

Precision Machine Design- Slider

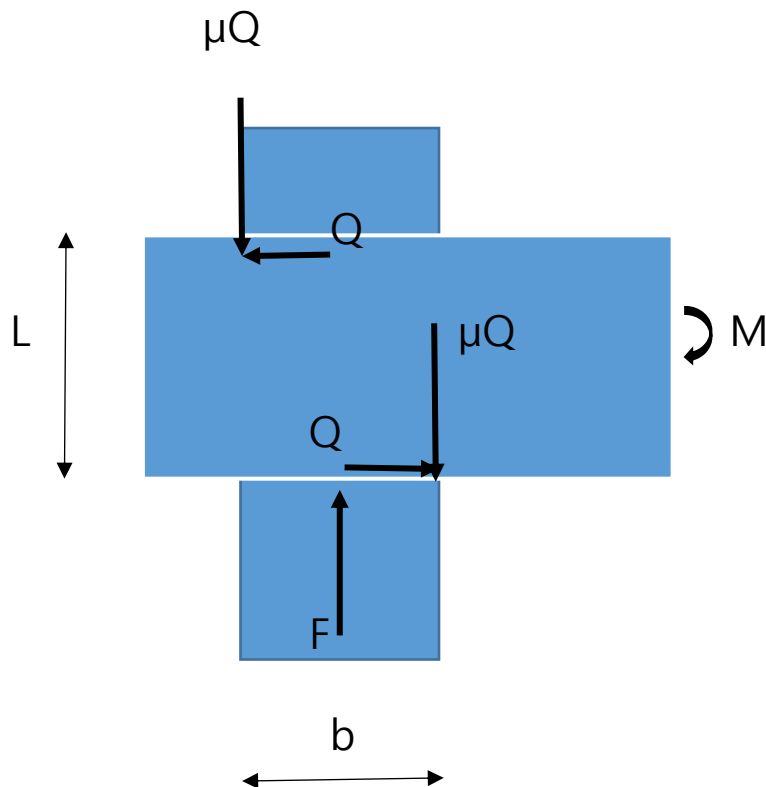
Long Slider

The minimum driving force is always desirable, as it gives less elastic deformation and needs minimum energy for less potential heat sources and thermal deformation. The slider design is one of key issues in translational motion, and the long slide is preferable for the high precision linear motion.

The long slider applications are shown in the following:

(source: Nakazawa's Principles of Precision Engineering)

1) Driving force at the centre of width of slide



Q: Normal force; μ : Friction coefficient; F: Driving force

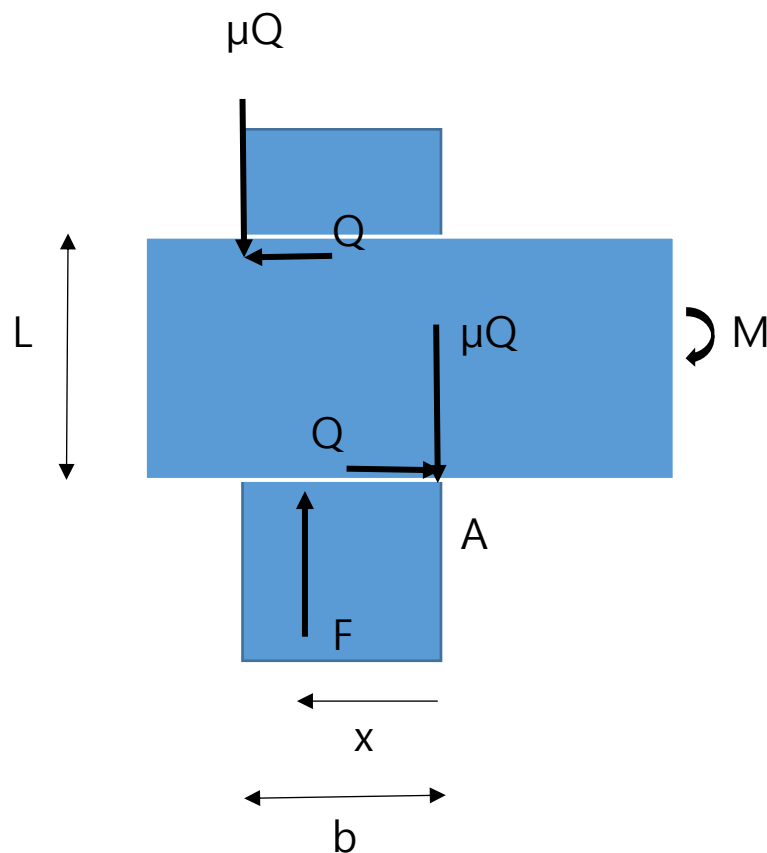
$F=2\mu Q$ when slide moves, and M is the applied moment due to payload. From equilibrium at the acting point F,

$$M - QL = 0 \therefore Q = M/L$$

Thus $F = 2\mu Q = 2\mu M/L$, and

Thus, large L gives small F.

2) Driving force at the offset from the centre of slide width



Force and moment equilibrium at A gives

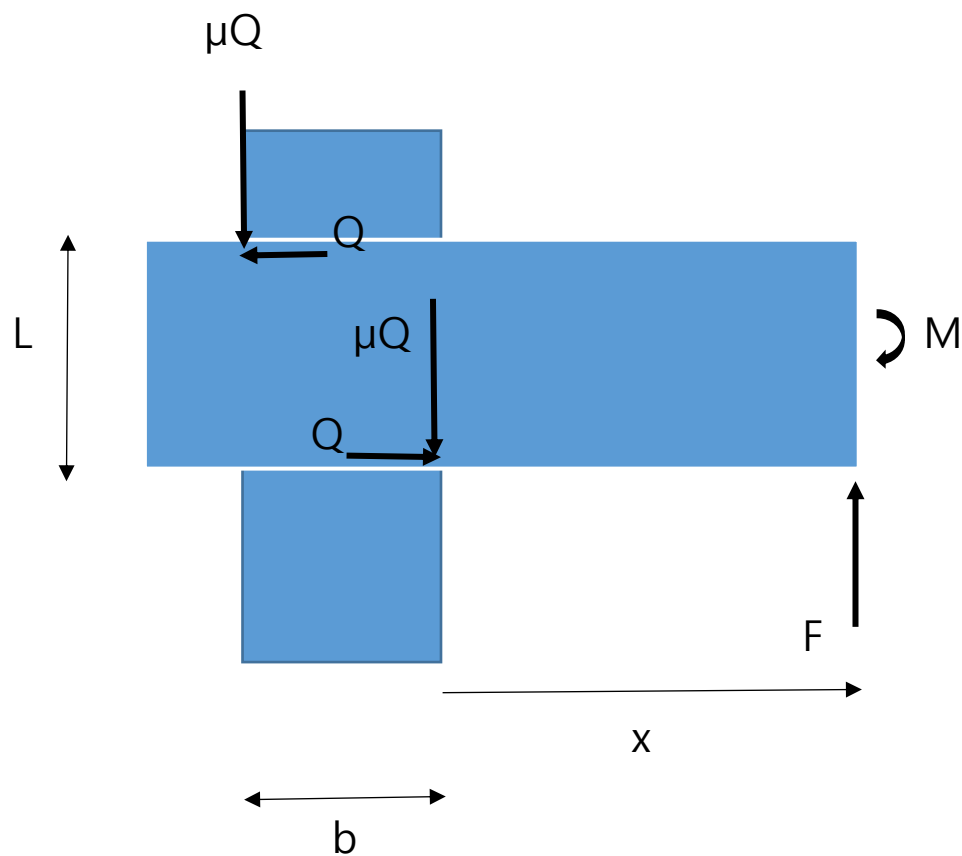
$$M + Fx - \mu Qb - QL = 0, \text{ and } F = 2\mu Q$$

$$\text{Thus } Q = M / [L + \mu(b - 2x)],$$

and $F = 2\mu M / [L + \mu(b - 2x)]$ when the slider moves

\therefore Large L with small friction gives small F .

3) Driving force outside the width of slide



Moment and force equilibrium at the point of F, thus

$$F=2\mu Q \text{ and}$$

$$M-\mu Qx-\mu Q(b+x)-QL=0$$

$$M=Q[L+\mu(b+2x)]=0$$

$$\therefore F=2\mu Q=2\mu M/[L+\mu(b+2x)] \text{ when the slide moves}$$

Thus large L with small μ gives small F.

Therefore the long slide is always recommended, in order to give the small driving force.

2. Location of Driving Force

: For high precision linear motion, the driving force should be applied at or close to the centre of the resisting forces including the inertia force, considering the moment equilibrium. This is to prevent or minimize any unwanted or unnecessary moment along the direction of constrained DOF. The unwanted or unnecessary moment may cause the unwanted deformation/motion/vibration along the constrained DOF and thus it may generate less precise motion along the planned DOF.

Some illustrating examples;

Resisting forces L_1 , L_2 are applied at A, B locations, respectively, as shown; then the resultant force, L_R , is applied at C location, considering the moment equilibrium. Thus the driving force is the same as the L_R in the opposite direction.

When the driving force is applied at the C' location with offset h from the C position, the unnecessary moment, $L_R h$, is generated, and it may cause the unwanted pitch angular motion of slide, depending on the stiffness of the bearing pad.

When the slide moves with acceleration a , the inertia force, $L_i = m a$, should be considered as it is applied at the centre of gravity of the moving slide, while m is the mass of the moving slide. The centre of gravity, \mathbf{r}_G , can be calculated as

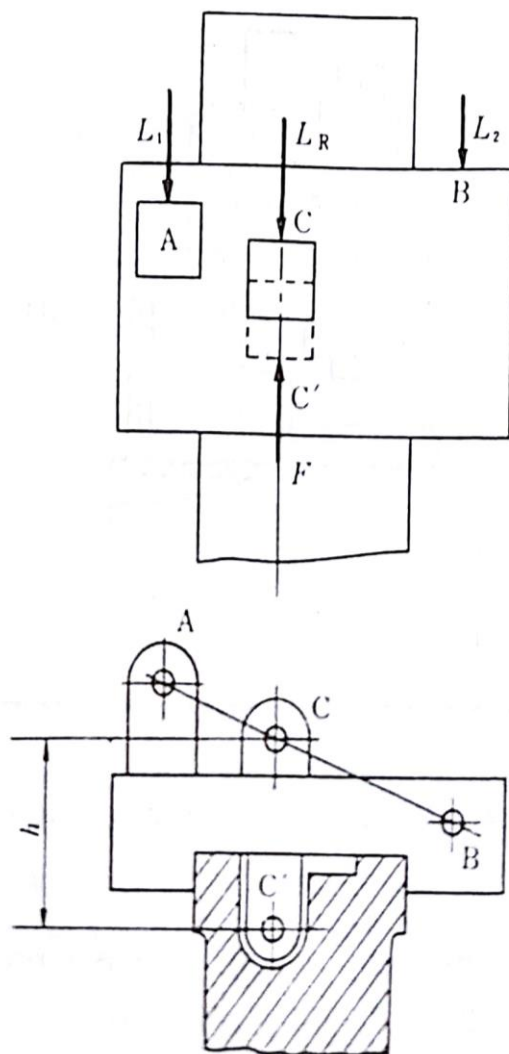
$$\mathbf{r}_G = \sum m_k \mathbf{r}_k / \sum m_k \quad \text{for } k=1 \text{ to } m, \text{ where } m = \text{number of masses}$$

The driving force location C' should be determined from the equilibrium of the resisting forces, L_1 , L_2 , and inertia force L_i

When L_k are the resisting forces (including the inertia force) at \mathbf{r}_k locations, the location of driving force, \mathbf{r}_D is;

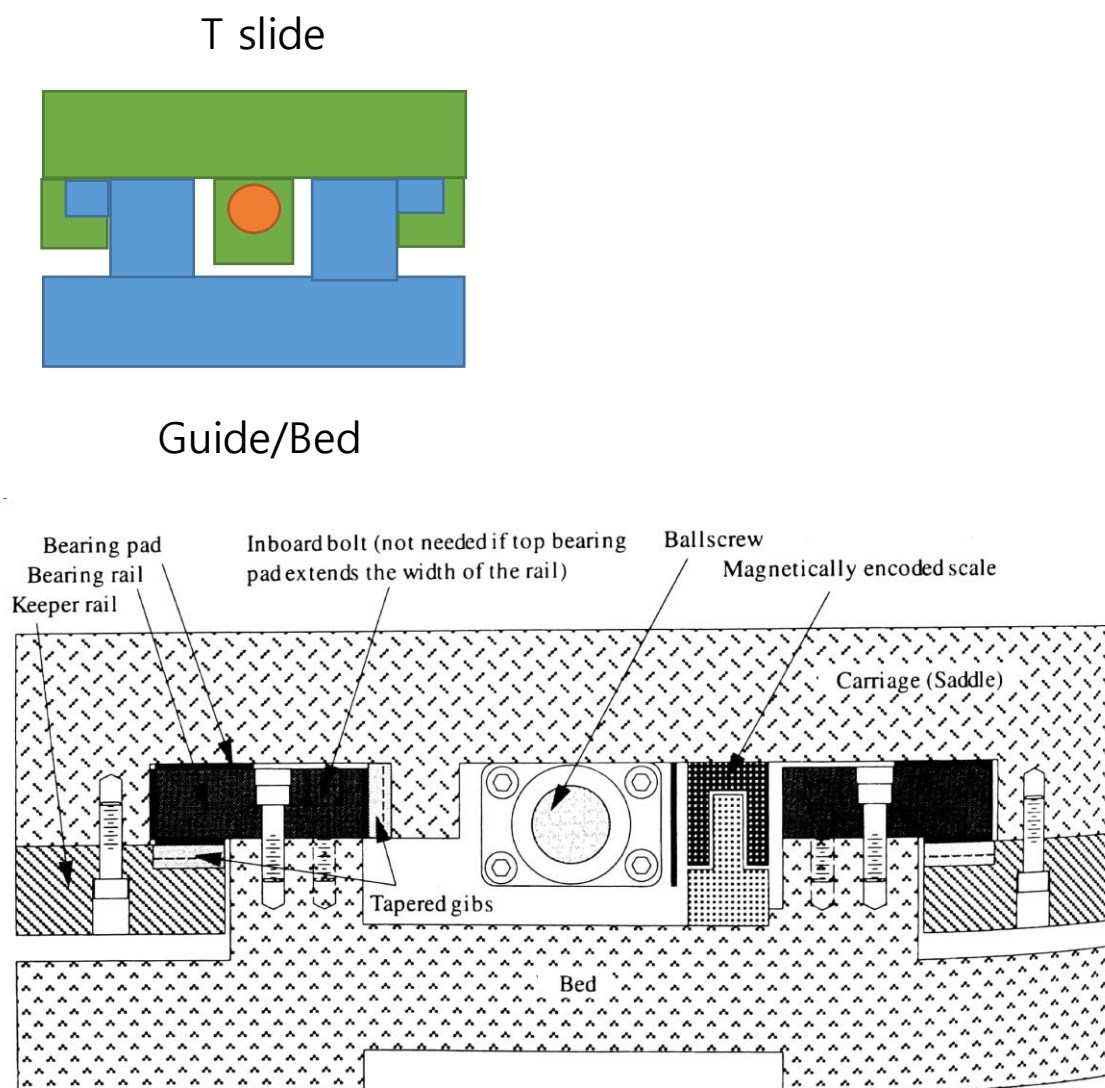
$$\mathbf{r}_D = \sum L_k \mathbf{r}_k / \sum L_k, \text{ for } k=1 \text{ to } n$$

In many practical cases, the resisting forces due to friction are quite small when compared with the inertia force, and thus the driving force location is applied at or close to location of the centre of gravity of the moving slide.



(source: Nakazawa's Principle of precision engineering)

Practical T slide design shows a good example; the driving ball screw is positioned as very close to the mass centre of the T slide just below the T slide location, where the red circle indicates the ball screw driving unit attached. The bottom fig. shows a typical detail design of the T-slide bearing configuration with the ball screw units attached.



T-slide bearing configuration and ball screw attached

(Source: Slocum's Precision Machine Design)

Friction in Bearing Surfaces

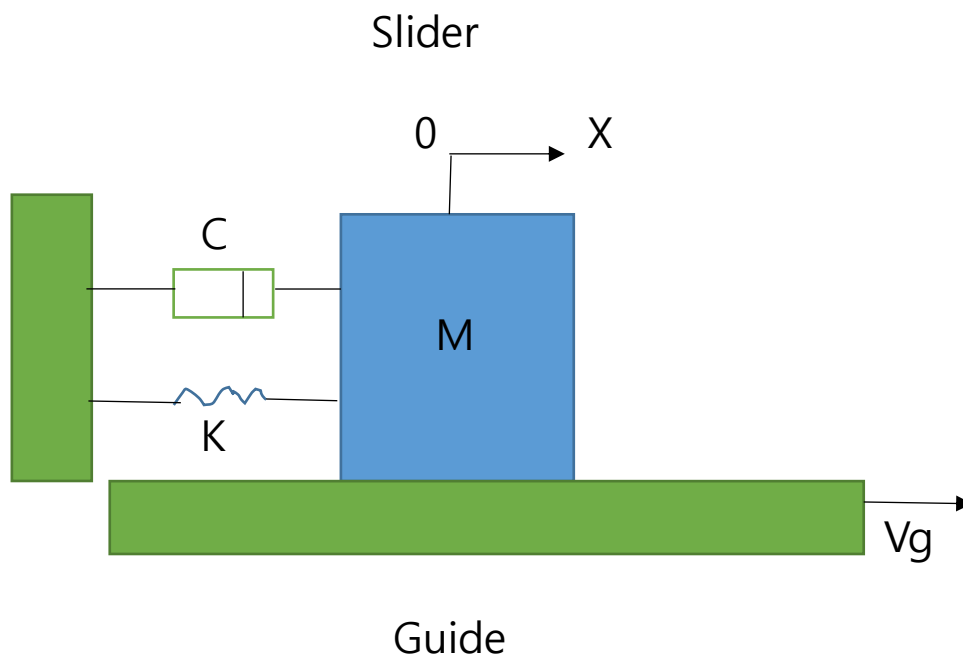
Bearing surfaces play key roles in accuracy and precision of motion, load carrying capacity, lifetime, energy loss, heat generation, etc. The friction between the bearing surfaces are of high engineering importance, such as stick-slip and fluid film friction.

Stick Slip or Stiction ("Dry friction")

Under certain conditions, a slider may experience go and stop, this is called as stick slip, and is due to the frictional characteristics between the sliding surfaces, especially when the static friction force is bigger than the dynamic friction force.

The stick-slip mechanisms can be explained as follows;

The stick-slip phenomena can be explained pretty well with the situation that the slider is pulling by the guide of speed, V_g . The slider mechanism also can be modeled as mechanical system of mass, M ; stiffness, K ; and damping coefficient, C , as in fig.



Initially, mass (M) 'sticks' to the moving guide until the spring force overcomes the static friction force between the mass and guide.

When the spring force overcomes the static friction, the mass starts to move backward ('slip'), and the friction force decreases (dynamic friction), thus the mass continues to move backward, and the spring force is decreased by the backward movement of mass.

When the spring force equals to the dynamic friction force between the mass and the guide, the mass stops. Now the dynamic friction is changed to the static friction, thus the mass 'sticks' to the surface by the static friction force until the backward spring force equals to the static friction force.

These procedures are repeated, thus stick-slip motion is generated

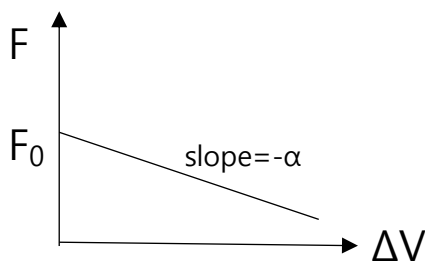
Detail modeling is as follows;

The driving force for the mass is the friction force pulled by the guide, and the friction force is depending on the relative velocity between the guide (V_g) and the mass (dX/dt).

“Dry friction case” (metal to metal contact)

The pulling force by the guide, F , can be modeled as in fig.

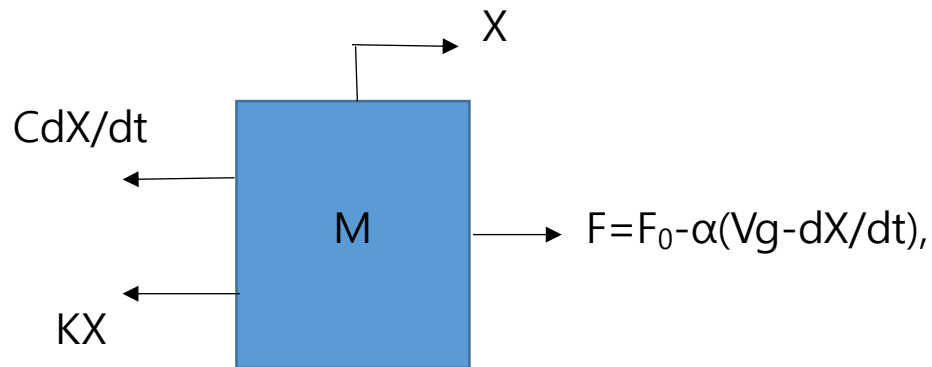
$F = F_0 - \alpha \Delta V = F_0 - \alpha(V_g - dX/dt)$, which indicates the friction force is decreasing from the static friction force (F_0), then proportionally decreases with the relative velocity between surfaces.



Friction force of negative slope with ΔV

F_0 is the static friction force when there is no relative velocity

between the surfaces, $-\alpha$ is the slope while α is the positive constant. This model can be considered as a quite common model for most dry friction conditions experienced at metal to metal contact surfaces.



Motion of equation:

$$M d^2 X / dt^2 = -C dX / dt - KX + F = -C dX / dt - KX + [F_0 - \alpha(Vg - dX / dt)]$$

$$= -(C - \alpha) dX / dt - KX + F_0 - \alpha Vg. \text{ Thus}$$

$$M d^2 X / dt^2 + (C - \alpha) dX / dt + KX = F_0 - \alpha Vg \quad \text{eq(1)}$$

Thus eq(1) gives;

$$X(t) = (F_0 - \alpha Vg) / K + X_0 \exp(-\xi \omega_n t) \cos(\omega_d t - \varphi)$$

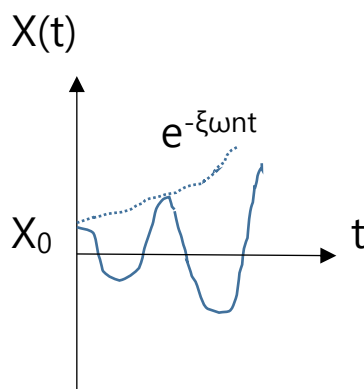
$$\approx X_0 \exp(-\xi \omega_n t) \cos(\omega_d t - \varphi) \text{ after ignoring the offset term;}$$

$$\text{where } 2\xi \omega_n = (C - \alpha) / M, \text{ and } \xi = \text{damping factor}$$

$\omega_n = \sqrt{K/M}$ = natural frequency of system

$\omega_d = \omega_n / \sqrt{1 - \xi^2}$ = damped natural frequency

Constants X_0 , and φ (phase delay) are determined from the initial displacement of slider such that $X(0) = X_0 \cos \varphi$



Negative damping case

When $C - \alpha < 0$, this is the case of negative damping, and a small disturbance is exponentially growing as time increases, thus unstable or uncontrollable situation.

When $C - \alpha \geq 0$, this case is the positive or zero damping, and the damping capacity is greatly reduced from C to $C - \alpha$, thus the disturbance lasts for a longtime until it is fully damped out.

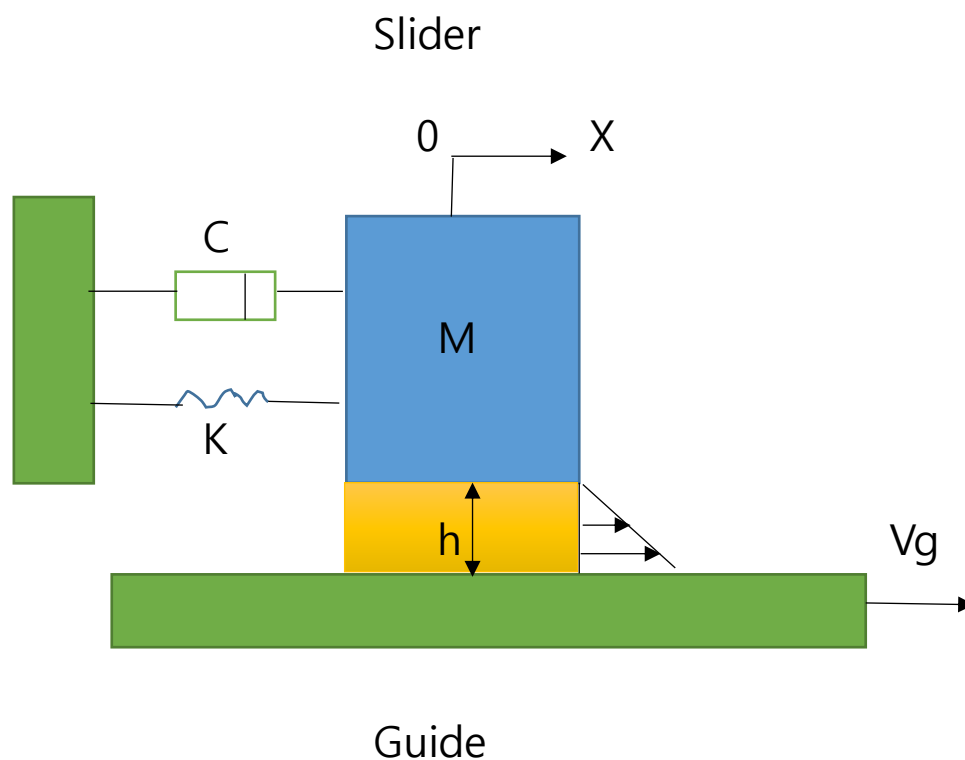
Practically, the stick-slip is repeating while the guide is moving continuously, and this is the reason that the resolution (or

resolving capability) of slider movement cannot be smaller during the stick-slip mode due to friction between the contact surfaces.

This situation can be avoided by introduction of fluid film between the contact surfaces.

Fluid Film friction("Wet Friction")

When the fluid film of thickness, h , is introduced between the slider and guide, the velocity distribution of fluid across the film thickness can be assumed as linear as in the fig.



Velocity gradient across the film

$$=dV/dh=V/h=(Vg-dX/dt)/h$$

The shear stress, τ , across the film is proportional to the velocity gradient by the Newton's equation of fluid;

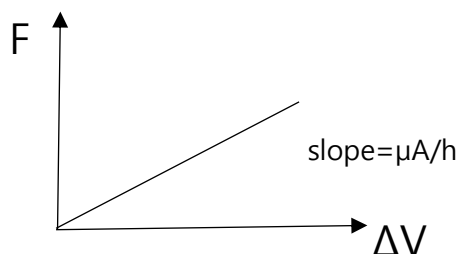
$$\tau=\mu dV/dh=\mu V/h=\mu(Vg-dX/dt)/h$$

where μ =viscosity of fluid

The pulling force by the guide, F , is then

$$F=\tau A=\mu AV/h=\mu A(Vg-dX/dt)/h$$

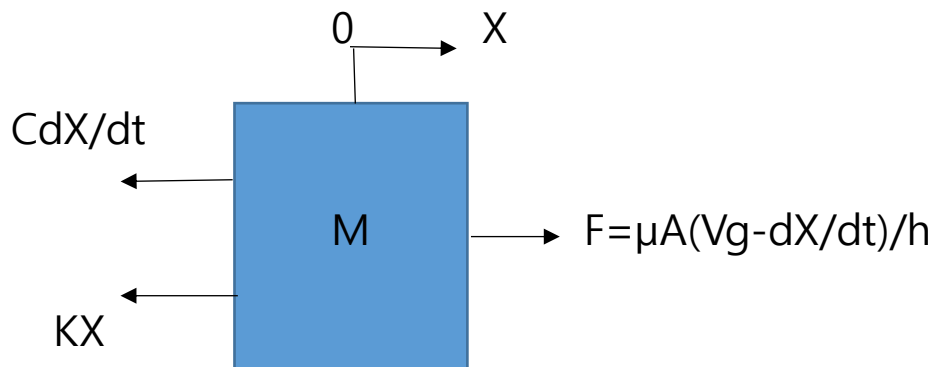
where A is the surface area of contact.



Friction force of positive slope with ΔV

Comparing the friction force models, it is to be noted that the friction force model is quite different such as the negative slope for the stick-slip friction, and positive slope for the fluid film friction or wet friction.

Motion of equation:



$$M d^2X/dt^2 = -KX - CdX/dt + F$$

$$= -KX - CdX/dt + \mu A(Vg - dX/dt)/h.$$

Thus

$$M d^2X/dt^2 + (C + \mu A/h) dX/dt + KX = \mu A V_s/h \quad \text{eq(2)}$$

$$X(t) = \mu A Vg/Kh + X_0 \exp(-\xi \omega_n t) \cos(\omega_d t - \varphi)$$

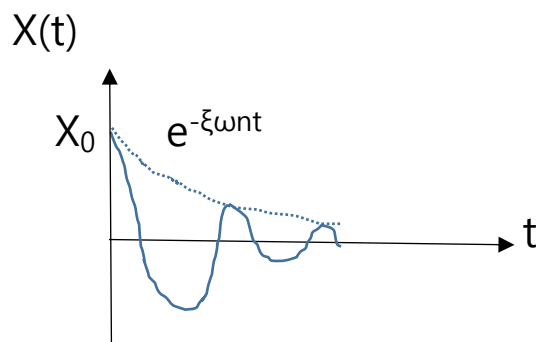
$\approx X_0 \exp(-\xi \omega_n t) \cos(\omega_d t - \varphi)$, after ignoring the offset term;

where $2\xi\omega_n = [C + \mu A/h]/M$, and ξ = damping factor

$\omega_n = \sqrt{K/M}$ = natural frequency of system

$\omega_d = \omega_n / \sqrt{1 - \xi^2}$ = damped natural frequency

Constants X_0 , and φ (phase delay) are determined by the initial displacement of slider such that $X(0) = X_0 \cos \varphi$



Positive damping

This is the case of the positive damping, and the damping capacity increases further from C to $C+\mu A/h$, thus the disturbance is damped out more quickly. Thus it is stable, controllable, and very fine resolution is achievable, that is, the desirable situation for the precision motion.

The friction model of positive slope with the relative velocity is the preferable characteristic for the stable, controllable situation for the precision linear motion.

Bearing surfaces with fluid film are good and typical applications; such as Hydrostatic bearing, Aerostatic bearing, Teflon, and Turcite, etc.

Further information is;

Logarithmic decrement, δ

=Natural log of the ratio of the amplitudes of any two successive peaks

$$=(1/n)\text{Log}_e[X(t)/X(t+nT)]=\text{Log}_e[X(t)/X(t+T)]=\text{Log}_e[X_0/X_1]$$

where n is number of period between the two peaks

T is the period of system $=2\pi/\omega_n$

$$\text{Damping factor}(\xi)=1/\sqrt{1+(2\pi/\delta)^2}=\delta/\sqrt{\delta^2+(2\pi)^2}\doteq\delta/2\pi$$