PASSIVE PNEUMATIC TELEOPERATION SYSTEM

Aly Elmasry, and Matthias Liermann
Mechanical Engineering department
American University of Beirut
P.O.Box 11-0236
Riad El Solh
Beirut 1107-2020, Lebanon
Alyelmasry0@gmail.com
matthias.liermann@aub.edu.lb

ABSTRACT
This paper describes the modeling and control of a pneumatic tele-operation scheme, where the connection of master and slave cylinder is realized signal based but also physically via long transmission lines. The system is called passive because of the passive physical connection, which can also serve as a safety fallback solution. The advantage of this scheme is that a limited force feedback is realized with a minimum of extra effort in comparison to a teleportation system without force feedback. The stiffness of the physical connection is enhanced through a cascaded position and pressure control scheme with two proportional valves as actuators for each pneumatic line. The paper presents the mathematical model of the setup, which is used to determine the relative stability of the dynamic system as a function of control parameters. An experimental setup is presented which was set up to validate the system model. For a distance of 5 m between master and slave cylinder a stiffness of \(2.4 \frac{N}{mm} \) could be established.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Cross sectional area of pneumatic line</td>
<td>(m^2)</td>
</tr>
<tr>
<td>(A_p)</td>
<td>Annulus area of piston</td>
<td>(m^2)</td>
</tr>
<tr>
<td>(b)</td>
<td>Critical pressure ratio</td>
<td></td>
</tr>
<tr>
<td>(B_h)</td>
<td>Damping constant of operator hand</td>
<td>N.s/m</td>
</tr>
<tr>
<td>c</td>
<td>Sonic conductance of valves</td>
<td>(m^3/Pa.s)</td>
</tr>
<tr>
<td>D</td>
<td>Inner diameter of pneumatic lines</td>
<td>m</td>
</tr>
<tr>
<td>(f)</td>
<td>Highest frequency of interest</td>
<td>Hz</td>
</tr>
<tr>
<td>(F_h)</td>
<td>Force applied by operator hand</td>
<td>N</td>
</tr>
<tr>
<td>(k_h)</td>
<td>Spring constant of operator hand</td>
<td>N/m</td>
</tr>
<tr>
<td>(K_p)</td>
<td>Inner control gain conversion factor</td>
<td>V/Pa</td>
</tr>
<tr>
<td>(K_s)</td>
<td>Outer control gain stiffness</td>
<td>N/m</td>
</tr>
<tr>
<td>(K_v)</td>
<td>Electromechanical valve gain</td>
<td>1/V</td>
</tr>
<tr>
<td>L</td>
<td>Length of pneumatic lines</td>
<td>m</td>
</tr>
<tr>
<td>(l_{vl})</td>
<td>Length of master and slave</td>
<td>m</td>
</tr>
<tr>
<td>(m_h)</td>
<td>Mass of operator hand</td>
<td>kg</td>
</tr>
<tr>
<td>(m_{1m})</td>
<td>Mass flow rate into chamber 1 of master</td>
<td>kg/s</td>
</tr>
<tr>
<td>(m_{l1})</td>
<td>Mass flow rate into line 1</td>
<td>kg/s</td>
</tr>
<tr>
<td>(m_p)</td>
<td>Mass of piston assembly</td>
<td>kg</td>
</tr>
<tr>
<td>(\dot{m}_{vl})</td>
<td>Mass flow rate delivered by valve 1</td>
<td>kg/s</td>
</tr>
<tr>
<td>(P_{m1})</td>
<td>Pressure of chamber 1 of master</td>
<td>Pa</td>
</tr>
<tr>
<td>(P_{ul})</td>
<td>Upstream pressure of valve 1</td>
<td>Pa</td>
</tr>
</tbody>
</table>
1 INTRODUCTION

Teleoperation means that the motion of a machine slave is controlled remotely, typically by a human operator in interaction with a master device. It is associated with robotics and used whenever it is not feasible for the human operator to physically interact with an object directly. It could be because of hazardous environment, physical and geometrical constraints or just remoteness. Examples for teleoperation systems are found in many areas such as marine exploration, robotic surgery, surveillance and inspection, nuclear engineering, space exploration, or search and rescue missions [1]. The term teleoperation does not specify the level and quality of interaction with the remote environment. This depends on the nature of information, which is fed back to the user, it could be visual, audio, tactile (touch) or haptic (force), among others. The character of visual and audio feedback is one-way or unilateral. This information is transmitted only from the remote location of the operator. Ideally, teleoperation systems should be “transparent”.

The operator should find it easy to imagine that he or she is directly interacting with the remote object. This requires tactile or haptic feedback. Haptic feedback is always bilateral [2]. The load measured at the slave device is used to control the force experienced by the operator on the master device. At the same time the force between operator and master influences the motion of the master device, which is the input for the position control loop. It has been shown in [3], that in principle, transparency and robust stability are conflicting design goals, which is why complete transparency cannot be practically achieved. Bilateral feedback using position and force feedback has to be designed with a compromise in mind between stiff transmission on the one hand, and an accurate reproduction of end-effector force for the operator on the other hand. Even though force feedback potentially improves quality and speed of teleoperation missions, it is often not used because it significantly increases the system complexity [4]. The complexity increases in the master and slave instrumentation and actuation as well as in the feedback control implementation. Force-feedback requires force or pressure sensing. The slave end-effector needs to be capable of measuring or estimating the force of interaction with the object and the master needs to be equipped with motors and force sensors as well to reproduce the reaction forces. Therefore a bilateral feedback scheme for two fully instrumented actuators is needed. The measured quantities are master and slave positions and forces. Both master and slave actuators are controlled with dedicated control units, the master and slave servers. The servers receive position and/or torque references through a bilateral controller, [2, 5-7].

Teleoperation systems require precision motion control. Most small scale applications use electric motors, while slaves designed to handle large forces are often driven by hydraulics. In special applications with high demands on cleanliness, explosion protection or non-magnetic behavior, pneumatic actuators are a natural choice. For example for use in Magnetic Resonance Imaging (MRI), pneumatic tele-robotic actuators have been developed [8-9]. Pneumatic tele-operation schemes for telesurgery with force-feedback are discussed in [10-11]. Irrespective of the kind of actuation, whether electric, hydraulic or pneumatic, the benefit of connecting forces and motion of master and slave through electronic communication is that manipulation over great distances is possible. Also with velocity and force feedback for both master and slave it is possible to achieve a high degree of transparency [3]. However, often, such as in biomedical robotic applications, the operator is near the patient, in the same or an adjacent room.

\[ P_{d1} \] Downstream pressure of valve 1 \[ \text{Pa} \]

\[ R \] Ideal gas constant \[ \text{N.m/s} \]

\[ T \] Gas temperature \[ \text{K} \]

\[ T_r \] Temperature of air at reference conditions \[ \text{K} \]

\[ T_{12} \] Upstream temperature of valve 2 \[ \text{K} \]

\[ u_1 \] Valve 1 input \[ V \]

\[ V_{lm} \] Volume of chamber 1 for master \[ m^3 \]

\[ V_{md} \] Dead volume of chamber 1 for master \[ m^3 \]

\[ \mu \] Mean air velocity \[ m/s^2 \]

\[ x_h \] Position of operator hand \[ m \]

\[ x_m \] Position of master \[ m \]

\[ x_s \] Position of slave \[ m \]

\[ x_{v1} \] Spool position of the valve 1 \[ m \]

\[ \beta \] Damping constant of cylinders \[ N.s/m \]

\[ \Delta P_{f} \] Pressure loss due to friction \[ Pa \]

\[ \zeta \] Damping ratio of the valves \[ \text{rad/s} \]

\[ \nu \] Air kinematic viscosity \[ m^2/s \]

\[ \mu \] Air dynamic viscosity \[ Pa.s \]

\[ \rho_0 \] Density of air at reference conditions \[ \text{rad/s} \]

\[ \omega \] Undamped natural frequency \[ \text{rad/s} \]
In this case, the effort and cost for actuation and instrumentation can be reduced to a large extent. For small to medium distances the purely electrical connection can be replaced using pneumatic connections, thereby reducing the cost and complexity. In the following section we present a pneumatic teleoperation scheme where slave and master actuators are connected through a pneumatic line. This means that we cannot control the motions and forces of master and slave independently. Instead we make use of the fact that they are coupled and would like to explore the usefulness of such an approach. We develop its mathematical model in section 3 and derive a linearized model in section 4. This linear model is analyzed in section 5. We discuss how the control parameters for a cascaded stiffness control influence the stability margin and how a reasonable compromise between stability robustness and transmission stiffness can be found. In section 6 we explain the experimental setup and compare the nonlinear simulation results with measurement results.

2 PROPOSED SIMPLIFIED TELE-HAPTIC SCHEME

The aim of this paper is to investigate the dynamic properties of a simplified pneumatic teleoperation scheme, where the slave actuator consists of less components than necessary in conventional tele-haptic master slave systems. The main idea is that we propose to use a passive physical pneumatic connection between master and slave cylinder, see Fig. 1.

The pneumatic connection establishes a passive physical relationship between the force experienced by master and slave actuators without the use of force sensors. Conventional haptic teleoperation systems require that both master and slave have force and velocity feedback loops. Such a configuration allows huge flexibility. For example scaling of master and slave motion or cancelling master and slave dynamics to achieve a high level of transparency is possible [3]. In our case, the physical connection does not allow this. Since master and slave cylinders have the same proportions, the force scale is 1:1. The main benefit of the scheme is that force feedback is possible, though limited by friction. The stiffness between the displacements of both cylinders is naturally low due to the high elasticity of the pneumatic coupling. But a set of two control valves situated near the master cylinder is used to increase the transmission stiffness by control, which is why a position transducer is needed on the slave cylinder.

The main features of the proposed scheme are:
- Single control and driver unit. No dedicated drivers and sensors for slave actuator needed.
- Direct fluidic connection of master and slave cylinders.
- Independent pressure control for each of the two lines connecting the cylinders
- Low friction cylinders

The same pressure differential is acting on the master and the slave cylinder. Therefore, force experienced by the operator is directly related to the force acting on the end-effector. Using low friction pneumatic cylinders, it is possible to keep the friction effect below a significant level [12]. A friction force of only 1-2% of the applied load characterizes a low friction cylinder from the company Airpot. The friction caused by air flow in the pneumatic lines influences the force experienced by the operator during motion of the cylinders like an increased damping.

3 MATHEMATICAL MODEL

In this section we derive a nonlinear mathematical model of the teleoperation scheme. Because of the pneumatic connection of master and slave cylinders, the modeling is more complex than for typical pneumatic servo systems. Because of the significant length of the pneumatic lines their modeling cannot be neglected. The equations are grouped into four parts; equations of motion for both master and slave actuators, pressure dynamics for chambers in both master and slave actuators, valve dynamics and connecting tube model.
3.1 Equations Of Motion Of Actuator

The operator contributes to the dynamics of the master device. We first model the hand dynamics with a simple mass-spring-damper system that is rigidly connected to one side of the master actuator. Figure 2 shows a schematic figure of the hand dynamics modeling and the master cylinder, see also [13].

\[
\begin{align*}
    m_h \ddot{x}_h + B_h \dot{x}_h + k_h x_h - B_s \dot{x}_m - k_s x_m &= F_h
\end{align*}
\]

Where \(m_h\) is the modeled mass of the hand, \(x_h\) is the hand position, \(B_h\) is the damping constant of the hand, \(k_h\) is the spring constant of the hand, \(x_m\) is the master cylinder position, and \(F_h\) is the hand force. Replacing the hand reaction force \(F_h\) using the equation of motion of the master cylinder gives the combined motion of equation of hand and master cylinder,

\[
\begin{align*}
    m_p \ddot{x}_m + (\beta + B_h) \dot{x}_m + k_h x_m - B_s \dot{x}_m - k_s x_m &= (P_{m1} - P_{m2}) A_p
\end{align*}
\]

Where \(m_p\) is the mass of the piston, \(\beta\) is the viscous friction of the master cylinder, \(P_{m1}\) and \(P_{m2}\) are the pressures across chamber 1 and chamber 2 of the master cylinder respectively, and \(A_p\) is the master cylinder’s piston area.

Since master and slave cylinders are identical we assume same values for \(\beta\), \(m_p\) and \(A_p\). The equation of motion for the slave cylinder is,

\[
\begin{align*}
    m_s \ddot{x}_s + \beta \dot{x}_s = (P_{s1} - P_{s2}) A_p
\end{align*}
\]

3.2 Pressure Dynamics Across Cylinder Chambers

Richer and Hurmuzlu [14] provide a detailed mathematical model of dual action pneumatic actuators; their modeling equations will be used for our pressure dynamics equations. Three assumptions are made to model the pressures in the chambers: The gas is ideal, the pressures and the temperatures are homogenous, and kinetic and potential energy are negligible. These assumptions can be safely applied to our model as well. Figure 3 shows schematic figure of the cylinder chambers.

From the ideal gas law and assuming isothermal operation, the rate of change in pressure in chamber 1 of each cylinder is,

\[
\dot{P}_i = \frac{R T}{V_{i1}} \dot{m}_i - \frac{P_i}{V_{i1}} \dot{V}_{i1}
\]

Where subscript \(i = m\) or \(s\) refers to either master or slave respectively, \(P_i\) is the pressure in chamber 1, \(R\) is the ideal gas constant, \(T\) is the temperature in chamber 1, \(V_{i1}\) is the volume of chamber 1 and \(\dot{m}_i\) is the mass flow rate in chamber 1. It is safe to assume isothermal operation in case we don’t expect fast movements in the teleoperation application. The volume of chamber 1 is written in terms of cylinder position as,

\[
V_{i1} = V_{d1} + x_i A_p
\]

Where \(V_{d1}\) is the dead volume for the master or slave actuator. After substituting Eq. (5), Eq. (4) becomes,

\[
\dot{P}_i = \frac{R T}{V_{d1} + x_i A_p} \dot{m}_i - \frac{P_i}{V_{d1} + x_i A_p} \dot{x}_i A_p
\]
Similarly the change in pressure in chamber 2 is expressed as,

\[
P_{1/2} = \frac{RT}{V_{1/2}} \dot{m}_{1/2} - \frac{P_{1/2}}{V_{1/2}} \dot{V}_{1/2} = \frac{RT}{V_{a/2} + (l_{off} - x)A_p} \dot{m}_{1/2} + \frac{P_{1/2}}{V_{a/2} + (l_{off} - x)A_p} \dot{x}, A_p
\]  

Where \( l_{off} \) is the total cylinder length. These equations are applied to both master and slave cylinders by substituting subscript \( i \) with either \( m \) or \( s \).

### 3.3 Valve Dynamics

The mass influx \( \dot{m}_{wa} \) from valve \( n=1,2 \) is related to the control input \( u \). According to ISO 6358 technical nozzles and orifices can be modeled as follows [15]:

\[
m_{wa} = \begin{cases} 
  x_v c P_0 \frac{T_0}{T_{wa}} \left( \frac{P_{wa} - b}{P_{wa} - P_{b}} \right) \frac{1}{1-b} & \text{for } P_{wa} > b \\
  x_v c P_0 \frac{T_0}{T_{wa}} & \text{for } P_{wa} \leq b
\end{cases}
\]

Where \( x_v \) is the spool position of the valve, \( c \) is the sonic conductance given by the valve manufacturer, \( P_0 \) is the density of air at reference conditions, \( P_{wa} \) is the upstream pressure in valve 1 or 2, \( T_0 \) is the temperature of air at reference conditions, \( T_{wa} \) is the upstream temperature of air, \( P_{wa} \) is the downstream pressure, while \( b \) is the critical pressure ratio given by the valve manufacturer. Furthermore the motion of the valve spool in terms of the signal input is modeled as a second order system. Equation (9) shows the second order model of the valves,

\[
\ddot{x}_{wa} + 2 \zeta \omega_x \dot{x}_{wa} + \omega_x^2 x_{wa} = K_x \omega_x^2 u_n
\]

Where \( \zeta \) is the damping ratio of the valves, \( \omega_x \) is the undamped natural frequency, \( K_x \) is the electromechanical valve gain, and \( u_n \) is the input voltage to the valves, where \( n=1 \) or 2 is input to valve 1 or 2 respectively.

### 3.4 Tube Modeling

Figure 4 shows the model sketch for the pneumatic lines. The pneumatic tele-operation system requires long pneumatic lines cross-coupled between the chambers of master and slave cylinders in the range of 5-15 meters. The flow friction and inductance of the mass flow in the long tubes from the valves to the slave cylinder are considerable [14]. Since the valves are connected closely to the master cylinder, it is safe to ignore the dynamics of those short tubes.

![Pneumatic Line Model](image)

**FIGURE 4. MASS FLOW RATES AND PRESSURE CHANGE IN PNEUMATIC LINES 1 AND 2.**

From the conservation of mass and continuity equations we get the following equation for line \( n=1,2 \)

\[
\dot{m}_{wa} = \dot{m}_{wa} + \dot{m}_{wa}
\]

(10)

Tube dynamics are modeled by a simplified equation, assuming the temperature change in and out of the lines is minimal. This assumption is valid if the highest frequency of interest \( f \) in rad/s follows this condition [15],

\[
f \leq \frac{4v}{A}
\]

(11)

Where \( v \) is the kinematic viscosity, and \( A \) is the cross sectional area of the pneumatic lines. For a line diameter of 4 mm, the assumption is valid for motion frequencies of up to 0.75 Hz. If we satisfy this condition we can model each pneumatic line with a simple resistance, capacitance and inductance characteristic [16]. Equations (12)-(15) describe line 1 dynamics, while line 2 equations are derived similarly. The pressure change in line 1, \( P_{i1} \), is characterized by the capacitance \( AL/(RT) \).
\[ \hat{\dot{P}}_{L1} = \frac{RT}{AL}(\dot{m}_{L1} - \dot{m}_2) \]  

(12)

Where \( L \) is the pneumatic line length. The change in mass flow rate out of the line is described by the inertance \( 1/L \) and the friction characteristic as \( \Delta P_{\text{friction}} \),

\[ \frac{\text{d}m_{L1}}{\text{d}t} = \frac{A}{L}(P_{\text{in1}} - P_{L1}) - \frac{A}{L} \Delta P_{\text{friction}} \]  

(13)

Pressure loss for laminar flow is expressed in terms of air dynamic viscosity \( \mu \), mean air velocity through the pneumatic line \( \bar{w} \), tube length \( L \), and tube diameter \( D \) as,

\[ \Delta P_{\text{friction}} = \frac{32\mu w DL}{D^2} = \frac{32\mu w L}{\rho AD^2} \]  

(14)

The third equation is,

\[ P_{L1} = P_{c2} \]  

(15)

\section*{4 LINEARIZING NON-LINEAR EQUATIONS}

A linearized model of the teleoperation scheme is useful for quantifying dynamic properties such as the bandwidth or stability margin and how those are affected by design parameters such as for example the cylinder or tube diameters and lengths.

\subsection*{4.1 Linearized Pressure Dynamic Equations}

We linearize the pressure dynamics of the cylinder chambers around chosen equilibrium points. The character "\( \sim \)" above a state space variable means that we consider the difference between the state space variable and its value at equilibrium. For example in Eq. (6) we have the parameter,

\[ \hat{\dot{P}}_1 = \dot{P}_1 - \dot{P}_{1\text{ Equi.}} \]  

(16)

This convention is applied throughout the rest of the paper. The equilibrium point is defined when the piston is in the middle position of the cylinder and not moving. This gives us the simplicity of setting the mass flow, piston velocity and change in pressures to zero. The equilibrium pressure in the cylinder chambers is chosen in such a way between supply and atmospheric pressure that the system responds symmetrically for positive and negative valve openings by equating Eq. (8) for positive and negative mass flow of the valves,

\[ x_{\text{eq}}cP_0 P_{\text{Equi.}} \left( \frac{T_r - b}{T_r - T_m} \right) \leq 1 \]  

(17)

While \( P_s \) is the pressure supply to the valves and \( P_r \) is the atmospheric pressure. The linearized equation corresponding to Eq. (6) becomes,

\[ \frac{\partial \dot{P}_1}{\partial x_j} \right|_{\text{Equi.}} = 0 \]

\[ \frac{\partial \dot{P}_1}{\partial \dot{m}_1} \right|_{\text{Equi.}} = \frac{RT}{V_d + 0.5\beta_c A_p} \]

\[ \frac{\partial \dot{P}_1}{\partial \dot{x}_j} \right|_{\text{Equi.}} = \frac{P_{1\text{ Equi.}} A_p}{V_d + x_j A_p} \]  

(19)

The equations describing linearized parameters for chamber 2 are the same as Eq. (17) - (19) if we exchange subscript 1 to 2. For the coefficients we obtain

\[ \frac{\partial \dot{P}_2}{\partial \dot{P}_2} \right|_{\text{Equi.}} = 0 \]

\[ \frac{\partial \dot{P}_2}{\partial \dot{m}_2} \right|_{\text{Equi.}} = \frac{RT}{V_d + 0.5\beta_c A_p} \]

\[ \frac{\partial \dot{P}_2}{\partial \dot{x}_j} \right|_{\text{Equi.}} = \frac{P_{2\text{ Equi.}} A_p}{V_d + x_j A_p} \]  

(20)
4.2 Linearized Valve Dynamic Equations

From Eq.(17) we can use $P_o$ as operating point for $P_m$ and $P_{in(g)}$ for $P_{de}$. The equilibrium point for the spool position is zero. The linearized version of Eq. (8) becomes,

$$
\dot{m}_{vo} = \frac{\partial \dot{m}_{vo}}{\partial x_{ve}} \dot{x}_{ve} + \frac{\partial \dot{m}_{vo}}{\partial P_{vo}} \dot{P}_{vo} + \frac{\partial \dot{m}_{vo}}{\partial P_{de}} \dot{P}_{de} \tag{21}
$$

with,

$$
\frac{\partial \dot{m}_{vo}}{\partial x_{ve}} = cP_oP_s \left(1 - \frac{P}{1 - b}\right)^2 \tag{22}
$$

4.3 Linearized Tube Modeling Equations

The linearized version of the tube modeling line 1 and 2 of Eq. (13) is,

$$
\dot{m}_{L1} = \frac{\partial \dot{m}_{L1}}{\partial P_{m1}} \dot{P}_{m1} + \frac{\partial \dot{m}_{L1}}{\partial P_{L1}} \dot{P}_{L1} + \frac{\partial \dot{m}_{L1}}{\partial m_{L1}} \dot{m}_{L1} \tag{23}
$$

with,

$$
\frac{\partial \dot{m}_{L1}}{\partial P_{m1}} = \frac{\partial \dot{m}_{L1}}{\partial P_{m2}} = \frac{\partial \dot{m}_{L1}}{\partial P_{L1}} = \frac{\partial \dot{m}_{L2}}{\partial P_{L2}} = \frac{A}{L} \tag{24}
$$

5 CONTROL DESIGN

The control is necessary to improve the relatively low open loop transmission stiffness, which can be determined from the non-linear model to be 0.29 N/mm. While position feedback of master and slave cylinder would be sufficient for the teleoperation task, a pressure feedback cascade is used to improve the performance and robustness. Figure (5) shows the control scheme.

The master slave position difference is multiplied by a stiffness parameter and then divided by the piston area so that the control signal has the physical interpretation of a desired pressure difference. This pressure difference is divided by 2 and applied to the pressure feedback loops of both pneumatic lines with different signs. A separate valve is used for each line; so the bias pressure can be arbitrarily chosen between supply and atmospheric pressure.

![Figure 5 Experimental Setup Cascaaded Stiffness Control](image)

FIGURE 5 EXPERIMENTAL SETUP CASCADEd STIFFNESS CONTROL

Figure (6) shows the linearized control scheme where we take advantage of the symmetry of the pneumatic system. That the input to valve 2 is simply equal to negative the input to valve 1. $G(S)$ is the linearized open loop transfer function where $u_i$ is the input and the measured pressure $P_{ml}$ and the positions of master $x_m$ and slave $x_s$ are outputs. $H(S)$ is the linearized pressure control cascade where the pressure difference across the pneumatic lines $P_{ref}$, calculated using the position error, is the input and the position error $(x_m - x_s)$ is the output.

![Figure 6 Linearized Control Scheme](image)

FIGURE 6. LINEARIZED CONTROL SCHEME
The objective of the control is to achieve high transmission stiffness while retaining a stable robust response. Therefore we are primarily interested in a high stiffness gain $K_s$, while maintaining a good phase margin as a measure of robustness. Figure (7) shows the phase margin plotted over different combinations of $K_p$ and $K_s$.

![Figure 7 Phase Margin Plotted Over Different Combinations of $K_p$ and $K_s$](image)

The lines for constant phase margin seem to have the shape of a hyperbola which indicates that the stability margin is a function of the product of $K_p$ and $K_s$ and that it is not important whether the pressure cascade or the error feedback cascade gets the higher feedback value. In the experimental results we have chosen a pair of control parameters $K_p = 2.9 \text{ V/bar}$ and $K_s = 2.4 \text{ N/mm}$ that provide approximately a phase margin of 80°. With these values for the loop gain we get a transmission stiffness of 2.4 N/mm, which is an increase by more than a factor of 10 compared to the open loop transmission stiffness of 0.23 N/mm.

6 EXPERIMENTAL AND SIMULATED RESULTS

Figure (8) shows the experimental setup of the proposed simplified tele-haptic scheme shown in Fig. (1). The low friction cylinders have a piston surface of 420mm² and a stroke of 275mm. Two resistive position transducers are connected to cylinders.

![Figure 8 Experimental Setup](image)

The servo-proportional valves have a sonic conductance at full opening of 0.45 l/(bar.s) and a critical pressure ratio of 0.21. The 5m plastic tubing between valves and slave cylinder is coiled under the setup. Each line is connected to a pressure transducer. The control unit uses Ethercat I/O terminals and the control is realized with the open source Etherlab software on a real time linux system based on the open source Rtai project. The cycle time is 1 kHz.

![Figure 9 Measured Experimental Position of Master and Slave Cylinders' Piston Rod](image)

Experimental values for master and slave cylinder positions have been extracted for control gains $K_p = 2.9 \text{ V/bar}$ and $K_s = 1.2 \text{ N/mm}$. Figure (9) shows the experimental master and slave position, where the operator performed cyclic motion covering a distance of approximately 25mm at around 0.4 Hz.

It shows that the slave cylinder follows the master cylinder with minimal phase lag and has a steady state error in the mm range.
In the simulation model we did not see such a steady state error. It was shown that the steady state error is due to valve offset and external leakage of the cylinders. Figure (10) compares the position error between the master and the slave cylinders between the experimental results and theoretical results. The theoretical results were obtained from a nonlinear Matlab model that was simulated with same load cycle as the experimental setup.

![Position Error Comparison](image)

**FIGURE 10. EXPERIMENTAL AND THEORETICAL ERROR BETWEEN MASTER AND SLAVE POSITIONS**

From Fig. (10) we observe an average experimental absolute error of 2 mm while in motion and steady state error of 0.6 mm. The position error in the simulation is smaller by factor three during motion and is zero in the resting position. It is easy to show in the simulation that the cylinder friction has a large influence on the precision. We determined the viscous friction parameter through experiment to be 0.33*E-3 N.s/mm but we did not implement a coulomb friction characteristic which could be the cause of this difference between experimental and simulation results during motion. Also the laminar flow friction model in the tube relies on some uncertain parameters. In the resting position, the cylinder leakages and incorrect valve offset are the cause of steady state deviation. The offset is not a serious problem because absolute positionning is not required.

From the control design we expect that the transmission stiffness, \( K_s = 2.4 \) N/mm. Since we did not have force sensors on the experimental setup, we can use the simulation to prove this outcome. The slave cylinder is held in the middle position while applying a direct force ramp on the master cylinder.

After reaching steady state we calculate the stiffness by dividing the steady state force on the master cylinder by the position error between the master and slave cylinders. Figure (11) shows the master cylinder position, the ramp force applied on the master cylinder reaching 10 N at steady state, and the slave cylinder position.

![Stiffness Calculation](image)

**FIGURE 11 PLOT SHOWING THE MASTER POSITION WHILE FORCE APPLIED TO THE MASTER CYLINDER AND SLAVE CYLINDER HELD INPLACE**

From Fig. (11) the effective transmission stiffness is calculated to be 2.389 N/mm

### 7 CONCLUSION

This paper presents the theoretical model of a pneumatic tele-operation system with both, signal based and passive physical interconnection. Control gains for a cascaded stiffness control are derived based on a desired transmission stiffness and a phase margin criterion. The control parameters are implemented in the experimental setup and work well despite uncertainty in important parameters such as piston and pipe flow friction. The analysis shows clearly the trade-off between stability and transparency. The proposed control scheme improves effective transmission stiffness by more than a factor 10 compared to the purely passive stiffness of the pneumatic connection.

This paper focuses on establishing a verified model for an unconventional teleoperation system and presents a preliminary analysis. In future work this model can be used to help optimizing the mechanical design of this setup. The most influential parameters should be optimized to get the best tradeoff between stability and transparency.
ACKNOWLEDGMENTS

The authors express their thanks to the University Research Board of the American University of Beirut for funding this research.

REFERENCES