Modeling, analysis and control of a variable geometry actuator

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Modeling, Analysis and Control of a Variable Geometry Actuator

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Abstract—A new design of variable geometry force actuator is presented in this paper. Based upon this design, a model is derived which is used for steady-state analysis, as well as controller design in the presence of friction. The controlled actuator model is finally used to evaluate the power consumption under worst case conditions.

I. INTRODUCTION

The main function of a suspension system is to reduce the effect of environmental vibrations, e.g. to limit the transmissibility. This can be achieved with passive elements, e.g., springs and dampers, but also with (semi-) active elements. Herein, semi-active elements have the property that they can only dissipate energy, where active elements can also add energy to the suspension.

Semi-active suspensions consist of dampers with a variable damping constant. This variation can either be achieved mechanically, [12], or using so called "smart fluids". The latter can be roughly divided into two classes: magneto-rheological (MR) and electro-rheological (ER) dampers, [5], [7]. Semi-active suspensions generally have a low energy consumption (typically related to electric actuation of valve spools), while still giving improved performance in comparison to passive suspensions. However, the in theory obtainable performance is profoundly lower than what is obtainable with active suspensions, [4].

The best known active suspension element is probably the air spring, [3]. However, its bandwidth is typically so low that it cannot be used for much other than load leveling and load compensation. As such, it also consumes relatively little energy. Hydraulic actuators on the other hand can reach far higher bandwidths, see for example [1]. Another example of an active hydraulic suspension (in an automotive context) is the *DaimlerChrysler Active Body Control system*, [6], which reaches a bandwidth of up to 5 Hz. However, hydraulic actuators generally have a high power consumption.

An alternative to lower the power consumption is the use of electro-magnetic actuators, [9], [13], as these have the capability to regenerate energy. Another alternative would be the use of a variable geometry (force) actuator, [15], [16].

In this paper a model is presented of the variable geometry actuator as described in [10]. It differs from the original

design, as shown in Fig. 1, as it has a fixed spring. As such, the actuator has a higher achievable bandwidth, improved packaging options and a lower complexity. Besides the actuator model, an electric motor and friction model are included. Using these combined models, a cascaded control structure is designed and validated. Furthermore, the power consumption of the actuator is evaluated under worst-case conditions.

The paper is structured as follows. In sections two and three, a model is presented of the variable geometry actuator under consideration and of the used electric motor respectively. Section four discusses the actuator control strategy and in section five the actuator power consumption is analyzed. The paper ends with the conclusions and an outlook on future work.

Fig. 1. The Delft Active Suspension (DAS), [15].

II. FORCE ACTUATOR MODEL

The variable geometry force actuator under consideration is inspired by the *Delft Active Suspension* [15], with the modification that it has a fixed spring [10], see Fig. 2. This is made possible, by using a flexible string that connects the spring to the outer end of a rotating arm. The benefits of this configuration over the conventional system with rotating spring include a higher achievable bandwidth (due to the lower inertia), better packaging (more compact system) and a reduced complexity (less pivots).

Fig. 2. Schematic representation of the variable geometry force actuator

The actuator consists of a wishbone with length l which is connected to a frame with a joint that allows a rotation α around the $\vec{e}_F^1$ -axis. Furthermore, attached to the wishbone at a distance l_a from the joint is an electric motor, which driven direction (γ) is perpendicular to the wishbone. The motor actuates an arm with length l_b , which is aligned with the $\vec{e}_F^1$ -axis for $\gamma = 0$. At the end of this arm, a string is connected through a rotational joint, which in turn passes through a hole in the frame. This hole coincides with the rotation axis of the electric motor, when the wishbone is aligned with the $\vec{e}_F^2$ -axis (is horizontal) and is located at a distance h_0 above the wishbone. At the other end of this string a plate is attached that constraints a spring with stiffness c_a and pre-tension F_s^0 . The force within this spring gives rise to a force F_{act} at the end of the wishbone. This is the actuator force, which varies for different values of α and γ .

The actuator is specifically designed for power efficiency. When the wishbone is horizontal ($\alpha = 0$), the outer end of the rotating arm describes a circle, of which the tangential velocity lies perpendicular to the spring force. As such, a fictional cone is created with height h_0 and radius l_b , similar to that obtained by the *Delft Active Suspension* [15]. Consequently, the rotating arm can be rotated over any angle γ without changing the spring elongation. So ideally, for $\alpha = 0$, in the absence of friction and inertia effects, any rotation γ (and thus any actuator force F_{act}) can be achieved without power usage. For any $\alpha \neq 0$ a disturbance moment (M_d), induced by the spring force, acts on the electric motor which needs to be compensated to maintain a certain angle γ . For simulation purposes, the actuator force F_{act} and the required motor moment M_r are required as a function of the angles α and γ . Therefore, a kinematic model is derived next using Lagrange's equations of motion [8].

We take as generalized coordinates $\underline{q} = [\alpha, \gamma]$. The kinetic energy is given by

$$T = \frac{1}{2} J_w \dot{\alpha}^2 + \frac{1}{2} J_r \dot{\gamma}^2, \quad (1)$$

with J_w and J_r the inertia's of the wishbone and electric motor with rotating arm and reductor respectively.

Fig. 3. Actuator spring elongation for $\alpha > 0$ and $\gamma = 90$ degrees.

From Fig. 3 it follows that the nonconservative forces can be written as

$$\underline{Q}_{nc} = \begin{bmatrix} -F_{load} l \cos \alpha \\ M_r \end{bmatrix}, \quad (2)$$

with M_r the driving moment and F_{load} the load from the suspended mass acting on the actuator.

The potential energy is given by

$$\begin{aligned} V &= \int_0^\epsilon (c_a y + F_s^0) dy \\ &= \frac{1}{2} c_a \epsilon^2 + F_s^0 \epsilon, \end{aligned} \quad (3)$$

where the spring compression ϵ is given by

$$\epsilon = \underbrace{|\vec{r}_{string}|}_{l_s} - l_s^0, \quad (4)$$

F_s^0 is the spring pre-tension and l_s^0 is the length of the string for $\alpha = 0$,

$$l_s^0 = \sqrt{h_0^2 + l_b^2}. \quad (5)$$

Fig. 3 shows that

$$\begin{aligned} \vec{r}_{string} &= \vec{r}_h - \vec{r}_s \\ &= \begin{bmatrix} -l_b \cos \gamma \\ l_a(1 - \cos \alpha) - l_b \sin \gamma \cos \alpha \\ h_0 - l_b \sin \gamma \sin \alpha - l_a \sin \alpha \end{bmatrix}^T \underline{e}_F. \end{aligned} \quad (6)$$

So, from (4) and (6) it follows that

$$\begin{aligned} l_s^2 &= l_b^2 + l_a^2(1 - \cos \alpha) + l_a l_b \sin \gamma(1 - \cos \alpha) + \dots \\ &\quad + h_0^2 - h_0 \sin \alpha(l_a + l_b \sin \gamma). \end{aligned} \quad (7)$$

Lagrange's equations of motion in the absence of constraints are (see for example [8]) given by

$$\frac{d}{dt} \left(\frac{\delta T}{\delta \dot{q}} \right) - \frac{\delta T}{\delta q} + \frac{\delta V}{\delta q} = \underline{Q}_{nc}^T. \quad (8)$$

Herein,

$$\frac{\delta V}{\delta q} = \begin{bmatrix} F_s^0 a_1 l_s^{-1} + c_a a_1 (1 - l_s^0 l_s^{-1}) \\ F_s^0 a_2 l_s^{-1} + c_a a_2 (1 - l_s^0 l_s^{-1}) \end{bmatrix}^T, \quad (9)$$

where

$$\begin{aligned} a_1 &= l_a^2 \sin \alpha + l_a l_b \sin \gamma \sin \alpha - h_0 l_a \cos \alpha - h_0 l_b \cos \alpha \sin \gamma \\ a_2 &= l_a l_b \cos \gamma(1 - \cos \alpha) - h_0 l_b \cos \gamma \sin \alpha. \end{aligned} \quad (10)$$

Combining (2, 8, 9), under steady-state conditions ($\ddot{\gamma} = \dot{\gamma} = \ddot{\alpha} = \dot{\alpha} = 0$) with the knowledge that $F_{act} = F_{load}$ it follows that

$$F_{act} = -\frac{1}{l \cos \alpha} (F_s^0 a_1 l_s^{-1} + c_a a_1 (1 - l_s^0 l_s^{-1})). \quad (11)$$

Moreover, under steady-state conditions, the disturbance moment acting on the electric motor is given by

$$M_d = -M_r = -F_s^0 a_2 l_s^{-1} - c_a a_2 (1 - l_s^0 l_s^{-1}). \quad (12)$$

The rotating arm with length l_b is driven by an electric motor, which is connected to the arm through a reductor with gear ratio

$$i = \frac{\omega_m}{\dot{\gamma}}, \quad (13)$$

where ω_m is the motor velocity. The arm, reductor and motor are assumed to behave as a single mass,

$$J_r \ddot{\gamma} = M_r. \quad (14)$$

The actuator behavior is illustrated using the parameters given in Table I, where the spring pre-tension is determined

TABLE I
ACTUATOR EXAMPLE PARAMETERS

parameter	value	unit
c_a	15.8	N/mm
m_l	350	kg
g	9.81	m/s ²
h_0	0.2	m
l	0.15	m
l_a	0.075	m
l_b	0.05	m

from the nominal design load m_l as

$$F_s^0 = m_l g \frac{l}{l_a} \frac{l_s^0}{h_0}. \quad (15)$$

Using these parameters and (11), the resulting steady-state actuator force can be plotted as a function of γ and α , see Fig. 4 (left). It can also be seen that the effective stiffness of the actuator c_{eff} (variation in actuator force as a function of $l \sin \alpha$) is overall much smaller than the stiffness of the spring c_a and is a nonlinear function of α and γ , see Fig. 4 (right).

Fig. 4. Actuator force (left) and effective actuator stiffness (right) for $\alpha = 0$ (solid), $\alpha = -15$ degrees (dashed) and $\alpha = 15$ degrees (dashed-dotted).

It is also possible to visualize the characteristics of the resulting disturbance moment, see Fig. 5. The disturbance moment on the electric motor will be largest around $\gamma = 0$ and will go to zero when $|\gamma|$ goes to 90 degrees. Moreover, it can be seen that due to the cone-principle, [15], no disturbance moment is present for $\alpha = 0$.

Fig. 5. Disturbance moment on electric motor for various α .

If the system inertias and the actuator motions are sufficiently small, the actuator behavior will follow this kinematic model. In the remainder of this paper it is assumed that J_w is negligible in comparison to the mass of the suspended mass. The remaining equation of motion is given by

$$J_r \ddot{\gamma} = M_r + M_d. \quad (16)$$

III. ELECTRIC MOTOR MODEL

The electric motor itself typically has a far larger bandwidth than the total actuator assembly. For that reason it might be warranted to assume exact tracking of any reference moment ($M_r = M_r^{ref}$). However, in order to gain insight in the power consumption of the actuator, it is necessary to obtain both motor current and voltage. Furthermore, under extreme circumstances saturation of current and/or voltage may occur. Therefore, a basic dc-motor model is adopted for the dynamic analysis, see [11].

The actual motor current is described by

$$L \frac{dI}{dt} = U - U_{emf} - RI, \quad (17)$$

where

$$U_{emf} = K \omega_m, \quad (18)$$

K is the motor constant, I is the current, U is the command voltage, L is the motor inductance and R is the motor resistance. Furthermore, the effective actuation moment is given by

$$M = (M_{rel} - M_{fric})i, \quad (19)$$

where the realized motor moment M_{rel} is given by

$$M_{rel} = KI, \quad (20)$$

and M_{fric} is the friction moment of the motor and reductor, which has to be incorporated in the model as it will be an important source of energy dissipation.

The friction moment is modeled using a *LuGre* model, [2] and is given by

$$M_{fric} = s_0 z + s_1 \dot{z} + s_2 \omega_m, \quad (21)$$

with s_0, s_1, s_2 friction parameters. Furthermore, z can be seen as the deflection of tiny bristles in the contact patch and

$$\dot{z} = \omega_m - \frac{|\omega_m|}{\beta} z. \quad (22)$$

Moreover,

$$\beta = \left\{ F_c + (F_s - F_c) e^{-(\omega_m/\omega_s)^2} \right\} / s_0, \quad (23)$$

where F_c, F_s and ω_s are parameters that determine the steady-state friction characteristic.

To illustrate the behavior of the friction model, we consider the friction parameters as given in Table II. These parameters are based upon motor specifications of a real dc-motor and our experience with this type of motor and reductor combination. Using these parameters, a simulation is run (stiff solver), with a sine-shaped velocity profile. The friction moment as a function of the rotational motor velocity is given in Fig. 6.

TABLE II
FRICTION EXAMPLE PARAMETERS

Friction parameters			Motor parameters		
parameter	value	unit	parameter	value	unit
s_0	15	Nm	K	0.1	V/rad/s
s_1	0.1	Nms	R	0.13	Ω
s_2	4.10^{-3}	Nms	L	$2.5 \cdot 10^{-4}$	H
F_c	0.2	Nm	i	59	-
F_s	0.555	Nm	J_r	0.70	kgm ²
ω_s	11.8	rad/s	I_{max}	75	A
			U_{max}	42	V

Fig. 6. Friction moment for sine-shaped velocity profile ($f = 0.1$ Hz). Increasing velocity (solid) and decreasing velocity (dash-dotted).

As the actuator has restrictions on the maximum current and voltage, the dynamic behavior of the actuator will also be limited. The effect of these limitations can be visualized by looking at the maximum size of a sine-shaped reference that can be followed, without saturation. The reference is given by

$$\gamma_{ref} = A \sin(2\pi f t). \quad (24)$$

Moreover, the reference and its derivatives are constrained as

$$\begin{aligned} |\gamma_{ref}| &\leq \gamma_{max} \\ |\dot{\gamma}_{ref}| &\leq \omega_m^{max} = \frac{U_{max}}{K i} \\ |\ddot{\gamma}_{ref}| &\leq \frac{M_{max} - |M_d|}{J} = \frac{i(K I_{max} - |M_{fric}|) - |M_d|}{J}, \end{aligned} \quad (25)$$

by the physical limitations of the actuator with electric motor. So, the actuator's working range is intrinsically limited by three constraints.

Again consider the parameters as given in Table II and assume that $\gamma_{max} = \pi/2$ rad. The working range of the actuator under these conditions, with $M_{fric} = 0$ and $\alpha = 0$ is represented in Fig. 7 by the area beneath the solid line. Furthermore, the three constraints are given by the dashed lines.

Fig. 7. Actuator working range (under the solid line) for $\alpha = 0$ and $M_{fric}^{max} = 0$, constraints (dashed) and third constraint for $M_{fric}^{max} = 0.9$ Nm, $\alpha = 15$ degrees (dotted).

The first constraint, given by the horizontal line can be increased by increasing γ_{max} . However raising it above 90 degrees will have no beneficial effect on the maximum actuator force. The second constraint, given by the dashed line with -1 slope, is determined by the maximum rotational velocity of the electric motor. It can be increased by increasing the maximum voltage U_{max} . The third constraint, given by the dashed line with -2 slope, is determined by the maximum rotational acceleration of the electric motor. It can be increased by increasing the maximum current I_{max} .

In reality the friction will be nonzero and also typically, there will be some rotation of the wishbone $|\alpha| > 0$. These two conditions will give rise to a shift of the third constraint. From Fig. 6 it can be derived that the maximum friction moment for high rotational velocities is approximately 0.9 Nm. When applying this friction moment and a maximum disturbance moment (for $\alpha = 15$ degrees at $\gamma = 0$), the worst-case working range limitation can be approximated. This approximation is given in Fig. 7 by the dotted line.

IV. ACTUATOR CONTROL

A cascaded control strategy is chosen to control the actuator. There is a motor controller, angle controller and force controller. The reason for choosing this strategy lies within the wish to constraint the reference angle to ± 90 degrees. As a result, it is possible to use linear techniques to generate the reference angle as a function of the reference force and actual force. The three controller modules are shortly discussed in the following sections.

A. Electric motor control

The inner most control loop is that of the torque controller, see figure 8. It consists of a standard PI-controller with anti-windup,

$$C_m = P_m + \frac{I_m}{s} \Big|_{U_{max}}, \quad (26)$$

and a resistance feed forward,

$$U_m^{ff} = RI_{ref}. \quad (27)$$

Herein, P_m , I_m are control gains, the transfer function H_m follows from (17),

$$H_m = \frac{M_{rel}(s)}{U(s) - U_{emf}(s)} = \frac{K}{Ls + R}. \quad (28)$$

and s is the Laplace variable. The controller brings the actual current close to the reference current and thus the realized moment M_{rel} , see (19) and (20), close to the reference moment

$$M_{ref} = KI_{ref}. \quad (29)$$

Moreover, both the maximum voltage and current are constraint (saturation blocks).

Fig. 8. Block-scheme of the motor control-loop.

B. Angle control

The second feedback loop is that of the angle controller, see Fig. 9. It consists of a lead-lag filter with integral action and anti-windup,

$$C_a = P_a \left(\frac{s + D_a}{s + L_a} \right) + \frac{I_a}{s} \Big|_{I_a^{max}}. \quad (30)$$

The closed-loop system H_{cm} is shown in Fig. 8 and H_{fric} is defined by (21, 22, 23).

The actuator hardware dynamics are modeled as

$$H_a : \gamma = \frac{M + M_d}{J_r s^2}, \quad (31)$$

where J_r is the inertia of the system (after the reductor), M_d is given by (12) and M is given by (19). Moreover, the relationship between F_{act} and γ is considered to be geometrically determined (no dynamics) and given by (11).

Fig. 9. Block-scheme of the angle control-loop.

C. Force control

The highest level control consists of the force controller. Its task is to minimize the difference between the generated actuator force and the reference actuator force. It consist of a feed forward part

$$\begin{aligned} \gamma_{ref}^{ff} &= C_f^{ff}(F_{ref}) \\ &= \sin^{-1} \left(\frac{l_s^0 F_{ref}}{l_b h_0 F_s^0} - \frac{l_a}{l_b} \right), \end{aligned} \quad (32)$$

which prescribes the "right" reference γ for $\alpha = 0$ and a feedback part

$$\begin{aligned} \gamma_{ref}^{fb} &= C_f(F_{ref} - F_{act}) \\ C_f &= P_f \left(\frac{s + D_f}{s + L_f} \right) + \frac{I_f}{s} \Big|_{I_f^{max}}. \end{aligned} \quad (33)$$

which corrects for disturbances α . This way, it is possible to filter α such that high frequent influences are reduced (which is beneficial for power consumption). An alternative would be to adjust for α with feed forward. Furthermore, the total reference angle γ_{ref} is bounded

$$\gamma_{ref} = \left(\gamma_{ref}^{ff} + \gamma_{ref}^{fb} \right) \Big|_{-\pi/2}^{\pi/2}. \quad (34)$$

Using the controller parameters as given in Table III (which are a result of tuning), a 10 Hz reference force with a (maximum) amplitude of 200 N can be tracked reasonably well, see Fig. 11.

Fig. 10. Block-scheme of the force control-loop.

TABLE III
CONTROLLER EXAMPLE PARAMETERS

parameter	value	parameter	value
P_a	1500	P_f	0.00026
D_a	4.49	D_f	10.47
L_a	94.25	L_f	94.24
I_a	4.49	I_f	0.0063
I_a^{max}	6	I_f^{max}	1.57

Fig. 11. Tracking a sine-shaped force reference with a 200 N amplitude and a frequency of 10 Hz for $\alpha = 0$, when starting from standstill. Simulated actuator force (solid) and reference force (dash-dotted) (left) and simulated power consumption (right).

V. ACTUATOR POWER CONSUMPTION

In order to get an idea of the worst case power consumption of the actuator several simulations are performed. First, for $\alpha = 0$, several sine shaped force references are tracked. Herein the amplitude and frequency of the sines is varied. The maximum amplitude A for a certain frequency is obtained from Fig. 7. The results are given in Fig. 12 both for simulations with (left) and without friction (right). It can be seen that the mean power consumption rises as the frequency of the sine rises. In other words, high frequent reference signals have a relatively large power consumption. Furthermore, it can be seen that friction effects account for the major part of the used power below 10 Hz.

Fig. 12. Actuator power consumption for $\alpha = 0$, when tracking sines of varying frequency and various amplitudes: maximum amplitude A (solid); $A/3$ (dash-dotted); and $A/10$ (dotted). Simulations with friction (left) and without friction (right).

On the other hand, from simulations using the maximum amplitude A and various fixed α , it followed that the influence of α on the power consumption under extreme circumstances is limited.

VI. CONCLUSIONS AND FUTURE WORK

A model has been derived for the variable geometry actuator under consideration. It is used, in combination with an electric motor model and LuGre friction model, to design a cascaded controller. Using a simulation example it is shown that the controlled actuator can easily track a 10 Hz sine with an amplitude of 200 N. Furthermore, the power consumption of the actuator under worst case conditions is evaluated. From this evaluation it follows that high frequent force reference signals have a relatively high influence on the power consumption. Moreover, it is shown that the influence of friction below 10 Hz is substantial. Still, the worst case power consumption is much lower than that of hydraulic actuators under the same conditions, [6], [14].

Future research will focus on implementation of this actuator in (full) vehicle simulation models and design of suitable active suspension vehicle controllers. This way, the added value of such an actuator can be evaluated under real-life conditions. Furthermore, experimental validation of the variable geometry actuator might also become a topic when the added value is proven by vehicle simulations.

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