Robust Sliding mode Control for a frequency approximated nonlinear model of a micro cantilever

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Abstract: In the present article a sliding mode controller is proposed for a micro-cantilever beam with fringing and squeezed film damping effects. The narrow micro cantilever beam can move via the application of an external electrically induced force. The introduction of the squeeze film parameters results in a frequency-dependent nonlinear system. Particular attention has been paid in order to approximate the frequency dependend model, with a valid, frequency independent one, in order to design a robust sliding mode controller. The suggested control technique enables compact realization of a robust controller tolerant in device characteristics’ variations, nonlinearities and types of inherent instabilities. Robustness of the proposed control scheme against disturbances is proved by Lyapunov’s second method. Finally, bifurcation analysis is presented in order to test the system’s characteristic values. Simulation results prove the efficacy of the suggested control technique.

Keywords: Micro beam, Nonlinear Control, Sliding mode control, MEMS

1. INTRODUCTION

MEMS cantilevers are one of the most popular resonators used in various applications of micro and nano technologies. The simplicity of their structure, the ease of fabrication over a wide range of dimensional vibrations and the ease of excitation and resonance measurements, are the primary reasons for their popularity. Several types of actuation methods for such structures as electromagnetic, electrostatic, piezoelectric, electrothermal have been investigated recently Chu et al. (2009). However electrostatic actuation is the most widely utilized method for the design of specific devices such as switches, micro mirrors and micro resonators Zolotas et al. (2007). In such narrow micro cantilever beams undergoing large deflections Rottenberg et al. (2007), the effects of the fringing fields on the electrostatic force are not negligible because of the non zero thickness and finite width of the beam. The inclusion of the effects of the fringing field capacitance gives a more accurate model of the cantilever beam structures.

Due to the dimensional constraints of MEMS devices, the air squeeze damping is a parameter that plays a dominant role in their performance, design and control. For micro devices with a proof mass that moves against a trapped film, squeeze film air damping has been of a great importance as the mechanism dominates the damping and thus substantially affects the system’s frequency response. The dynamic behavior of accelerometers optical switches, micro-torsion mirrors, resonators, etc is significantly related to the squeeze film air damping of the mechanical structures. The presence of the squeezed film damping effect increases the complexity of MEMS design while making the system’s description more accurate.

An important parameter of electrostatic microactuators is the pull in voltage. In static equilibrium, the electrostatic and mechanical forces are equal to each other, resulting in a stable condition of the micro actuator. As the applied voltage increases the electrostatic force overcomes the mechanical one, resulting in instability or collapse conditions and as a result a contact between the two plates is formed.

In the present work a sliding mode controller is proposed for the nonlinear frequency independent structure of a microbeam that experiences parametric uncertainties Yazdi et al. (2003, 2004). Sliding mode control is a particular type of Variable Structure Control (VSC) characterized by a suite of a feedback control law and a decision rule known as switching function. The overall control design is based on the usage of Lyapunov functions. In this certain case, the control law is designed so that the system trajectory always reaches a desired sliding surface. Once the sliding surface is reached, the control structure is changed, in order to maintain the system on that surface. High frequency control switching, leads to the so called chattering effect which is exhibited by high frequency vibration of the controlled plant and can be dangerous when applied on the system. In order to avoid this problem the discontinuous term of the switching function is replaced by a smooth and continuous one. In addition, the controller scheme incorporates the system’s parametric uncertainties quite common in such MEMS devices that are highly nonlinear with micro scale effects He et al. (2009); Vagia and Tzes (2009, 2010).

The present article is structured as follows: the modeling of the micro cantilever beam with fringing and squeezed film damping effects is presented in Section 2. In Section 3 the design of the state feedback control scheme is presented. Simulation results that prove the efficacy of the proposed control architecture are presented in Section 4, while the Conclusions are drawn in the last Section.

2. MODELING OF THE ELECTROSTATIC MICRO CANTILEVER BEAM WITH FRINGING AND SQUEEZED FILM DAMPING EFFECTS

The electrostatically actuated μCB is an elastic beam suspended above a ground plate, made of a conductive material. The cantilever beam moves under the actuation of an electrostatic force. The conceptual geometry of an electrostatic actuator composed of a cantilever beam separated by a dielectric spacer of the fixed ground plane is shown in Figure 1.
The governing equation of motion of the µCB presented in Figure 1, is obtained if considering that the mechanical force of the beam is modeled in a similar manner to that of a parallel plate actuator with a spring element Batra et al. (2006).

The dynamical equation of motion due to the mechanical, electrostatic and damper force is equal to:

\[ m\ddot{\eta} + F_d + k\eta = F_{el} \]  

(1)

where \( \eta \) is the displacement from the relaxed position, \( m \) is the beam’s mass, \( k \) is the spring’s stiffness, \( F_d \) is the force caused by the squeezed film damping effect and \( F_{el} \) is the electrically-induced force Chowdhury et al. (2005); Batra et al. (2006); Sun et al. (2007).

The cantilever beam shown in Figure 1 can be viewed as a semi-infinitely VLSI on-chip interconnect separated from a ground plane (substrate) by a dielectric medium (air). If the bandwidth-airgap ratio is smaller than 1.5, the fringing field component becomes the dominant one. The electrically introduced force in this case equals to Vagia and Tzes (2010):

\[ F_{el} = \frac{e_0\ell w(U^2)}{2(\eta_{max} - \eta)^3} + \frac{0.1325e_0w^{0.25}U^2}{(\eta_{max} - \eta)^{1.25}} \]

\[ + \frac{0.265e_0h^{0.3}U^2}{(\eta_{max} - \eta)^{1.5}} + \frac{0.1325e_0\ell w^{0.25}U^2}{(\eta_{max} - \eta)^{1.25}} \]  

(2)

2.1 Squeezed Film Damping Effect

The behavior of the gas between the two surfaces is in general governed by both viscous and inertial effects within the fluid. In the present paper, the effect of non-uniform gap variation between the oscillating structure and the fixed substrate, as shown in Figure 1, on the squeeze film damping force Li (2008) due to flexibility of the structure, is presented. In order to calculate the squeeze film damping effect for a system where the displacement is non-uniform, the elastic equation that gives the dynamic displacement has to be coupled with the conventional Reynolds equation Pandey (2007).

The effect of flexibility on squeeze film damping, involves the coupling of plate vibration equations and the Reynolds equations under the assumptions of small strains and displacements Blench (1983). The coefficients of the viscous damping force, due to the squeezed film air damping are (under a sinusoidal motion with frequency \( \omega \)) Nayfeh (2004); Li and Fan (2009):

\[ b_1 = \frac{p_0\ell w}{\omega(\eta_{max} - \eta)} + \frac{64\sigma}{\pi^3} \sum_{j,m=\text{odd}} (jm)^2 \left( \frac{b_2}{(\eta_{max}^2 + \eta^2)^2 + \sigma^2/\pi^4} \right) \]  

(3)

where

\[ \sigma = \frac{12\mu w^2}{P_a(\eta_{max} - \eta)^2} \]  

(5)

\( P_a \) is the ambient pressure, \( \mu \) is the viscosity coefficient, and \( x = w/\ell \) is the aspect ratio of the cantilever structure. The parameter \( b_{\text{min}} \) is expressed as:

\[ b_{\text{min}} = \frac{j^2\pi^2}{2(-1)^{j+1}} \frac{1}{b_0} + \frac{j^2\pi^2}{4(\sigma^2 + \pi^4)} \frac{\eta_1(\eta_1 + \gamma(\sin(\alpha) + \cos(\alpha)))}{b_1} \]  

(6)

where the parameters \( \alpha, \gamma, b_1 \) are equal to:

\[ \alpha = \frac{1}{(\eta_{max} - \eta)} \sqrt{\frac{12\mu}{P_a}} \]  

(7)

\[ \gamma = -\frac{(\cos(\alpha) + \cos(\alpha))}{(\sin(\alpha) + \sin(\alpha)),} \]  

\[ b_1 = \frac{(\cos(\alpha) - \cos(\alpha) + \gamma(\sin(\alpha) + \sin(\alpha)))}{b_1} \]  

(9)

for the first spatial deflection mode \( \alpha = 1.875. \) As a result, the damping force \( F_d \) caused by the thin film of air appearing in Equation (1) is equal to:

\[ F_d = b_1(\eta, \omega)\eta + k_1(\eta, \omega)\eta. \]  

(10)

The nonlinear equation of motion presented in Equation (10) can be rewritten according to (10):

\[ m\ddot{\eta} + b_1(\eta, \omega)\eta + (k_1(\eta, \omega) + k)\eta = \frac{e_0\ell w(U^2)}{2(\eta_{max} - \eta)^3} + \frac{0.1325e_0w^{0.25}U^2}{(\eta_{max} - \eta)^{1.25}} \]

\[ + \frac{0.265e_0h^{0.3}U^2}{(\eta_{max} - \eta)^{1.5}} + \frac{0.1325e_0\ell w^{0.25}U^2}{(\eta_{max} - \eta)^{1.25}} \]  

(11)

Equation (11) is a nonlinear equation due to the presence of the terms \( \omega, \eta, U. \) When \( \omega \in [0, 10]^3 \) Hz there is no significant change in the values of \( b_1, k_1 \) in this frequency range. In respect to this observation, the value of the damping and elastic spring factors may be approximated nonlinearly, in terms of \( \eta. \) The functions describing the approximated, frequency independent quantities are equal to:

\[ b_{\text{min}}(\eta) = p_1\eta^2 + p_2\eta + p_3 \approx b_1(\eta, \omega), \forall \omega \in [0, \omega_{\text{max}}]. \]  

(12)

\[ k_{\text{min}}(\eta) = r_1\eta^2 + r_2\eta + r_3 \approx k_1(\eta, \omega), \forall \omega \in [0, \omega_{\text{max}}]. \]  

(13)

It should be noticed that for typical gaps encountered in µCBs (0.1 \( \mu m \leq \eta_{\text{max}} \leq 40 \mu m \)) over an operating frequency range of less than \( \omega_{\text{max}} = 10^2 \) Hz, the coefficient \( k_{\text{min}} \) increases with the frequency and is significantly smaller than \( k, \) \( k_{\text{min}} < k, \forall \omega \in [0, \omega_{\text{max}}] \) and so can be omitted.

The final nonlinear system with the approximated damping term is equal to:

\[ m\ddot{\eta} + b_{\text{min}}(\eta)\eta + k\eta = \frac{e_0\ell w(U^2)}{2(\eta_{max} - \eta)^3} + \frac{0.1325e_0w^{0.25}U^2}{(\eta_{max} - \eta)^{1.25}} \]

\[ + \frac{0.265e_0h^{0.3}U^2}{(\eta_{max} - \eta)^{1.5}} + \frac{0.1325e_0\ell w^{0.25}U^2}{(\eta_{max} - \eta)^{1.25}} \]  

(14)
3. SLIDING MODE CONTROLLER DESIGN

A special approach to robust controller design for nonlinear models, considering discrepancies due to unmodeled dynamics, variation in system’s parameters or the approximation of complex behavior by the straightforward model is the so-called sliding mode control methodology. Sliding mode control is a particular type of Variable Structure Control (VSC) Yazdi et al. (2004). This control method is characterized by a suite of feedback control law and a decision rule known as switching function. Consider the second-order nonlinear dynamic expression of the system presented in Equation (14):

\[ \eta - \bar{\eta} = \frac{\phi(\eta, \bar{\eta})}{\beta(\eta)} + \frac{1}{m} \left[ \frac{e_\psi \psi(t)}{2[q_0^T - \eta]} + \frac{0.1325 e_\psi \psi(t)}{(q_0^T - \eta)^{2.5}} + \frac{0.265 e_\psi \psi(t)}{(q_0^T - \eta)^{2.5}} + \frac{0.1325 e_\psi \psi(t)}{(q_0^T - \eta)^{2.5}} \right] \eta(t) \]

\[ \dot{\eta} = \dot{\bar{\eta}} + \dot{\beta}(\eta) \mu(t) \]

where \( \phi(\eta, \bar{\eta}) \) and \( \beta(\eta) \) are unknown but bounded real continuous functions, \( u(t) = U^2 \in \mathbb{R}^+ \) and \( y \in \mathbb{R}^+ \) are continuous input and system output respectively, and \( H = [\eta, \dot{\eta}]^T \in \mathbb{R}^2 \) is a state vector of the system which is assumed to be available for measurement. In a real experimental case, measurement of the state vector can be difficult especially when the velocity is concerned. However, this problem can be surpassed via the design of an observer.

For the system to be controllable it is required that \( \beta(\eta) \neq 0 \) for all \( \eta \) in a certain controllability region \( U_c \in \mathbb{R}^2 \). Since \( \beta(\eta) \) is continuous, without loss of generality, it is assumed that \( \beta(\eta) > 0 \) for all \( \eta \in U_c \). Assuming that all parameters of the system are well known precisely, the nominal system of the system will be equal to:

\[ \dot{\bar{\eta}} = \hat{\phi}(\eta, \bar{\eta}) + \hat{\beta}(\eta) \mu(t) \]

where \( \hat{\phi}(\eta, \bar{\eta}), \hat{\beta}(\eta) \) are the nominal values of \( \phi(\eta, \bar{\eta}), \beta(\eta) \) respectively. As a result, the system Equations described in (15,16,17) can be reformulated as:

\[ \dot{\bar{\eta}} = \bar{\eta} \]

\[ \bar{\eta} = \dot{\phi}(\eta) + \dot{\bar{\eta}} + \frac{\Delta \phi(\eta)}{\dot{\beta}(\eta)} \mu(t) \]

with \( \Delta \phi(\eta, \bar{\eta}), \Delta \beta(\eta) \) are the uncertainty values of the system. The following assumptions need to be made:

\[ \Delta \phi = [\phi(\eta, \eta) - \phi(\eta, \bar{\eta})] \leq G(\eta, \eta) \]

\[ \Delta \beta = [\beta(\eta) - \beta(\bar{\eta})] \leq \Lambda(\eta) \]

The control objective, is to find a suitable control law so that the vector \( H \) can track a specified bounded reference trajectory \( H_d = [\eta_d, \dot{\eta}_d]^T \in \mathbb{R}^2 \). The design of a sliding mode control system can be divided into two steps: a) finding a sliding function vector and b) design a proper control vector in order to meet the requirements of sliding mode reaching condition. In this case, the sliding function is defined as:

\[ s = \dot{e} + \lambda e \]

with \( \lambda \) a strictly positive constant. Defining the tracking error vector as \( E = [e, \dot{e}]^T \) whose elements are defined by the following equations:

\[ e = \eta - \eta_d \]

\[ \dot{e} = \dot{\eta} - \dot{\eta}_d \]

Theorem 1. For a nonlinear system as the one described in Equations (15,16,17), if Equations (21, 22) are satisfied, and if a control law is \( u = u_{eq} + u_c \), is applied then the tracking errors \( E = [e, \dot{e}]^T \) converge to zero.

In order to prove the above theorem, the values \( u_{eq} \) and \( u_c \) of the controller are assumed to be equal to:

\[ u_{eq} = \frac{1}{\hat{\beta}(\eta)} \psi \]

\[ u_c = -\frac{K}{\hat{\beta}(\eta)} \text{sign}(s) \]

with the values of \( K \) and \( \psi \) defined as follows:

\[ K \geq \Lambda(\eta) ||\psi|| + \frac{\beta(\eta)}{\beta(\eta) - \Lambda(\eta)} G(\eta) + \rho \]

and also \( \rho \) a strongly positive constant and

\[ \psi = -\dot{\lambda} + \dot{\eta}_d - \hat{\phi}(\eta, \bar{\eta}) \]

In the following the proof of the above Theorem is carried in order to show that the control law applied on the system will be equal to \( u = u_{eq} + u_c \) as described before.

Proof 1. In order to prove the Theorem 1, the first derivative of \( s \) is needed:

\[ \dot{s} = \dot{\dot{e}} + \dot{\lambda} e \]

\[ = (\eta - \eta_d) + \dot{\lambda} \eta \]

\[ = \phi(\eta, \bar{\eta}) + \beta(\eta) u - \eta_d + \dot{\lambda} e \]

\[ \Rightarrow \dot{s} = \dot{\phi}(\eta) + \Delta \phi(\eta, \eta) + \dot{\beta}(\eta) + \Delta \beta(\eta) \eta - \eta_d + \dot{\lambda} (\eta - \eta_d) = 0 \]

The equivalent control for the nominal system is the control such that \( s = 0 \), for the system without uncertainties \( \Delta \phi(\eta, \eta) = \Delta \beta(\eta) = 0 \). The control law designed, that would give the nominal system the desired closed loop dynamics, will be called equivalent control law \( u_{eq} \) and will be equal to:

\[ u_{eq} = \frac{1}{\hat{\beta}(\eta)} [ -\dot{\lambda} (\eta - \eta_d) - \dot{\phi}(\eta, \bar{\eta}) + \hat{\eta}_d ] \]

By the choice of a Lyapunov function candidate equal to \( V = \frac{1}{2} s^2 \), a reaching condition is obtained as:

\[ V = \frac{1}{2} s^2 \leq -\rho |s|, s \neq 0 \]

with \( \rho \) a strictly positive number.

Essentially, Equation (32) states the squared “distance” to the surface, as measured by \( s^2 \), decreases along all system trajectories. So, this Equation, provides a sufficient reaching condition such that the tracking errors \( E = [e, \dot{e}]^T \) will asymptotically converge to zero. In order to meet that condition the control law \( u_{eq} \) is augmented by a switching control law, \( u_c \).
\[ \dot{\varepsilon} = \dot{\phi} (\eta, \dot{\eta}) + \Delta \phi (\eta) + (\dot{\beta} (\eta) + \Delta \beta (\eta)) \mu + \lambda (\dot{\eta} - \dot{\eta}_d) - \ddot{\varepsilon}. \]  

(33)

The real dynamics of the surface of the system with uncertainties is expressed by:

\[ \dot{\varepsilon} = \frac{\Delta \beta (\eta)}{\beta (\eta)} - \lambda (\dot{\eta} - \dot{\eta}_d) + \Delta \phi (\eta, \dot{\eta}) + \mu \text{sign}(s) \]  

(34)

with the value of \( \mu \) being equal to:

\[ \mu = 1 + \frac{\Delta \beta (\eta)}{\beta (\eta)} \]  

(35)

Taken into account the uncertainties given in Equations (21,22), the first time derivative of the Lyapunov function candidate \( V = \frac{1}{2} \dot{\varepsilon}^2 \) is equal to:

\[ V = s \left( \Delta \phi (\eta, \dot{\eta}) + \Delta \beta (\eta) \beta^{-1}(\eta) \left[ -\lambda (\dot{\eta} - \dot{\eta}_d) + \dot{\phi} (\eta, \dot{\eta}) \right] \right) - s \left( \left( \dot{\beta} (\eta) + \Delta \beta (\eta) \right) \dot{\beta}^{-1}(\eta) \text{sign}(s) \right) \]  

(36)

The condition (32) is met if:

\[ K \geq \frac{\Delta \beta (\eta)}{\beta (\eta) + \Delta \beta (\eta)} \left| \psi \right| + \frac{\dot{\beta} (\eta)}{\beta (\eta) + \Delta \beta (\eta)} \left( \Delta \dot{\phi} (\eta) \right) + \rho \]  

Recall that \( \left| \Delta \dot{\phi} (\eta, \dot{\eta}) \right| < G \) and \( \Delta \beta (\eta) \) and under the condition of \( V \leq -\rho |s| \) the sufficient condition of asymptotic convergence is obtained if:

\[ K \geq \frac{\Delta \beta (\eta)}{\beta (\eta) + \Delta \beta (\eta)} \left| \psi \right| + \frac{\dot{\beta} (\eta)}{\beta (\eta) + \Delta \beta (\eta)} \left( G + \rho \right) \]  

which proves that \( \dot{s} < 0 \).

The final control law describing the system has been proven to be equal to Yazdi et al. (2004):

\[ u = u_a + u_c = \frac{1}{\beta (\eta)} - \lambda (\dot{\eta} - \dot{\eta}_d) + \dot{\phi} (\eta, \dot{\eta}) - \frac{K}{\beta (\eta)} \text{sign}(s) \]  

(38)

In practice the signum function (\text{sign}(s)) in Equation (38) can be replaced by a continuous function to alleviate chattering where \( \gamma \) is a positive scalar constant.

4. SIMULATION RESULTS

Simulation studies were carried on a \( \mu CB \)'s non-linear model. The parameters of the system unless otherwise stated are equal to those presented in the following Table.

<table>
<thead>
<tr>
<th>Parameter (Unit)</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( w (m) )</td>
<td>Beam Width</td>
<td>( 7.5 \times 10^{-9} )</td>
</tr>
<tr>
<td>( h (m) )</td>
<td>Beam Height</td>
<td>( 1.2 \times 10^{-6} )</td>
</tr>
<tr>
<td>( l (m) )</td>
<td>Beam Length</td>
<td>( 100 \times 10^{-6} )</td>
</tr>
<tr>
<td>( \eta_{\text{max}} (m) )</td>
<td>Maximum Distance</td>
<td>( 4 \times 10^{-6} )</td>
</tr>
<tr>
<td>( \mu (kg \cdot m/sec^2) )</td>
<td>Viscosity Coefficient</td>
<td>( 18.5 \times 10^{-6} )</td>
</tr>
<tr>
<td>( \rho (kg/m^3) )</td>
<td>Density</td>
<td>( 1.155 )</td>
</tr>
<tr>
<td>( \varepsilon_0 (Coul^2/N^m^2) )</td>
<td>Dielectric constant of the air</td>
<td>( 8.85 \times 10^{-12} )</td>
</tr>
<tr>
<td>( P_0 (N/m^2) )</td>
<td>Ambient Pressure</td>
<td>( 10^5 )</td>
</tr>
<tr>
<td>( k (N/m) )</td>
<td>Stiffness of the spring</td>
<td>( 0.249 )</td>
</tr>
</tbody>
</table>

The beam’s gap is (for the open-loop system) constrained within \( \eta \in [0, 1.33] \mu m \) resulting in an operating regime below the well-known bifurcation point (stable system). These are the points where the behavior of the system changes from stable to unstable and vice versa.

In Figure 2, the x-axis corresponds to \( \eta \), the y-axis corresponds to the value of the frequency \( \omega \) and the z-axis corresponds to the measured quantity. The extrema of these graphs present the bifurcation points of the system. In this case, there is a change in regard to the location of the bifurcation points of the system depending on the values of \( \omega \). In such cases, there exists a single bifurcation point at \( \eta^b = \eta_{\text{max}}^b \). In Figure 3 the extrema of the plot describes the bifurcation point of the approximated, frequency independent system described in Equation (14). In the present Figure, the x-axis corresponds to \( \eta \) while the y-axis corresponds to the values of the applied voltage \( U \). As it is shown in the plot, the approximated system has a bifurcation point in \( \eta^b = \eta_{\text{max}}^b \).

![Fig. 2. Bifurcation points of the \( \mu CB \) for different \( \omega \)](image)

![Fig. 3. Bifurcation points of the approximated system \( \mu CB \)](image)
Fig. 4. Electrical forces with and without the fringing effects because of the thin film damping effect can be omitted as it is described in the system modeling section.

Fig. 5. Values for $k_a$ and $b_a$ with respect to the distance

The following Figures 6-8 present the results of the proposed control scheme when the sliding mode controller is applied. Figure 6 (7) presents the responses (control input) of the system for a reference trajectory and for different values of the $\lambda$ parameter. The applied controller achieves good tracking performance and the value of $\lambda$ plays a significant role in the systems’ response. The greater the value of $\lambda$ the more oscillatory, but on the other hand the faster the system’s response becomes. So, the designer needs to compensate between the velocity and smoothness of the system in order to decide which $\lambda$ value is appropriate. Finally, Figure 8 presents the phase portraits of the closed loop system for different $\lambda$ values. Starting from any initial condition, the state trajectory reaches the the time varying surface on a finite time, and then slides along the surface towards $\eta_d$ exponentially, with a time-constant equal to $\lambda$.

5. CONCLUSION

In this article, a sliding mode controller has been designed for an approximated-linearized model of a $\mu$CB with fringing and squeezed film damping effects. The robust sliding mode control scheme, achieves robustness against disturbances and terminal accuracy. The proposed control requires disturbance limit and therefore exact information of disturbance is not necessary. The effectiveness of the presented method is proved by the second Lyapunov method and the robustness of the controller against disturbances is demonstrated by simulation results. Also, frequency variations are inquired in order to find any relation between the frequency and the bifurcation points of the system. The overall control scheme is applied in numerous simulation test cases in order to prove its efficacy.

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