A force-controlled planar haptic device for movement control analysis of the human arm

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Abstract

This paper describes the design and application of a haptic device to study the mechanical properties of the human arm during interaction with compliant environments. Estimates of the human endpoint admittance can be obtained by recording position deviations as a result of force perturbations. Previous studies attempted to estimate the impedance by recording force as a result of position perturbations, but these experiments do not require a feasible task of human beings. A general problem of force-controlled haptic devices is the occurrence of contact instability, especially where a small virtual mass is required. This negative effect is reduced by the use of a lightweight but stiff manipulator and a robust servo-based admittance controller. The virtual admittance is accurate to at least 13 Hz, attaining a minimum virtual mass of 1.7 kg (isotropic, without damping and stiffness). The properties of known test loads were estimated with an accuracy higher than 98%, up to 20 Hz. The application of the manipulator is evaluated by an experiment with a subject performing a position maintenance task. With this device it is possible to study the adaptability of the neuromuscular system to a variety of environments, enabling a new and functional approach to human motion research.

Keywords: Haptic device; Contact instability; Virtual environment; Arm admittance; Identification; Reflexive feedback

1. Introduction

Many daily life human arm motion tasks imply physical interaction with objects, hereafter referred to as the environment, of which the mechanical properties can be diverse. Most environments are well manipulable, meaning that humans are able to control their hand position by generating the appropriate muscle forces. By changing the arm admittance the combined admittance of arm plus environment can be modified. From the combined admittance, important dynamic properties can be derived such as mechanical stability margins and the resistance to external force perturbations (De Vlugt et al., 2002; Van der Helm et al., 2002). The admittance is defined as the dynamic ratio of position to force (m/N), and equal to the inverse of the mechanical impedance. A low admittance (~ high impedance) is desirable, which means small position deviations for a given magnitude of force perturbations. The arm admittance comprises inertial, viscous and elastic properties from intrinsic structures (muscles, connective tissues, bones) and reflexive feedback properties through muscle spindles, golgi tendon organs and the central nervous system.

In previous studies that have aimed to characterize the admittance of human limbs, a manipulator was used to impose a mechanical perturbation while the human reaction did not have an effect on the position of the manipulator (Acosta et al., 2000; Cathers et al., 1999; Dolan et al., 1993; Gomi and Osu, 1998; McIntyre et al., 1996; Mussa-Ivaldi et al., 1985; Perreault et al., 2001, 2002; Tsuji et al., 1995; Won and Hogan, 1995; Zhang and Rymer, 1997). In those studies, the subjects were required to exert a constant force while the hand was displaced by the use of different position perturbations, which is considered not to be a very natural task.
Resulting visco-elastic properties from intrinsic and reflexive origin were often explained in terms of contributions to movement stability. However, there is no functional relevance to actively preserve stability because the movement is simply imposed by an inherently stable, position-controlled manipulator having an extremely low admittance.

Force-controlled manipulators, on the contrary, facilitate interaction with compliant environments such as presented in this study. With these manipulators, force perturbations can be applied in combination with position related tasks which comply to natural movement conditions. The effect of the combination of perturbation type and task instruction on the neuromuscular response has been previously studied (Akazawa et al., 1983; Kanosue et al., 1983; Doemges and Rack, 1992; Mirbagheri et al., 2000). These studies showed that muscle spindle responses to force perturbations during a position-holding task were significantly larger than in the case of position perturbations while maintaining a constant force.

In this report, a force-controlled two-degree-of-freedom (2DOF) haptic device is described consisting of a two-linkage anthropomorphic arm having a force sensor at the tip. The manipulator is driven by two powerful hydraulic actuators. A position-servo-based haptic controller is used to increase the endpoint admittance, i.e. decrease the real mass of the linkages to a virtual mass experienced by the subject. The bandwidth is sufficient to capture all the dynamic properties of the human arm, which can be identified using a multivariable system identification method (De Vlugt et al., 2003). With the use of this device, the adaptability of the admittance of the human arm to different environments and types of force perturbations can be analyzed in the horizontal plane, which has never been done before.

Existing, comparable haptic devices are the MIT-MANUS (Colgate, 1988) and the Hopkins manipulandum (Shadmehr and Brashers-Krug, 1997). These manipulators have been designed to study slow, human induced motions and have never been used for the identification of the arm admittance. Another manipulator, which is rather strong and compliant at the endpoint, is the PFM manipulandum (Gomi and Kawato, 1996). The PFM has been used to measure the planar stiffness and viscosity properties of the human arm using force pulses (Gomi and Kawato, 1996; Gomi and Osu, 1998). However, the design and constructional aspects of the present manipulators have never been formulated from the perspective of functional control of the human arm during continuous interaction with (virtual) environments. Also, previous applications were directed at relatively low frequency properties only (stiffness and damping), while the purpose of the present manipulator is to identify the arm admittance over a broad frequency range.

A fundamental problem in haptic control is the occurrence of contact instability when the subject firmly grips the handle (Carignan and Cleary, 2000; Hogan and Colgate, 1989; Van der Linden, 1997). Since the haptic device actively generates mechanical energy, contact instability can pose a direct physical threat to the human subject and needs to be avoided at all times. Contact instability is the result of limited controller bandwidth, which in turn is the result of mechanical resonances of the linkage system. Consequently, the manipulator admittance can only be ‘replaced’ adequately over a limited frequency range. Outside this range, stability cannot be guaranteed. Within the current design, we maximized the bandwidth of the haptic device by the choice of strong and lightweight materials and the optimal adjustment of a servo-based controller. The result is the realization of a compliant and stable environment for worst case loading conditions. The accuracy of the load estimates is tested by comparing the estimated values with the true values of different technical mass-spring systems. Its final application is demonstrated by showing the estimated arm admittance of a subject performing a posture maintenance task.

2. Haptic device

2.1. Manipulator–actuator chain

A diagram of the two-linkage manipulator is shown in Fig. 1A. Both linkages (l = 0.60 m) are constructed as hollow cages and made of 3 mm thick aluminum alloy having a high stiffness to mass ratio. Each linkage rotates on a vertical axis, indicated by the angles \( \theta_1 \) (around the main axis) and \( \theta_2 \) (the secondary axis). The motion is constrained to the horizontal plane. Torques are generated by two identical direct drive hydraulic motors around each axis (supply pressure 120 bar; maximum torque 480 N m; maximum angular velocity 15 rad s\(^{-1}\)). The actuators are vertically aligned on the main axis. One actuates the (inner) first linkage and the other the (outer) second linkage by means of two parallel bars. The mass of the inner linkage is 5.4 kg (including the pull bars) and that of the outer linkage is 3.6 kg. Angular rotation is measured by optical encoders (Heidenhain ROC417, 17 bits per 360\(^\circ\)). Angular velocities are derived analogously. Oil flow is controlled by critical four way valves (Moog D760-2817A) with valve position feedback (300 Hz bandwidth from input voltage to valve position). Pressure difference between both sides of the rotor vane is measured (Paine transducers). Each motor is equipped with a pressure controller (150 Hz bandwidth) to compensate for pressure fluctuations due to movements, leakage and oil compressibility (Heintze et al., 1995). Strain gauges
are mounted in the handle to measure the hand reaction force (range −300 to 300 N) in two orthogonal directions.

Interaction of the human arm with the environment, as simulated by the manipulator, is clarified in the control scheme of Fig. 1B. The total force acting upon the environment is the summation of the hand reaction force $F$ and an independently generated external force perturbation signal $F_{ext}$. The hand position $X$ is the output of the total system.

2.2. Safety system

To prevent the subject from entering the area of motion of the manipulator, the body is securely strapped to the chair back by two shoulder belts (see Fig. 1A). The shoulder rotation center is horizontally strapped to the chair back by two shoulder belts (see motion of the manipulator, the body is securely anchored to the chair by two shoulder belts). The subject is able to stop the manipulator immediately by pressing an emergency button with his or her free hand. To prevent the manipulator from swaying when the subject loosens contact, the manipulator is stopped when the absolute hand reaction force is below 0.1 N for 10 ms.

2.3. Haptic controller

The haptic controller is optimized using linear analysis tools. This implies that optimal controller settings are determined for movements of the manipulator around different positions that are small enough to approximate the system behavior by linear transfer functions. Linear transfer functions facilitate direct access to important system properties in the frequency domain, like resonance frequencies, phase lags and system bandwidth. In accordance with notations used in previous studies, the endpoint admittance, force and stiffness are defined in the Cartesian frame having its origin in the subject shoulder rotation center.

The admittance controller configuration is shown in Fig. 2. The desired trajectory is given by:

$$X_{des}(s) = V(s)(F_{ext}(s) - F(s))$$

with $s = 2\pi f$ the Laplace operator ($f$ the frequency in Hz), $X_{des}(s) = [X_{des,x}(s) X_{des,y}(s)]^T$ the desired endpoint position ($T$ indicates the transposed), $F(s) = [f_x(s) f_y(s)]^T$ the hand reaction force, $F_{ext}(s) = [f_x(s) f_y(s)]^T$ the external force perturbation, and $V(s)$ the multivariable transfer function (MTF) of the virtual endpoint admittance according to:

$$V(s) = \left( \begin{bmatrix} m_{xx} & m_{xy} \\ m_{yx} & m_{yy} \end{bmatrix} s^2 + \begin{bmatrix} b_{xx} & b_{xy} \\ b_{yx} & b_{yy} \end{bmatrix} s + \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} \right)^{-1}$$

(2)

with $m_{ij}$ the virtual mass; $b_{ij}$, virtual damping and $k_{ij}$ the virtual stiffness components ($i, j \in \{x, y\}$).

The servo is angular based such that kinematic transformations to, and from, the Cartesian frame are required. The servo is the closed-loop subsystem from $\Theta_{des}(s)$ to $\Theta(s)$ (Fig. 2), with $\Theta(s) = [\theta_1(s) \theta_2(s)]^T$ the vector of rotation angles. For small deviations around a
mean (not necessarily fixed) reference position, the variation of the reference angles are obtained by inverse transformations from the Cartesian frame:

$$\partial \Theta(s) = J^{-1} \partial \mathbf{X}(s)$$

with $J$ the Jacobian (Craig, 1989):

$$J = \begin{bmatrix} -l_1 \sin \bar{\theta}_1 - l_2 \sin(\bar{\theta}_1 + \bar{\theta}_2) & -l_2 \sin(\bar{\theta}_1 + \bar{\theta}_2) \\ l_1 \cos \bar{\theta}_1 + l_2 \cos(\bar{\theta}_1 + \bar{\theta}_2) & l_2 \cos(\bar{\theta}_1 + \bar{\theta}_2) \end{bmatrix}$$

and $\bar{\theta}_1$ and $\bar{\theta}_2$ the means of $\theta_1$ and $\theta_2$, respectively.

The stability of the haptic device is determined by the loop MTF. Since the admittance control configuration consists of a servo (inner) loop and a force (outer) loop (see Fig. 2), the dynamic properties of both loop MTFs are important to the overall behavior of the haptic device.

The open-loop servo MTF from $\epsilon_o(s)$ to $\Theta(s)$ equals:

$$H_{\text{servo}}(s) = \Theta(s) \epsilon_o^{-1}(s) = G(s) K(s)$$

with $K(s)$ the servo controller to be designed. $G(s)$ is the combined admittance of the manipulator $P(s)$ (in joint coordinates) loaded with the human arm $H(s)$ (in Cartesian coordinates), i.e. from $T_{\text{act}}(s)$ to $\Theta(s)$, according to:

$$G(s) = [I + P(s)J^{-1}H^{-1}(s)]^{-1} P(s)$$

where $H^{-1}(s)$ is the impedance of the human load and $I$ the $2 \times 2$ identity matrix. For the force loop (from $\epsilon_f(s)$ to $F(s)$), the open-loop MTF $H_{\text{force}}$ equals:

$$F(s) \epsilon_f(s)^{-1} = H_{\text{force}}(s) = H^{-1}(s) J_{\text{servo}}(s) J^{-1}(s) F(s)$$

where $H_{\text{servo}}(s)$ is the closed-loop servo MTF according to:

$$H_{\text{servo}}(s) = \Theta(s) \Theta^{-1}(s) = \left[I + H_{\text{force}}(s)ight]^{-1} H_{\text{force}}(s)$$

The constraint to the servo controller, $K(s)$, is that both the servo loop (Eq. (3)) and force loop (Eq. (5)) must be unconditionally stable.

The performance of the haptic device becomes clear when considering the MTF from $\mathbf{F}_{\text{ext}}(s)$ to $\mathbf{X}(s)$, which is derived as follows. From Eq. (5) it follows that:

$$H_{\text{force}}(s) \epsilon_f(s) = F(s)$$

Substituting, $F(s) = H^{-1}(s) \mathbf{X}(s)$ gives:

$$\epsilon_f(s) = H^{-1}(s) H^{-1}(s) \mathbf{X}(s)$$

Substituting $\epsilon_f(s)$ in Eq. (7) by:

$$\epsilon_f(s) = \mathbf{F}_{\text{ext}}(s) - F(s) = \mathbf{F}_{\text{ext}}(s) - H^{-1}(s) \mathbf{X}(s)$$

Fig. 2. Linear block scheme of the admittance controller for the multivariable two-linkage manipulator. The controller consists of two loops; the servo loop having high loop gain to track the desired angles ($\Theta_{\text{des}}$) and the force loop to include the measured interaction force. Both, force sensors and angular encoders are assumed to be ideal. The desired angles are obtained from the virtual dynamics ($V(s)$). Based on the difference between desired and measured angles ($\epsilon_o$), an actuating torque $T_{\text{act}}$ is obtained from the servo controller $K(s)$. The load comprises that of the manipulator $P(s)$ and the subject’s arm $H(s)$. The actual position ($\mathbf{X}$) is imposed to the subject’s arm, which therefore, appears as an impedance ($H^{-1}(s)$). The cascade of the virtual dynamics and the servo offers the virtual admittance the subject experiences (from $F$ to $\mathbf{X}$). The external force perturbation signal ($\mathbf{F}_{\text{ext}}$) is added to the subject’s hand reaction force. The difference ($\epsilon_f$) is the input to the virtual admittance. Transformation from small angular deviations (around a working point) to Cartesian coordinates is indicated by the Jacobian $J$. 


\[ F_{\text{ext}}(s) = [I + H^{-1}(s)]H^{-1}(s)X(s) \]

and finally the combined admittance at endpoint is:

\[ X(s)F_{\text{ext}}(s)^{-1} = H(s)[I + H^{-1}(s)]^{-1} \]

The general purpose of the controller \( K(s) \) is to obtain high loop gains such that the gains of \( H(s) \) are high and consequently \( H_{\text{servo}}(s) \rightarrow 1 \) and \( \Theta(s) \rightarrow \Theta_{\text{def}}(s) \). Also, the force loop (Eq. (5)) will converge to \( H^{-1}(s)V(s) \) such that the combined admittance at endpoint (Eq. (8)) becomes:

\[ X(s)F_{\text{ext}}(s)^{-1} \approx H(s)[I + V^{-1}(s)H(s)]^{-1} \]

\[ = [V^{-1}(s) + H^{-1}(s)]^{-1} \]  

Eq. (9) describes the ideal input–output behavior from external force to endpoint position that is formed by the parallel configuration of the human arm (\( H(s) \)) and the virtual environment (\( V(s) \)), i.e. the manipulator admittance (\( P(s) \)) is perfectly masked.

The virtual filter and the position servo are designed in Simulink and implemented in a 16 bit DSP signal processor (DS1003 60 MHz, DSpace GmBh) at 1 kHz sample frequency.

### 3. Method

#### 3.1. Servo controller optimization

The servo controller \( K(s) \) is optimized to realize a maximum bandwidth of the closed-loop servo (Eq. (6)). The servo controller is implemented as a proportional-differential (PD) controller for each axis:

\[ \tau_{\text{act},n} = k_{p,n} (\dot{\theta}_{\text{des},n} - \dot{\theta}_n) + k_{d,n} (\ddot{\theta}_{\text{des},n} - \ddot{\theta}_n) \]  

with \( \tau_{\text{act},n} \) the actuator torque and \( k_{p,n} \) and \( k_{d,n} \) the gains of proportional and differential action, respectively (\( n \) denotes the actuator, i.e. \( n \in [1, 2] \)).

Because the dynamics of the manipulator and the human arm change with configuration, the servo controller is optimized at five different endpoint positions of the manipulator, being: central [0, 0.45] [m], left [-0.2, 0.35] [m], right [0.2, 0.35] [m], proximal [0, 0.35] [m] and distal [0, 0.55] [m] (numbers adopted from Gomi and Osu, 1998). These coordinates are defined in the subject’s frame and cover a sufficiently wide range of human arm positions. To obtain a stable servo for many different loading conditions, the worst case load is taken, being the subject generating maximum resistance to the force perturbations. Stability is the constraint to the optimizations for which different margins are taken, ranging from wide to rather narrow (Section 3.3.1). Optimizations for all positions are performed simultaneously, resulting in one set of optimal controller gains for each size of the servo stability margin. The optimization is performed independently from the force loop.

#### 3.2. Force loop optimization

Given the optimized servo, the force loop (Eq. (5)) is optimized to find the maximum attainable stable virtual admittance. Apart from the servo, the force loop also comprises the human arm impedance and the virtual admittance. Similar to the optimization of the servo, the worst case loading condition occurs when the human arm generates maximum resistance resulting in a high impedance and consequently in potentially destabilizing high loop gains. The worst contribution of the virtual admittance is when it only has mass terms, which at the same time are small, causing high loop gains at low frequencies and 180° phase lag at all frequencies. Also, five different stability margins are taken as constraints to the optimization of the force loop (Section 3.3.1).

#### 3.3. Identification

To determine the effect of the controller \( K(s) \) on both force and servo loop, the MTFs of the manipulator (\( P(s) \)) and that of the human arm (\( H(s) \)) are estimated to calculate \( H(s) \) and \( H_{\text{servo}}(s) \) (i.e. Eqs. (5) and (3), respectively). To capture all important dynamics, both MTFs are estimated using both input and output measurements containing frequencies up to 100 Hz. Using the closed-loop multivariable identification method from De Vlugt et al. (2003), the (two-input two-output) multivariable frequency response function (MFRF) of the corresponding MTFs can be estimated.

The method requires the hand reaction force, hand position and the independent external force perturbation.

For the estimation of the test loads, a force disturbance signal including frequencies to 20 Hz is used. The reason for this range is that all important mechanical properties of the human arm are excited by these frequencies (Perreault et al., 2001). Estimation accuracy beyond 20 Hz is, therefore, considered not relevant. The time period of observation is 20 s in all cases.

Since MFRFs are only valid descriptions of input–output systems when the underlying system behaves almost linearly, the partial and multiple coherence functions are also estimated as an indication of linearity (De Vlugt et al., 2003; Van der Helm et al., 2002). These functions are equal to one if the system output \((X)\) is a linear function of its input \((F_{\text{ext}})\) and decrease with system nonlinearities, or additional unmeasured inputs like voluntary forces or measurement noise.
3.3.1. Characteristic loci and M-circles

An adequate method to infer stability and performance properties of systems under feedback control is to plot the characteristic loci of the open-loop system transfer functions, i.e. the MFTFs of the force and servo loops in the present case (Maciejowski, 1989). Characteristic loci are the frequency-dependent complex eigenvalues of a multivariable system, describing the dynamic relation between a system input and output signal vector. Another useful tool in control engineering practice is the use of M-circles. These circles apply to the (open-loop) loci and mark the points in the complex plane where the gain of the (closed-loop) system is equal to $M$. M-circles greater than one are used to determine the stability margins. For instance, if all characteristic loci lie outside the $M = 1.3$ circle, the maximum overshoot of the closed-loop system following an input step is less than 30%. Stability margins corresponding to five different M-circles are used for both the servo ($M_s$) and the force ($M_f$) loop, ranging from 1.3 to 5 (Table 1). $M_s$ smaller than one are used to determine the performance of the closed servo loop. One standard value of $M = (1/2)\sqrt{2}$ is used. The frequency where the loci cross this circle equals the bandwidth of the closed-loop MTF, i.e. where $|H_{\text{servo}}(s)| = (1/2)\sqrt{2}$.

As an example, Fig. 3 shows the characteristic loci of the servo loop for the central position and an initial (stable) set of controller gains, being: $k_{p,1} = k_{p,2} = 2400$ and $k_{d,1} = k_{d,2} = 24$. The direction of increasing frequency ($f$) is indicated by arrows. The circle on the left is the $M_s = 1.3$ circle and the larger circle on the right is the $M_s = (1/2)\sqrt{2}$ circle. The loci do not cross the $M_s = 1.3$ circle, meaning that the servo is stable and sufficiently damped for these initial controller gains. These gains appeared to be stable for the other positions also, so that all the necessary MFRFs of the manipulator and those of the subject’s arm admittance were estimated and used to determine the optimal controller gains (Section 3.3.2). The $M_s = (1/2)\sqrt{2}$ circle is crossed at 2 and 6 Hz for the first and second locus, respectively (Fig. 3).

<table>
<thead>
<tr>
<th>$M$</th>
<th>$\varphi$ (°)</th>
<th>$\Delta$ (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.3</td>
<td>50</td>
<td>0.44</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>0.33</td>
</tr>
<tr>
<td>3</td>
<td>20</td>
<td>0.25</td>
</tr>
<tr>
<td>4</td>
<td>15</td>
<td>0.20</td>
</tr>
<tr>
<td>5</td>
<td>12</td>
<td>0.17</td>
</tr>
</tbody>
</table>

The phase and amplitude margins are determined with respect to the point $[-1, 0]$ in the complex plane: phase margin $\varphi$ equals the remaining angle at unity gain; amplitude margin $\Delta$ equals the remaining amplitude at $-180°$ phase shift.

3.3.2. Implementing the optimization

The servo controller gains are optimized by minimizing the following cost function:

$$ P(k) = P_s(k) + P_h(k) $$  \hspace{1cm} (11)

where $P_s(k)$ is related to the servo performance (desired effect) and $P_h(k)$ to high-frequency amplification beyond the servo bandwidth (undesired effect, i.e. actuator overdrive). Both functions are dependent on the controller parameter vector $k = \{k_{p,1}, k_{p,2}, k_{d,1}, k_{d,2}\}$ (see Eq. (10)).

For the servo performance, the bandwidths of both servo loci are taken as direct measures according to:

$$ P_s(k) = \frac{1}{f_{b,1}(k)} + \frac{1}{f_{b,2}(k)} $$  \hspace{1cm} (12)

with $f_{b,1}$ and $f_{b,2}$ the bandwidth of the first and second locus, respectively.

The usage of $P_h$ in Eq. (11) is to prevent the actuator torque from varying too quickly. For the hydraulic actuators in the present case, the pressure differences in the valves would otherwise be too large leading to undesired noise in the pressure control loops and annoying ‘sawing’ sounds from highly turbulent oil flows. High-frequency excitation is undesired in general and not a specific issue in the hydraulic case in this study. In a preliminary experimental setup we experienced comparable high-frequency noise in the current controller of an electrically powered wrist manipulator.

The common problem is that differential (D)-action is required inside the servo bandwidth to induce the necessary phase lead, which at the same time amplifies the gains at frequencies beyond the bandwidth. Therefore, excessively high differential (D)-action in the servo controller is penalized by weighting the gains of the servo loci above 25 Hz in the present case, which approximately equals the frequency where the first locus of the manipulator admittance starts to increase again (Fig. 4A). The following cost function $P_h$ to be minimized is, therefore, taken:

$$ P_h = c \sum_{nM=25}^{nM=100} |\lambda_{s_1}(2\pi n\Delta f)| + |\lambda_{s_2}(2\pi n\Delta f)| $$  \hspace{1cm} (13)

with $\lambda_{s_1}, \lambda_{s_2}$ the servo loci and $c$ a weighing factor. Different values for $c$ are used, being 0.25, 0.50, 0.75, 1.00, 1.25 and 1.50.

The constraint to the minimization of Eq. (11) is that the servo loci may not cross the stability boundaries given by the $M_s$-circles. For each value of $c$ and for each value of the stability margin $M_s$ (30 combinations), the characteristic loci of the servo loop MTFs (Eq. (3)) are optimized by minimizing $P(k)$.

Given the optimized servo, the force loop is optimized by determining the minimal virtual mass for each combination of $M_s$ and $c$ and for the five different
values of $M_f$ (resulting in 150 values of the virtual mass). This is achieved by sizing the loci of the force loop for different values of the virtual mass such that the loci just touch, but do not intersect the $M_f$-circle. The virtual mass is taken isotropically for simplicity, such that it can be described by one parameter: $m_v = m_{xx} = m_{yy}$ with $m_{xy} = m_{yx} = 0$.

### 3.4. Test loads

The overall quality of the optimally controlled haptic device is judged on two properties: the accuracy with which the desired virtual admittance is simulated and the ability to retrieve reliable estimates of the human arm.

The estimator is tested by identification of known mass-spring loads. The estimated load is then compared with the true one. For this purpose, different combinations of masses and springs are attached to the handle of the manipulator in the central position. All loads are taken symmetrically. Three different isotropic masses were used, weighing 0.99, 1.30 and 1.61 kg. With the smallest mass, four spring configurations were applied: the combination $(k_{xx}, k_{yy})$ was taken equal to (300, 300), (600, 300), (300, 600) and (600, 600), respectively (values in N m$^{-1}$). The values are taken as a representative range for the human arm stiffness (Tsuji et al., 1995). Due to the configuration of the springs, the actual stiffness in one direction is increased by the springs in the perpendicular direction. A schematic representation of the spring configuration and the derivation of the true stiffness values is presented in Appendix A. To judge the accuracy of the estimates, a second order mass-spring model has been fitted in the frequency domain through the diagonal elements of the estimated FRFs using a straight forward least squares algorithm. For all combinations of the test loads, the virtual admittance of the manipulator was set at zero stiffness and a symmetric mass of 5 kg. A small amount of virtual damping (5 N s m$^{-1}$) was used to suppress oscillations of the endpoint. In the case where the test load includes no springs, the virtual stiffness was set to a small value of 50 N m$^{-1}$ to exclude drifting of the endpoint position.

After the estimation accuracy is determined, the admittance of the virtual environment is estimated and compared with the real admittance of the manipulator. The estimation is performed while the subject holds the handle and performs the maximal resistance position.
maintenance task. The difference is expressed as the ratio in eigenvalues of the estimated virtual mass ellipse and the real mass ellipse. The mass ellipse is graphically displayed as a force vector in response to a unit acceleration vector generated by sin and cos functions, according to:

Fig. 4. Estimated frequency response functions (FRFs) of the characteristic loci (gain and phase plots) of the manipulator. (A) First locus of the admittance of the manipulator loaded with the human arm, i.e. the combined admittance $G(s)$ (solid lines), and that of the unloaded manipulator $P(s)$ (dashed lines). (B) The second locus; loaded (solid lines) and unloaded (dashed lines). The loci correspond to the MFRF between the actuator torque ($T_{act}(s)$) and rotation angles ($\theta(s)$).
\[
\begin{bmatrix}
f_1 \\ f_2
\end{bmatrix} = M \begin{bmatrix}
\cos \phi \\ \sin \phi
\end{bmatrix}
\]
for \(0 < \phi < 2\pi\) and where

\[
M = \begin{bmatrix}
m & 0 \\ 0 & m
\end{bmatrix}
\]

is the endpoint mass matrix (for small excursions of the endpoint around a fixed position). Since the mass is taken isotropically, the ellipse is a circle of which the two eigenvalues are equal, i.e. \(\lambda_1 = \lambda_2 = m\). In contrast, the real mass of the manipulator at endpoint is displayed as an ellipse having non-zero off-diagonal terms and unequal diagonal terms, which is a general property of a chain of rotating linkages (Craig, 1989).

### 3.5. Subject

The arm admittance of one subject (male, 29 years) was estimated. The subject gave informed consent to the experimental procedure. The experiment was carried out with the right arm. The subject was asked to take a firm grip on the manipulator handle (Fig. 1). Gravitational forces were compensated for by supporting the upper arm with a brace that was fixed by a rope to the ceiling. The subject had positioned his or her hand in the center position. The force perturbation always started after the subject had positioned his or her hand in the center of the reference circles.

### 4. Results

#### 4.1. Manipulator dynamics

Fig. 4 shows the estimated characteristic loci, presented as FRFs, of the manipulator loaded with the human arm, i.e. the combined admittance, \(G(s)\) (dark lines), and the unloaded manipulator, \(P(s)\) (dashed lines). Distinct troughs and strongly alternating phase shifts are clearly seen, especially for the first locus (Fig. 4A), and are typical for systems with distributed regions of low stiffness. The gain of the first locus of the combined admittance exhibits two collapses around 22 and 53 Hz. The loci of the manipulator only are largely comparable to those of the combined admittance. The main differences are the increase of the frequency where the first trough occurs and the absence of the second trough. Both differences are the result of changes in eigenfrequencies due to the additional load of the human arm.

#### 4.2. Controller gains

The optimized servo bandwidth and minimal virtual mass are given in Fig. 5 for high-frequency weighting \(c = 1.25\). Lower weighting of the high-frequency gains in the servo loop \((c < 1.25)\) resulted in hydraulic overdrive meaning that the corresponding controller gains were not of practical use. For the largest weighting \((c = 1.5)\) the controller was too conservative resulting in a smaller bandwidth and a slightly larger minimum for the virtual mass. Because the bandwidth of the second locus is substantially higher (\(> 30\) Hz), only the bandwidth of the first locus, \(f_b,1\), is given. The bandwidth increases with decreasing servo stability margins (increasing values of \(M_f\)). The minimal virtual mass also increases with \(M_s\) for the smallest stability margins of the force loop \((M_f = 5)\) and remains almost the same for the largest margins \((M_f = 1.3)\). It is clear that there is a trade-off between bandwidth and minimal attainable virtual mass. As the best choice between smallest virtual mass and highest bandwidth, we have chosen \(m_e = 1.72\) kg and \(f_{b,1} = 12.9\) Hz, corresponding to \(M_f = 2\) and \(M_f = 5\) as indicated by the encircled values in Fig. 5. The corresponding controller parameters are \(k_{p,1} = 12432.0\), \(k_{p,2} = 6240.0\), \(k_{d,1} = 244.8\), \(k_{d,2} = 8.6\).

The values of the bandwidth are minima, and always occurred at the proximal position where the manipulator was highly extended. The highest bandwidth always occurred at the distal position and was approximately 20% higher.

The optimal loci of the servo are shown in Fig. 6A for the proximal (limiting) position. The bandwidths are indicated by filled dots on the crossing point of the loci with the \(M = (1/2)\sqrt{2}\) circle. Fig. 6B shows the loci of the force loop. The ‘touch’ with the corresponding \(M\)-circles is clearly visible and always occurred for the second locus.

Fig. 7 shows the four FRFs of the two-by-two servo MFRF in the Cartesian frame. The decomposition of the endpoint MFRF into its four Cartesian FRFs is a common way of studying the planar admittance and is used as the default representation hereafter. The \(x\)-direction is restrictive, having a bandwidth comparable to the first locus of the MFRF \((\approx 13\) Hz). In the \(y\)-direction, the bandwidth is much higher (\(> 20\) Hz) and comparable with the second locus of the MFRF. Within the smallest bandwidth, the Cartesian directions are reasonably decoupled, as indicated by the reduced gains of the cross-terms. For both loading conditions in all positions, the multiple coherence is higher than 0.85, indicating highly linear behavior of the servo.
4.3. Test load estimation

Fig. 8A shows the estimations and model fits in the case of the smallest test mass without springs. Fig. 8B shows the results in the case of added springs. The estimated FRFs show the typical responses of mass-spring systems. In the case of the mass load, the gain decreases monotonically with frequency decade at a constant phase lag of $-180^\circ$. Added springs decrease the low frequency gain and introduce an undamped oscillation peak at the eigen-frequency. The estimated cross-terms are much smaller compared with the diagonal terms. This is expected because all loads are purely diagonal. An additional indication of the amount of coupling in the (combined) system is given by the partial coherence functions, shown in Fig. 9. For the cross-terms, the partial coherence functions are almost zero within the servo bandwidth. On the contrary, values close to one are found for the diagonal terms indicating that the output in one direction is mainly determined by the input in the same direction within the bandwidth.

The estimated multiple coherence functions are high ($>0.85$ up to 15 Hz), indicating highly linear behavior. The model fits are accurate for all load combinations and yield parameter estimates showing negligible deviations from the true values ($R > 0.9995$) with absolute differences smaller than 2% of the true value.

4.4. Human arm admittance estimation

Fig. 10A shows the estimated MFRF of the subject in the central position. The estimated FRFs show rather flat or slightly declining gains with frequency to approximately 6 Hz and decline steeper with further increase of frequency. The corresponding phase lag increases from 0 to $-180^\circ$ for the diagonal terms. Oscillatory behavior appears around 6 Hz. In the other positions, the magnitudes of these oscillations were different (or sometimes absent) while the oscillation occurred within the same frequency range (not shown). Multiple coherence functions are high ($>0.75$). The estimated FRFs of the virtual admittance closely resemble those of the desired admittance, being an
isotropic mass of 1.72 kg (Fig. 10B). Around 13 Hz, the
FRFs deviate slightly from the intended ones (dotted
lines), in particular in the x-direction. The multiple
cohesion functions indicate highly linear behavior over the
whole frequency range.

Fig. 11 shows the ellipses of the real mass of the
manipulator (dotted) and the estimated virtual envi-
ronment (solid). The eigenvalues of the real mass matrix of
the manipulator are \( \lambda_{1,\text{real}} = 4.97 \) kg and \( \lambda_{2,\text{real}} = 0.90 \)
kg. The eigenvalues of the estimated virtual mass ellipse
are equal to the minimal attainable mass, i.e. \( \lambda_{1,\text{vir}} \)
\( \lambda_{2,\text{vir}} = 1.72 \) kg. The real mass is, therefore, reduced by
\( \frac{\lambda_{1,\text{vir}}}{\lambda_{1,\text{real}}}/4.97 \) to 35% while in the opposite di-
rection the real mass is increased by a factor \( \frac{\lambda_{2,\text{vir}}}{\lambda_{2,\text{real}}}/1.72/0.90 \) (almost doubled).

5. Discussion

5.1. Stability and performance of the haptic device

The servo controller has a minimal bandwidth of 12.9
Hz. Within this frequency range, the virtual admittance
is almost equal to the desired one provided by the virtual
dynamics (Fig. 10). The real mass of the manipulator is
strongly reduced in the linear direction of the outer
linkage (35%) but is almost doubled in the opposite
direction. Because the virtual mass is minimized in two
directions (x and y) equally, due to its prescribed
isotropic form, the minimum is probably not the
ultimate for each direction individually. That is, the
optimum perhaps elicits a virtual mass that is more
ellipsoidal when optimized in both directions indepen-
dently. The facility to use an ellipsoidal virtual mass, how-
ever, is not of prime interest. Rather, changing the
virtual stiffness or damping in different directions seems
a more logical approach to investigate the direction
dependent visco-elastic disturbance behavior of the
human arm (De Vlugt et al., 2002).

The minimal value of the virtual mass is 1.72 kg and is
the result of the most appropriate choice between a least
dominant environment and a large servo bandwidth.
The phase and amplitude margins are sufficient for the
servo loop \( (M_s = 2) \) but narrow for the force loop
\( (M_f = 5) \). We have successfully validated stable interaction
in the case of several goal-directed movements
within the boundaries given by the separate positions. In
the case contact instability still occurs, more conserva-
tive controller gains (at the cost of bandwidth) or a
larger virtual mass should be taken.

Due to the material stiffness, the gain of the manip-
ulator admittance does not decay for higher frequencies
(Fig. 4A). Consequently, the increase of the servo loop
gain by the controller is limited to avoid actuator
overdrive, and thus the desired virtual admittance
cannot accurately be realized (Eq. (9)). This indicates
the need to use stiff and lightweight materials for all
moving parts (Carignan and Cleary, 2000).
When wide servo stability margins are imposed \((M_s = 1.3)\), D-action is necessary at the cost of additional P-action. In contrast, for small stability margins D-action is not required and is, therefore, directly penalized by high-frequency weighting. In that case, additional P-action is allowed and beneficial for increasing the bandwidth at the cost of a higher virtual mass. The latter is explained by the lack of phase advance from the D-action. The trade-off illustrates the conflict of two desired effects: high phase advance inside the bandwidth and sufficient gain attenuation outside the bandwidth. Therefore, the application of several stability boundaries and control effort weighting at high frequencies is a practical and efficient method to adjust the servo controller.

An important aspect regarding the stability of the haptic device in general is the sampling frequency of the controller. Discretization (sample and hold circuits) introduces phase lags that increase with frequency. For the present case, the sampling frequency of 1 kHz appeared to be the upper limit gained with our hardware, which can be regarded as sufficient for these types of admittance controlled applications (Carignan and Cleary, 2000).

Regarding contact instability as the result of badly controllable and oscillatory states, the addition of any dissipative elements can be used to suppress those oscillations. The first location where those elements could be applied is in the force loop, by simply providing the virtual dynamics (Eq. (2)) with damping terms that directly reduce phase lags from \(-180\) to \(-90\) at the lower frequencies. In this case one has to accept a certain minimal virtual damping to be present which limits the haptic device in its range of virtual admittances. Energy dissipative elements can also be applied in the servo loop, by adding an arrangement of physical dashpots to the linkage system. It is expected that such a facility will improve the servo but also at lower frequencies only. Furthermore, such an arrangement will increase the inertia of the device, which in turn will enhance oscillatory behavior.

5.2. Test load estimation

The estimation of the load is accurate over the whole frequency range to 20 Hz. The reason is that the applied closed-loop estimator separately estimates the load admittance irrespective of the properties of the environment (De Vlugt et al., 2003). This property only holds for the condition that the servo does not introduce too much noise beyond its bandwidth. Since the combined system response is highly linear, as proved by the high
multiple coherence functions in all cases (Figs. 9 and 10), the limited bandwidth does not affect the estimation of the MFRF of the human arm admittance.

High correlations between the true and estimated parameters of the mass-spring loads validate the accuracy of the identification procedure. The validation is performed on the diagonal elements only. This is because it was not possible to create cross-terms in the load configuration with the test frame as used in this study (see Fig. 12 in Appendix A). For that purpose, a

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Fig. 8. Estimated FRFs (solid lines) and model fits (dotted lines) for two different test loads, isotropic mass of 0.99 kg (A) and the same mass with diagonal spring stiffness in $x$-direction $k_{xx} = 600$ N m$^{-1}$ and in $y$-direction $k_{yy} = 300$ N m$^{-1}$ (B).
more sophisticated test bed would have to be used that is morphologically comparable with the human arm. Such a facility comprises multi-DOF rotating levers with additional (and adjustable) joint visco-elasticity. Such an elaborate mechanism is much harder to construct and was not at hand.

If springs are used as test loads, wave propagation (longitudinal and transversal) of the springs probably causes additional reaction forces that are (partly) uncorrelated with the external force perturbation and, therefore, reduce the multiple coherence above 10 Hz (Fig. 9).

5.3. Human arm admittance estimation

Some general features seen in the estimated arm MFRF that reflect underlying mechanical properties will be briefly discussed. The endpoint MFRF measured at the hand is comprised of the rotational dynamics around the wrist, elbow and shoulder joint. Assuming that the joint impedance MTF is of second order, the joint admittance MTF (matrix inverse of the impedance MTF) is then of sixth order, i.e. three second order systems in series. Consequently, the endpoint admittance of the arm is also of sixth order, as it follows from pre- and post-multiplication of the joint admittance MTF by the Jacobian (J) and its transposed (J^T), respectively. For the shoulder-elbow combination (fixated wrist) a moderate amplification around 2 Hz was measured (amongst others) by Gomi and Osu (1998) and Perreault et al. (2001). In those studies, the subjects performed a submaximal force task while in the present study maximal performance position tasks were performed resulting in higher joint stiffness values. Such an oscillation was not clearly seen in our estimates (Fig. 10A, upper part). The resonance at 6 Hz probably indicates the influence of the wrist.

5.4. Other control strategies

Generally, there are two control strategies that can be used in haptic control: feedforward and feedback control.

5.4.1. Feedforward control

Feedforward control is based on forward coupling of either the measured interaction force or the endpoint position. Feedforward controllers enforce accurate models of the manipulator admittance to completely cancel out all the dynamics and insert any virtual admittance instead. The cancellation is performed using the inverted admittance model (impedance) yielding...
high amplification at high frequencies, resulting in unacceptably fast changes of the actuator control signals, i.e. actuator overdrive. Accurate modeling of the admittance is possible for the low frequency dynamics but is almost impossible for higher frequencies due to mechanical resonance resulting from material weakness. Imperfect cancellation will easily lead to unstable interaction. Additionally, the resonance frequencies shift with configuration of the linkages, such that a generic model will end in an excessively high computational burden. For these reasons, feedforward control is not appropriate for the current application.

5.4.2. Feedback control

The other control strategy is based on feedback control which does not require accurate system knowledge (Hogan, 1985). Feedback control for haptic applications comes within two basic configurations:

Fig. 10. (A) Estimated MFRFs (gain and phase) of the subject’s endpoint admittance and of the virtual environment it interacts with (B) in the central position. Bottom row, multiple coherence functions. The desired endpoint admittance, corresponding to the minimal virtual mass of 1.72 kg, is also shown (dotted lines).
impedance and admittance control. The latter is applied
in this study. Both consist of an inner feedback loop to
track the desired endpoint force (impedance control) or
endpoint position (admittance control). The interaction
force is included by an outer feedback loop. The benefit
of feedback is that excellent high bandwidth tracking
can be established when high gain of the inner feedback
loop can be realized. Again, mechanical resonance
prevents high gains which leads to limited servo
bandwidths.

As the result of a limited bandwidth, and therewith an
improper (but stable) realization of the virtual admit-
tance, the force loop can become unstable for high loop
gains, i.e. the problem of contact instability (Adams and
Hannaford, 1999; Carignan and Cleary, 2000; Hannaford
and Jee-Hwan, 2002; Van der Linden, 1997). High
force loop gain occurs when the human impedance is
high, e.g. when one tightens the grip and fully cocon-
tracts the muscles around the shoulder, elbow and wrist
joints. In that case, the human load causes force loop
gains that increase with frequency.

Comparing both feedback controllers, impedance
control has a practical drawback and that is the
requirement to measure the acceleration for adaptation
of the virtual mass. Accelerometers are commercially
available in different types but are less accurate than
force sensors. Also, on-line differentiation of the posi-
tion is very susceptible to noise and slow drift. Con-
sidering this drawback, an admittance controller has
been chosen for the current application.

Fig. 11. Mass ellipses at the endpoint of the manipulator (central
position) in the subject's coordinate frame. Dark lines schematically
represent the linkages of the manipulator. Estimated virtual mass
(solid) and real mass of the manipulator (dotted). The radius of the
calibration circle corresponds to 1 kg. Mass values in both principle
directions are indicated by the corresponding eigenvalues.

Fig. 12. Technical spring system used for calibration, consisting of four linear springs at 90° angles (see inset) constituting a static equilibrium (point A). $K_x$, stiffness of springs in $x$ direction; $K_y$, stiffness of springs in $y$ direction. The stiffness in one direction (either $x$ or $y$) consists of a longitudinal (linear) and a tangential part. A small movement in the $y$ direction is visualized to elaborate on the stiffness contribution in the $x$ direction. See text for further explanation and derivation of the resulting stiffness field.
6. Conclusions

The haptic manipulator presented in this study facilitates the analysis of human arm motion control during natural interaction with compliant environments. It offers a stable and accurate testbed to estimate the mechanical admittance of the human arm at different positions in the horizontal plane. From the mechanical admittance, important physiological properties related to intrinsic and reflexive muscle mechanisms can be analyzed to study the adaptability of the human arm neuromusculature to deal with different types of environments.

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Appendix A: Spatial derivation of the true stiffness field

The total stiffness field of the applied technical spring system at the interconnection point A (Fig. 12, inset) equals:

\[ K = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{xy} & K_{yy} \end{bmatrix} \]  

Since the configuration is taken to be symmetrical, no force in the y direction emerges when moving in the x direction and vice versa, i.e. \( K_{xy} = K_{yx} = 0 \). The stiffness field is, therefore, only determined by the diagonal terms being:

\[ K_{xx} = \frac{\partial F_x}{\partial x} \]

\[ K_{yy} = \frac{\partial F_y}{\partial y} \]

E.g. the stiffness \( K_{yy} \) is derived from a movement of the interconnection point in the y direction (from A to B, Fig. 12; springs in the y direction are omitted for clarity). Similar derivation holds for the stiffness \( K_{xx} \).

At initial lengths \( a_0 \) (O-A, (null) gray springs) the force exerted by one spring in the x direction (x spring) \( F_{x_0} \) equals:

\[ F_{x_0} = K_x(a_0 - l_0) \]

with \( K_x \) the stiffness of the x springs and \( l_0 \) the length at zero spring force. In the case where the connection point is moved from A to B, the x spring forces increase to:

\[ F_x = F_{x_0} + K_x(a - a_0) \]

\[ = K_x(a_0 - l_0) + K_x(a - a_0) \]

\[ = K_x(a - l_0) \]  

(16)

with \( a \) the x spring length (O-B). In point B, the force in the y direction \( F_y \) equals:

\[ F_y = 2K_y y + F_x \sin \alpha \]

with \( K_y \) the stiffness of the y springs. Substituting Eq. (16) (for both x springs) into Eq. (17) gives:

\[ F_y = 2K_y y + 2K_x(a - l_0) \sin \alpha \]  

(18)

Expressing \( a \) and \( \sin \alpha \) into the initial length \( a_0 \) and the displacement \( y \):

\[ a = \sqrt{a_0^2 + y^2} \]

\[ \sin \alpha = \frac{y}{a} = \frac{y}{\sqrt{a_0^2 + y^2}} \]

and substituting into Eq. (18) gives:

\[ F_y = 2K_y y + 2K_x(\sqrt{a_0^2 + y^2} - l_0) \frac{y}{\sqrt{a_0^2 + y^2}} \]

\[ = 2K_y y + K_x \left( \sqrt{a_0^2 + y^2} - l_0 \right) \frac{y}{\sqrt{a_0^2 + y^2}} \]  

(19)

Derivation of Eq. (19) around \( y = 0 \) results in the stiffness \( K_{yy} \):

\[ \frac{\partial F_y}{\partial y} \bigg|_{y=0} = 2 \frac{\partial}{\partial y} \left[ K_y y + K_x \left( \sqrt{a_0^2 + y^2} - l_0 \right) \frac{y}{\sqrt{a_0^2 + y^2}} \right] \]

\[ = 2K_y + K_x \left( \sqrt{a_0^2 + y^2} - l_0 \right) \frac{1}{\sqrt{a_0^2 + y^2}} - K_x l_0 y \frac{\partial}{\partial y} \left( \frac{1}{\sqrt{a_0^2 + y^2}} \right) \]

\[ = 2K_y + K_x \left( \sqrt{a_0^2 + y^2} - l_0 \right) \frac{1}{\sqrt{a_0^2 + y^2}} + K_x l_0 y^2 \left( \frac{1}{\sqrt{a_0^2 + y^2}} \right)^3 \]

\[ = 2K_y + K_x \left( 1 - \frac{l_0}{a_0} \right) \]  

(20)

In the case where the springs have zero initial force (\( F_{x_0} = 0 \)) such that it is possible to let \( a_0 \) be equal to \( l_0 \), the additional stiffness of the x-springs is zero, i.e. \( K_{yy} = 2K_y \). Normally, springs do have an initial force at rest length (pre-tension) such that \( a_0 > l_0 \) and so the x springs increase the net stiffness in the y direction.

In summary, for small displacements around the initial equilibrium position \( y \), the stiffness field is diagonal and equal to:
\[ K_{xx} = 2 \left( K_x + K_y \left( 1 - \frac{I_y}{d_y} \right) \right) \]
\[ K_{yy} = 2 \left( K_x + K_y \left( 1 - \frac{I_y}{d_y} \right) \right) \]

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