_h(s)}{X_m(s)} = \frac{-(b_1 s + k_1)(b_2 s + k_2)}{m_h s^2 + (b_2 + b_1)s + (k_1 + k_2)} \quad (8)
\]

Fig. 11. Experimental setup. (a) human hand holding the input knob, (b) various loading conditions measured in open-loop frequency response experiments.

Fig. 13. Assumed human impedance model [24].
In Figure 14, the frequency response as taken from the measured actuator output position, $x_o$, to the measured force, $F_m$, is shown. The test cases shown in Figure 14 correspond to two user tests where the test subject was instructed to hold the device end-point as they would in a typical haptic exploration environment. While somewhat subjective in nature, the frequency response of the two test subjects was similar. For the purposes of analysis, we have chosen to fit a human-impedance model shown in Fig. 13 to the experimental data. The additional spring and damper used in the model represents a soft contact between the hand and the device which typically is how a clinician would hold on to an end effector similar to the one on this device. The device input knob is similar to a stylus described in [24] and so a rigid grasp is not possible as in a round handle used by Kosuge et al. [25]. The human impedance transfer function $T_{h,o}$ and $T_{h,m}$ given by equations (7) and (8) is taken from [24] and is overlaid on the experimental data, shown in Fig. 14.

![Fig. 14. Overlaid plot of the experimental frequency response of the human impedance and the fitted model response for two different human subjects.](image1)

4.4 Discussion

The closed loop system behaviour can now be analysed with the help of the experimentally determined position controller model and the fitted human impedance model. Specifically, the stability limits of the system for various virtual admittances can be studied by considering the open-loop frequency response of the system shown in Fig. 10(a) cascaded with the transfer function of the desired virtual admittance. As seen in Fig. 15, the stability margins of the open-loop system model, exclusive of the desired virtual admittance, is dominated by the closed-loop bandwidth of the position controller, $\omega_c$. For example, in the case of a rendered inertia, the open-loop gain, and thus minimum inertia, is limited by the phase loss introduced above $\omega_c$. As seen in Fig 16, the stability analysis underestimates the minimum inertia as compared to the user experimental data discussed earlier. In this case, the limiting factor is not the phase loss above $\omega_c$ but, instead, the velocity saturation limits of the USM actuators used in the prototype as well as the small but finite amount of a backlash in the mechanism design. However, the analysis in Fig 16 does serve to highlight possible limitations of the approach as well as the beneficial effect of achieving a high closed loop position bandwidth. It is also interesting to note the stabilizing effect of the human impedance model at higher frequencies. While the minimum inertia was determined through user tests and the frequency response plots, the transparency of the device was not directly measured. The users were not able to feel the friction of the device as at low frequencies as they were limited more by the rendered inertias of the order of 200gms. In future, impedance measurements will be performed to quantitatively determine the friction of the device and its transparency.

![Fig. 15. Frequency response of haptic system model – based on experimental data.](image2)

![Fig. 16. Frequency response of haptic system model cascaded with virtual inertia admittance ($m_v = 1$).](image3)
5 CONCLUSIONS AND FUTURE WORK

In this paper the haptic performance of a parallel actuation approach that combines the output motion of two parallel USMs through the use of a differential mechanism was discussed. The differential mechanism acts as a motion summer, where the output is a linear combination of the two parallel USM’s motion. The experimental test bed for the parallel actuation approach was used to evaluate the haptic performance and study the closed loop response of the system. The minimum mass of the system was determined both theoretically and experimentally.

REFERENCES


