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Vibration Suppression Design for Virtual Compliance Control in Bilateral Teleoperation

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Academic Year 2017/2018

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Symbol	Parameter	Value	Unit
	Master-Slave system		
J_m	Master Inertia	$5 \cdot 10^{-4}$	$\mathrm{kg} \cdot \mathrm{m}^2$
J_s	Slave Inertia	$5 \cdot 10^{-4}$	$\mathrm{kg} \cdot \mathrm{m}^2$
	Desired cut-off frequencies		
g_1	1st cut-off frequency	$5 \cdot 10^{1}$	rad/s
g_2	2nd cut-off frequency	$5 \cdot 10^{2}$	rad/s

Table 1: Parameters adopted in simulations.

Behaviour	K_v	B_v	J_v
Virtual compliance	$20.0\ \frac{\mathrm{N\,m}}{\mathrm{rad}}$	$4.4 \cdot 10^{-1}\ \frac{\mathrm{N\,m}}{\mathrm{rad/s}}$	$3 \cdot 10^{-4}$ kg m^2
Rigid coupling	$10^2\ \frac{\mathrm{N\,m}}{\mathrm{rad}}$	$1.5 \cdot 10^{-1}\ \frac{\mathrm{N\,m}}{\mathrm{rad/s}}$	0 kg m^2

Table 2: Sets of chosen virtual parameters.

1 Simulations

1.1 Chosen parameters

In regards of the simulation scenarios we are deliberately neglecting the critical aspects of the communication between master and slave.

Therefore all the following simulations has been run assuming ideal conditions as an instantaneous and loss-less signal transfer between master and slave subsystems.

The table n.**??** describes the parameters chosen such as inertiae and cut-off frequencies, consequently the table n.**??** describes the virtual coefficients computed as explained in section **??**.

1.2 Disturbance rejection performances

We consider at first the rigid coupling case, in which, as being said, almost full transparency is achieved between master and slave. The vibrations transmitted by the environment on the slave-side will be felt almost with the same intensity on the master-side whatever would be the vibration frequency.

For this reason, in order to reach better task execution performances we want to reduce the impact of environment vibrations at minimum.

The simulations aims to compare two opposite behaviours: **rigid coupling** and **induced virtual compliance**, achieved through the choice of the desired cut-off frequencies.

Fig.**??** describes the frequency response of the function (**??**) with several virtual parameters sets, defined starting from different spring stiffness values K_v varying from $2.5\ \mathrm{N} \cdot \mathrm{m/rad}$ to $20\ \mathrm{N} \cdot \mathrm{m/rad}$, and the rigid coupling control set.

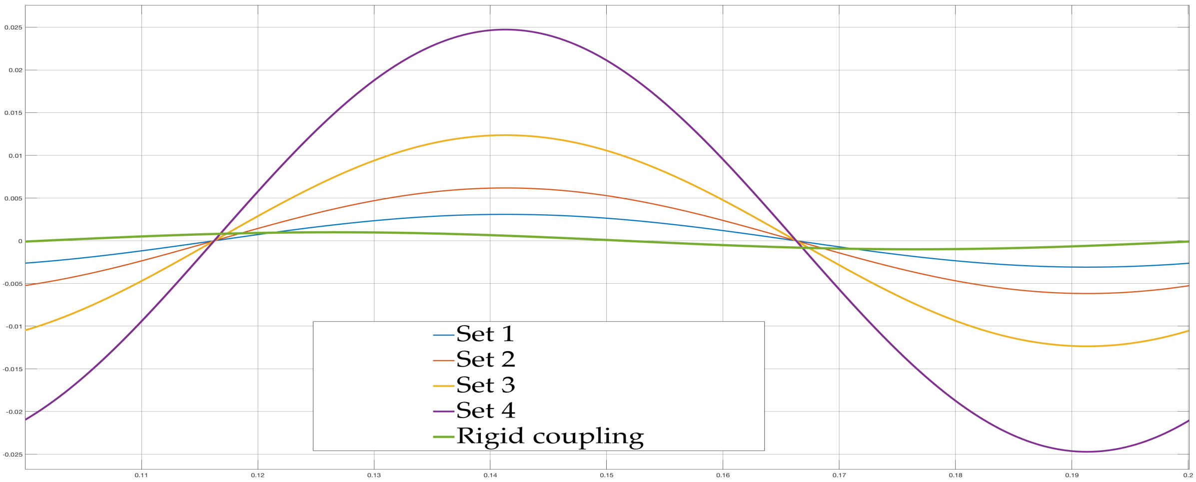
At low frequencies the signals are preserved in both the rigid coupling and the virtual compliance controls. In particular, the magnitude of the response increases with inverse of the virtual spring stiffness, describing the **compliance** of the system. At high frequencies the disturbance rejection is exerted more effectively by the virtual compliance control: the position response to high frequency external torque is reduced due to the low compliance.

A comparison of the vibration suppression applied on three different input frequencies is interesting, (figs.**??**) [1]:

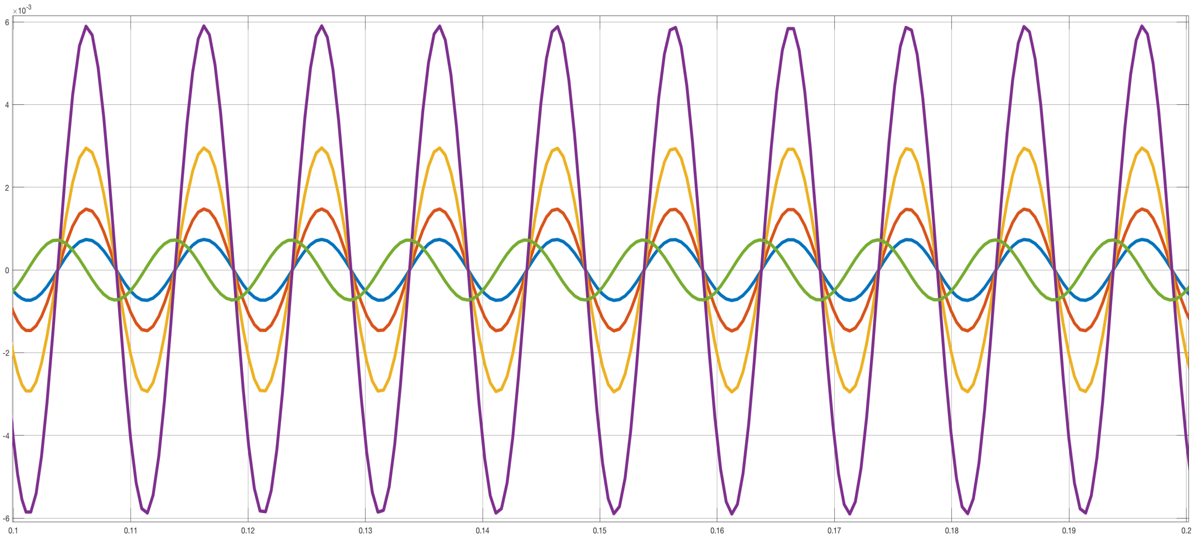
- 10 Hz input : in fig.**??** is shown how the inputs at lower frequencies are preserved by the **control set 4** (defined by $K_v = 2.5\ \mathrm{N} \cdot \mathrm{m/s}$ - purple line). The response is less large as stiffness increases. The phase is almost the same for virtual compliance and rigid coupling controls;
- 10^2 Hz input : as before, the response decreases as the virtual spring stiffness increase. The difference here is the almost the same response of the **control set 1** ($K_v = 20\ \mathrm{N} \cdot \mathrm{m/s}$ - blue line) and the rigid coupling ($K_v = 10^2\ \mathrm{N} \cdot \mathrm{m/s}$ - green line), despite the second one is defined by a larger value of stiffness. The phase relative to the **control sets** is delayed respect to the rigid coupling control (fig.**??**);
- 10^3 Hz input : at frequencies higher than the cut-off ones, the inputs are dumped more effectively by the **control sets** than the **rigid coupling control**, for which we have larger response (fig.**??**).

[1] For the sake of simplicity the different control set are called:

- Control set 1 : defined by $K_v = 20.0\ \frac{\mathrm{N \cdot m}}{\mathrm{rad}}$;
- Control set 2 : defined by $K_v = 10.0\ \frac{\mathrm{N \cdot m}}{\mathrm{rad}}$;
- Control set 3 : defined by $K_v = 5.0\ \frac{\mathrm{N \cdot m}}{\mathrm{rad}}$;
- Control set 4 : defined by $K_v = 2.5\ \frac{\mathrm{N \cdot m}}{\mathrm{rad}}$.



(a) 10 Hz



(b) 100 Hz

1.3 Task execution analysis

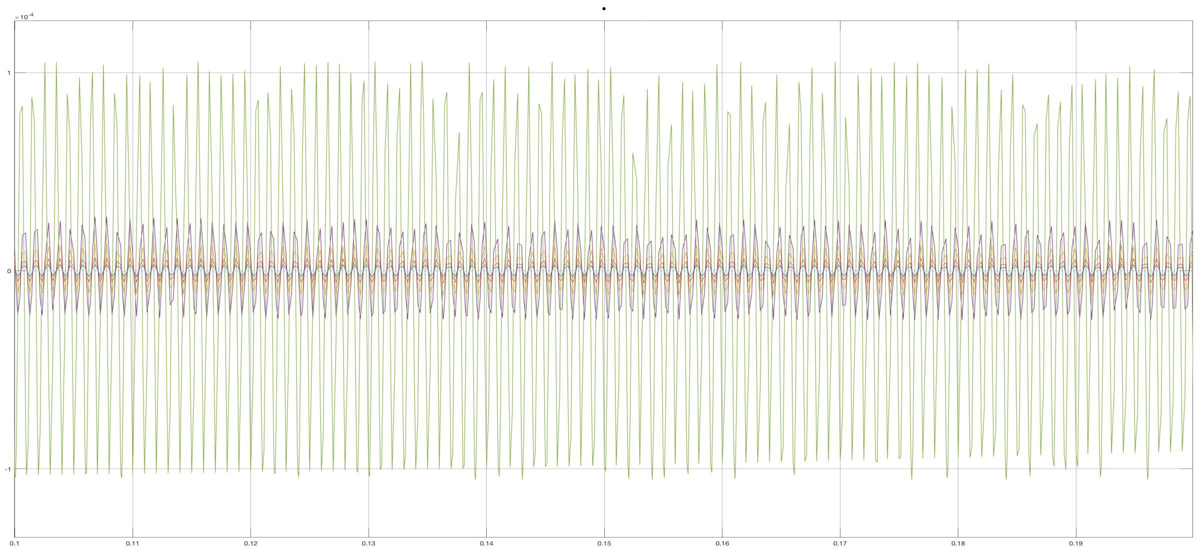
Simulation setup

The operator is modeled as spring-damper system ($K = 200.0$ N/m and $B = 4.0$ N·s/m). The master-slave system has arms of length equal to 0.1 m.

The environment with which the slave manipulator comes into contact is modeled as:

- free motion: the environment has no stiffness but just a small damping $B_{env} = 0.01$ N $\cdot$ s/m;

- contact motion: the environment has spring stiffness $K_{env} = 100.0$ N/m after a



(c) 1000 Hz

Figure 1: Slave position response to a sinusoidal input with three different frequencies.

chosen value of displacement, modeling a not totally rigid body.

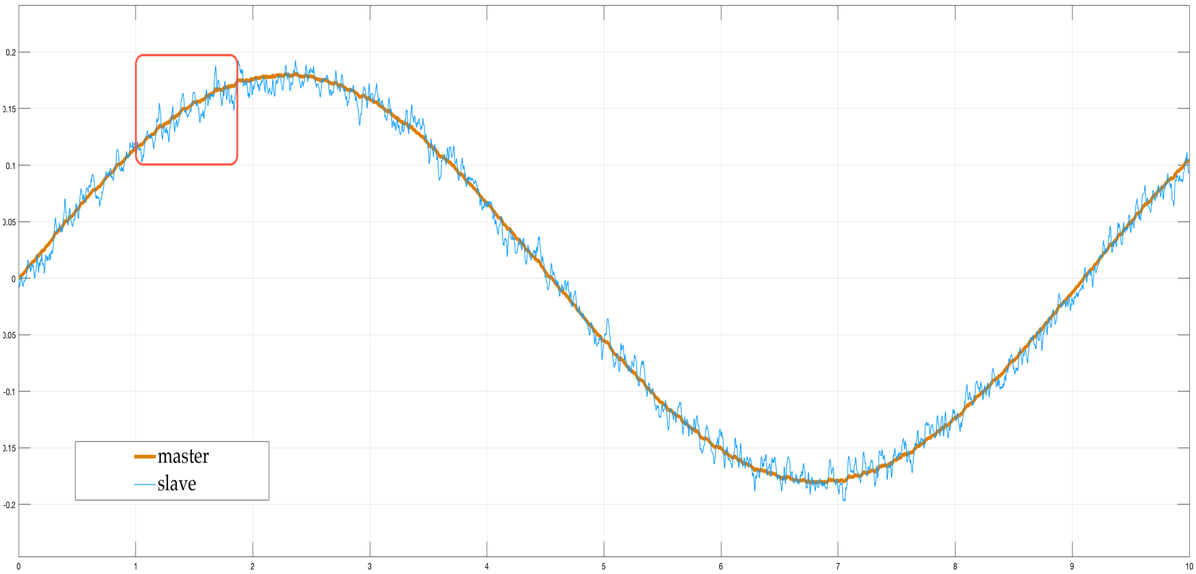
The simulation are performed with the **control set 4** ($K_v = 20.0\frac{\mathrm{N\,m}}{\mathrm{rad}}$, $B_v = 4.4 \cdot 10^{-1}\ \frac{\mathrm{N\,m}}{\mathrm{rad/s}}$, $J_v = 3 \cdot 10^{-4}$ kg m^2) and are compared with the **rigid coupling control** ($K_v = 10^2\ \frac{\mathrm{N\,m}}{\mathrm{rad}}$, $B_v = 1.5 \cdot 10^{-1}\ \frac{\mathrm{N\,m}}{\mathrm{rad/s}}$, $J_v = 0$ kg m^2).

1.3.1 Free motion with high frequency input

At first, we present an execution in free motion. The slave manages to mirror the master which is moved according to a 0.11 Hz sinusoidal trajectory. Applying both *rigid coupling control* (fig.**??**) and *virtual compliance control* (fig.**??**) there is almost no task error.

The noise of the system is modeled as a mixture of white noise and sinusoidal oscillations both at frequency of 50 Hz on the slave side.

A comparison between the fig.**??** and the fig.**??** shows how is the slave response under the two different type of controllers: the response is slower using the virtual compliance control respect to the rigid coupling control. By consequence, the operator "feels" less disturbances.



(a) Positions of the master-slave system in free motion - virtual compliance control.

1.3.2 Free motion with low frequency input

Comparatively, two other simulations have been performed under the same conditions of the previous ones. Here the input frequency is lowered to 20.0 Hz.

Usually, one wants this type of inputs to be detectable, and hence, they should be preserved.

This phenomenon shows up in fig.**??** and fig.**??**. Infact the two profiles confirm that *rigid coupling control* cancel out most of the useful information from the signal, while *virtual compliance control* saves the signal information.

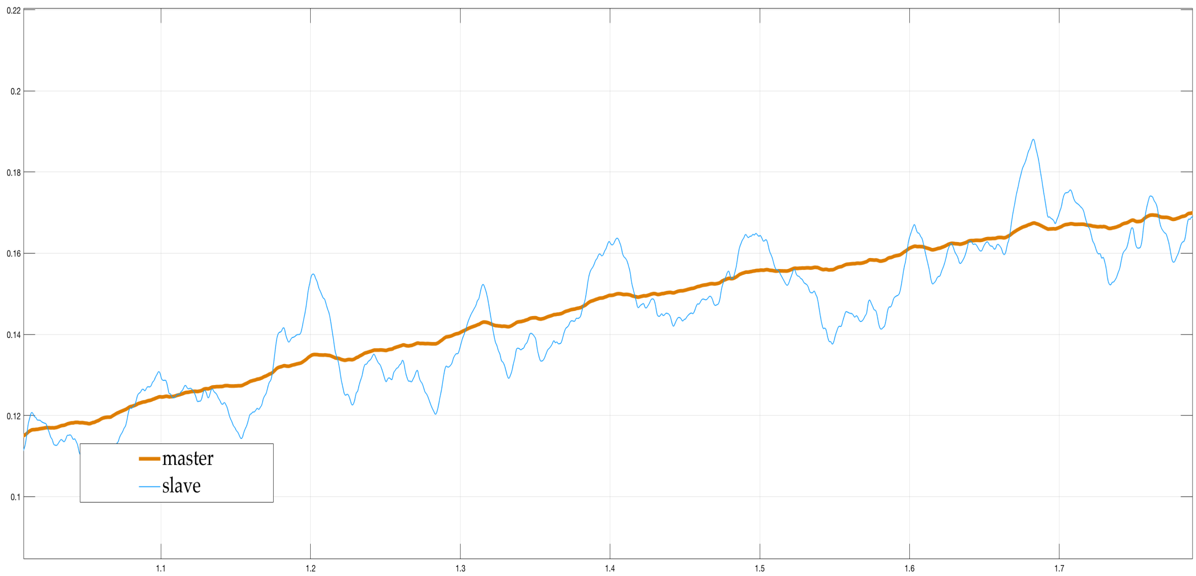
1.3.3 Contact motion

In this section the reference trajectory for master is equal to the one used for the simulations in free motion.

The slave, overcoming a certain angle value, is in contact with the environment. This will not allow a perfect tracking by the slave.

The comparison between *virtual compliance control* and *rigid coupling control* in presence of a contact with the environment can be deduced by the differences in **magnitude** of the arrows drawn in the fig.**??** and fig.**??**.

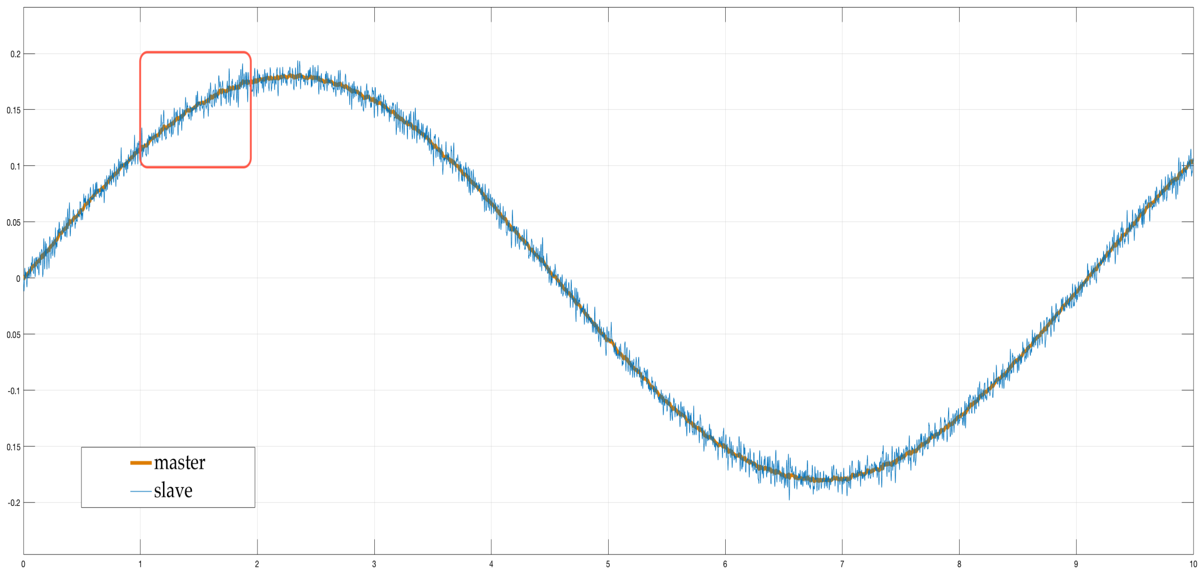
During the contact motion phase, a gap between the master and the slave position emerges. Fig.**??** shows as the *virtual compliance control* leads to a larger position error then the *rigid coupling control* (fig.**??**): with high spring stiffness value the gap is closer.



(b) Detail describing the highlighted area in fig.**??**

Figure 2: **High** frequency disturbances response - virtual compliance control.

From the point of view of the torque exerted, the proposed control (fig.**??**) performs more efficiently than *rigid coupling coupling* (fig.**??**), suppressing the high frequency noise from the environment to the master side.



(a) Positions of the master-slave system in free motion - rigid coupling control.

2 Conclusions

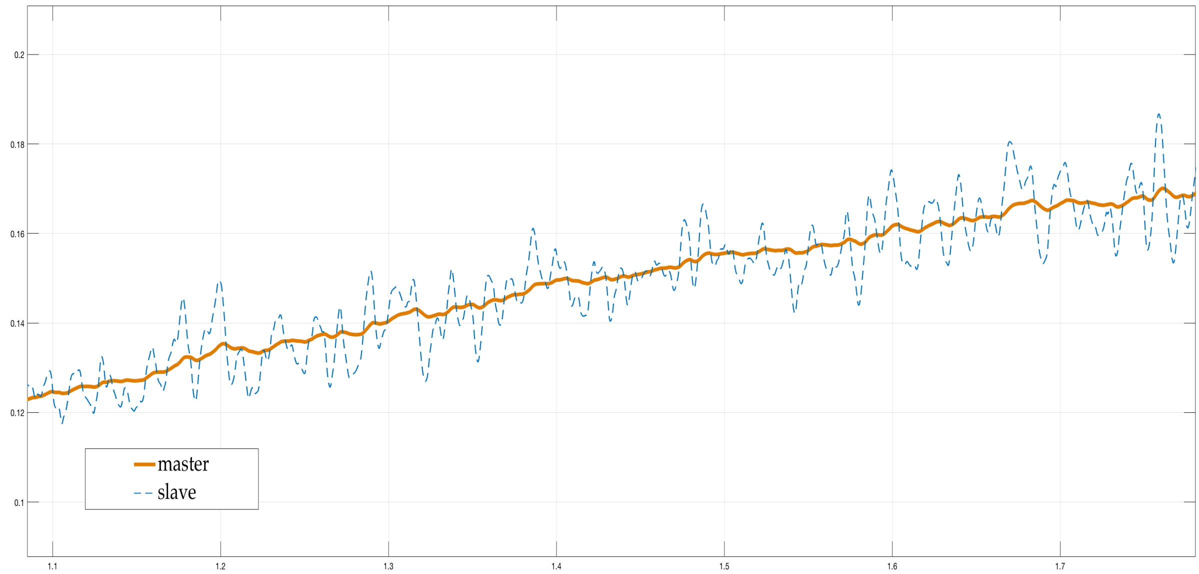
Vibration suppression in the contest of bilateral teleoperation is an open issue.

The proposed solution is based on a virtual spring-damper system with additional inertia. The spring stiffness and the cut-off frequencies are chosen according to the system requirements.

To summarize, when the virtual stiffness has been fixed, the other virtual parameters could be calculated from the equations in order to obtain the desired cut-off frequencies.

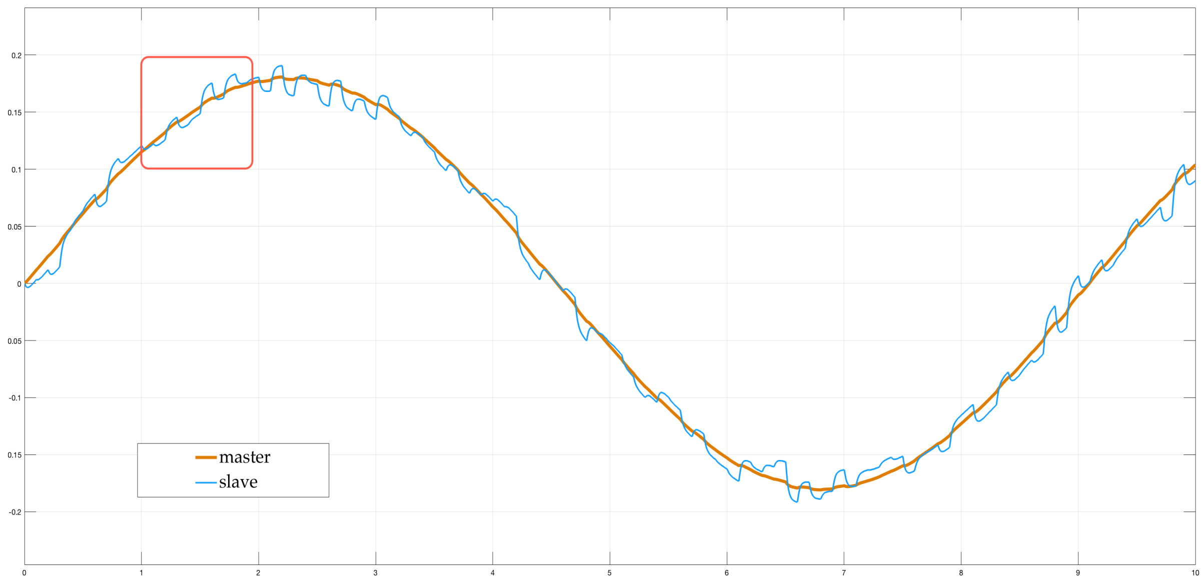
The tracking error in *free motion* is almost the same, using both **virtual compliance control** and **rigid coupling control**. However,the proposed controller shows promising results, since it preserves the useful (*low*) frequency inputs and reject the noisy ones (*high*).

In *contact motion* the usage of the proposed controller leads to a position gap between master and slave and hence can be applied only to tasks that involve the handling of *soft* materials.

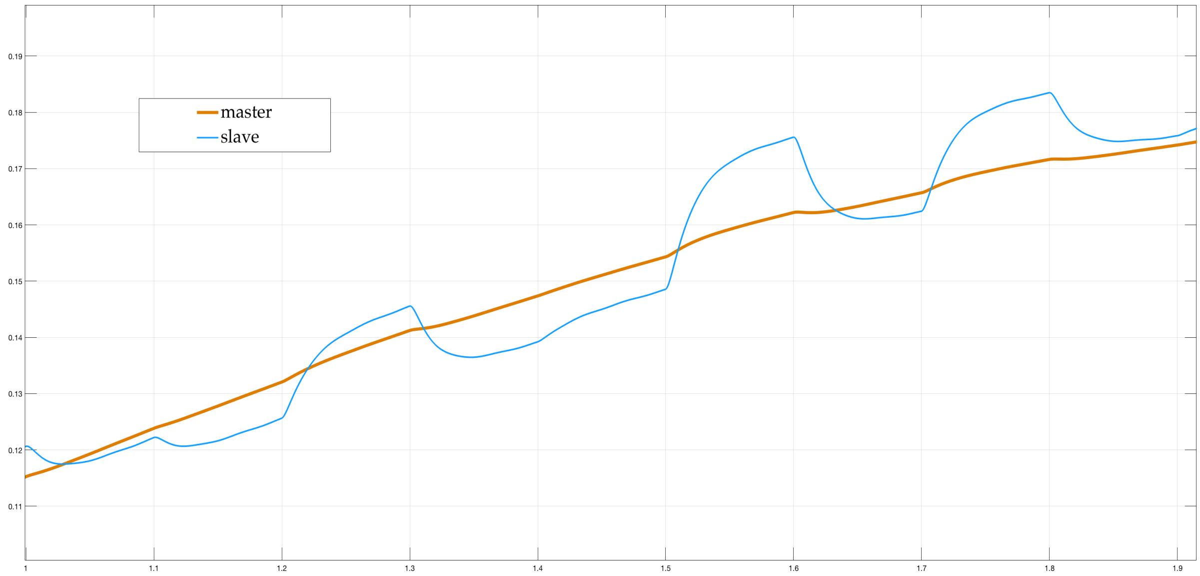


(b) Detail describing the highlighted area in fig.**??**.

Figure 3: **High** frequency disturbances response - rigid coupling control.

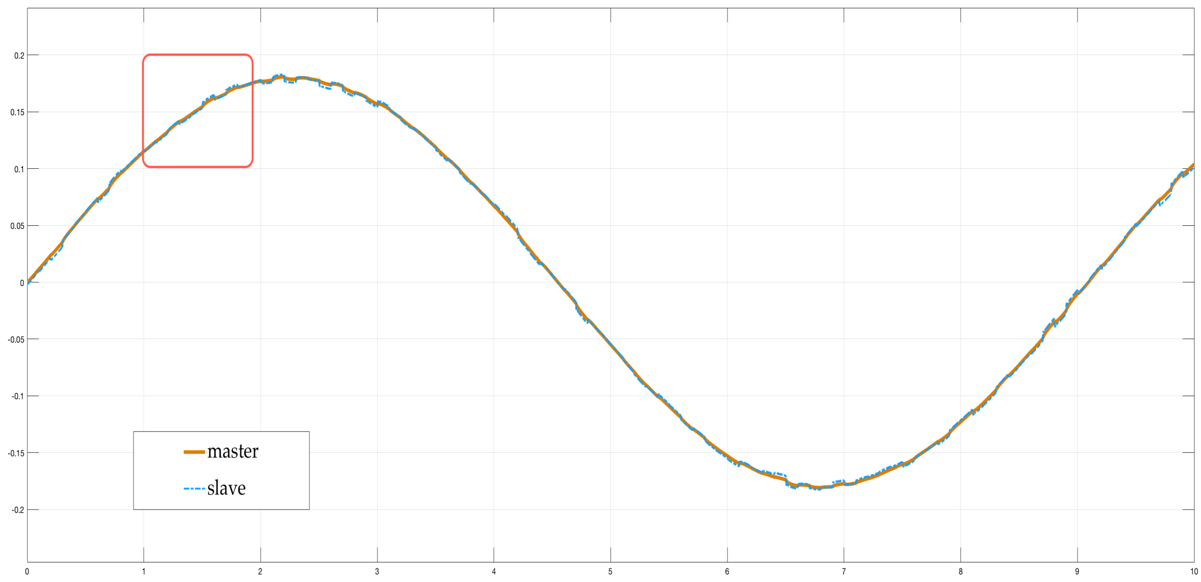


(a) Positions of the master-slave system in free motion - virtual compliance control.

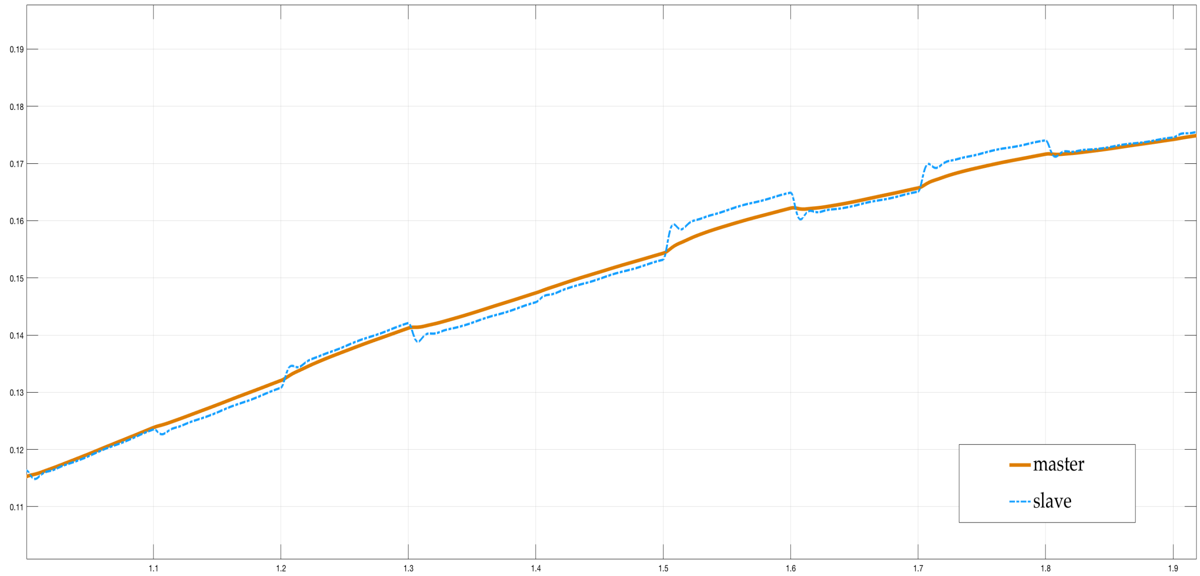


(b) Detail describing the highlighted area in fig.**??**

Figure 4: **Low** frequency disturbances response - virtual compliance control.

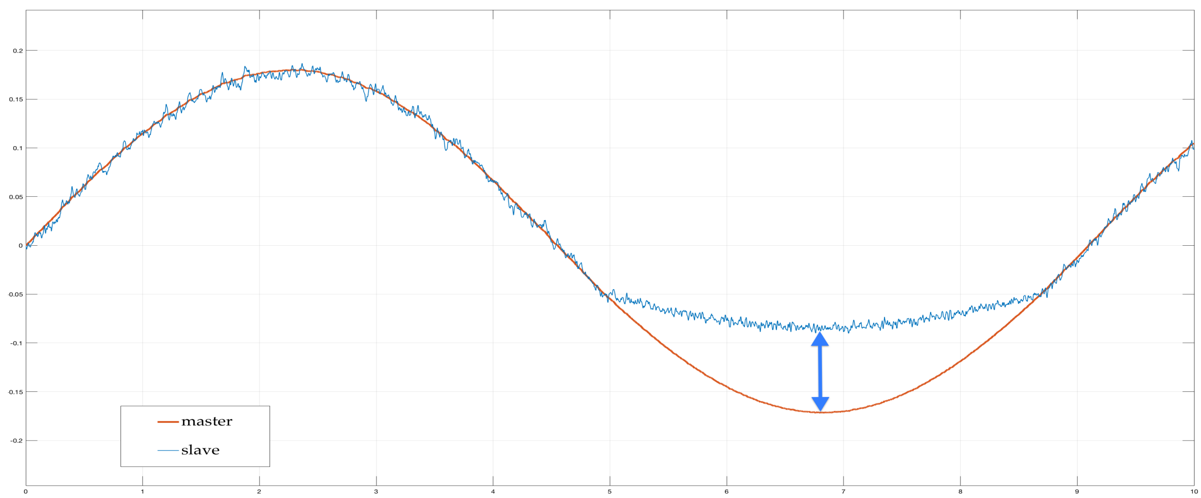


(a) Positions of the master-slave system in free motion - rigid coupling control.

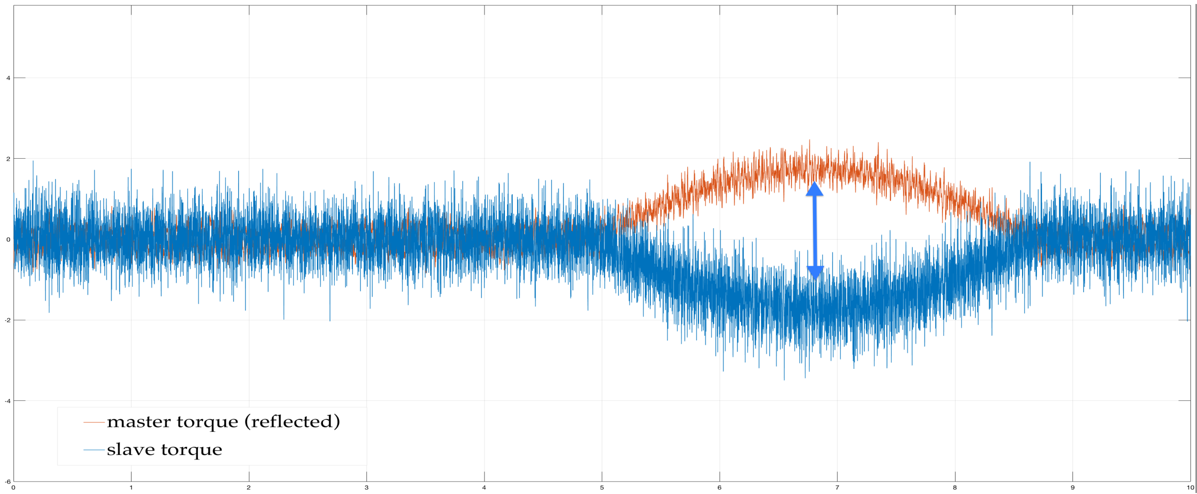


(b) Detail describing the highlighted area in fig.**??**

Figure 5: **Low** frequency disturbances response - rigid coupling control.

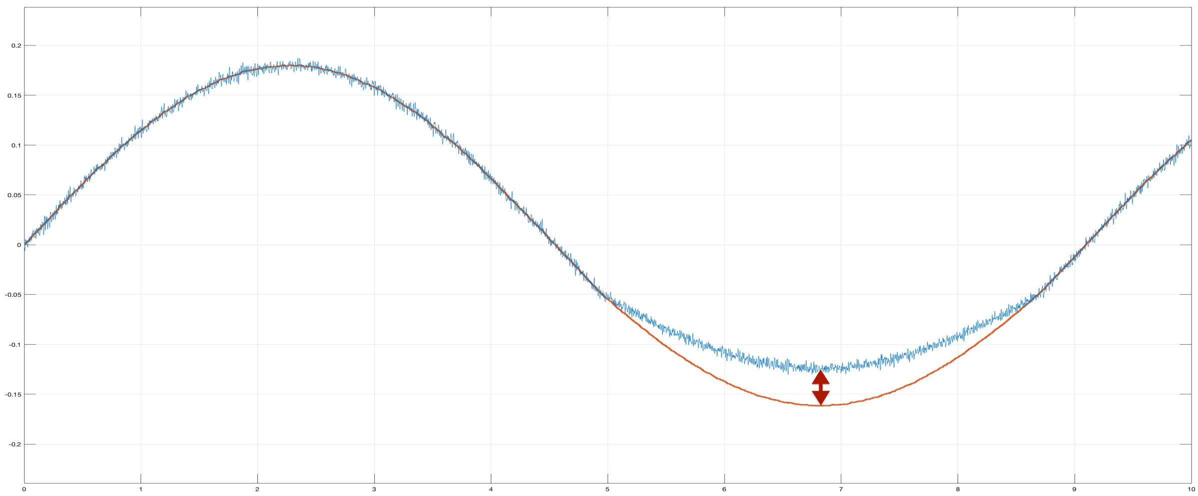


(a) Position assumed during trajectory tracking.

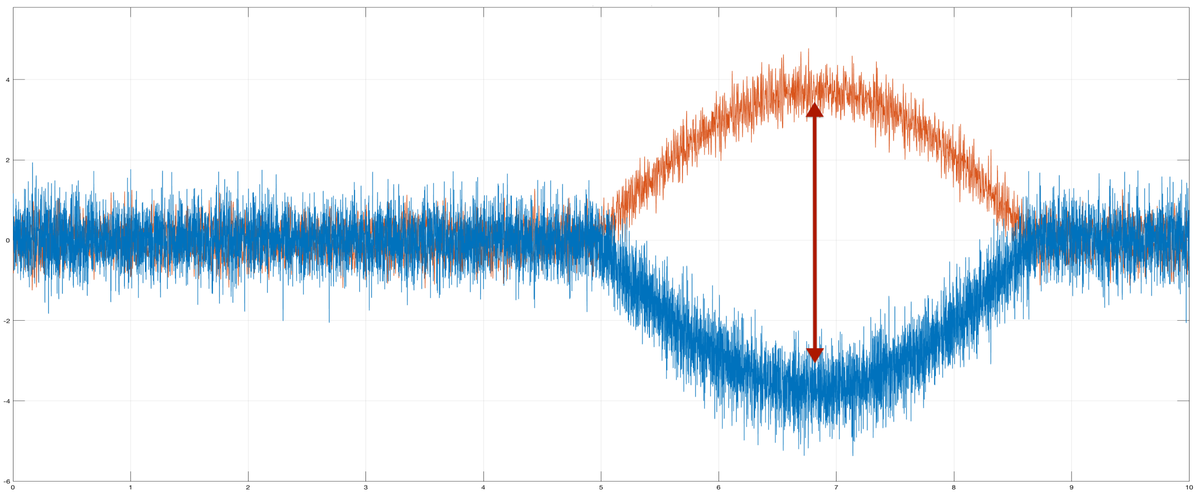


(b) Torques exerted over time.

Figure 6: Contact motion simulation - virtual compliance control.



(a) Angular position assumed during trajectory tracking.



(b) Torques exerted over time

Figure 7: Rigid coupling simulation in *contact* with the environment