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Modeling of Mold Oscillation

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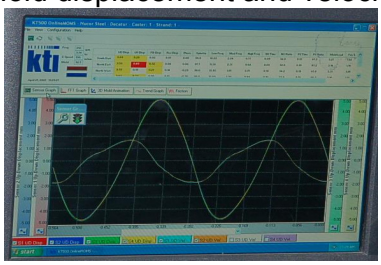
Mold Oscillation System at NUCOR

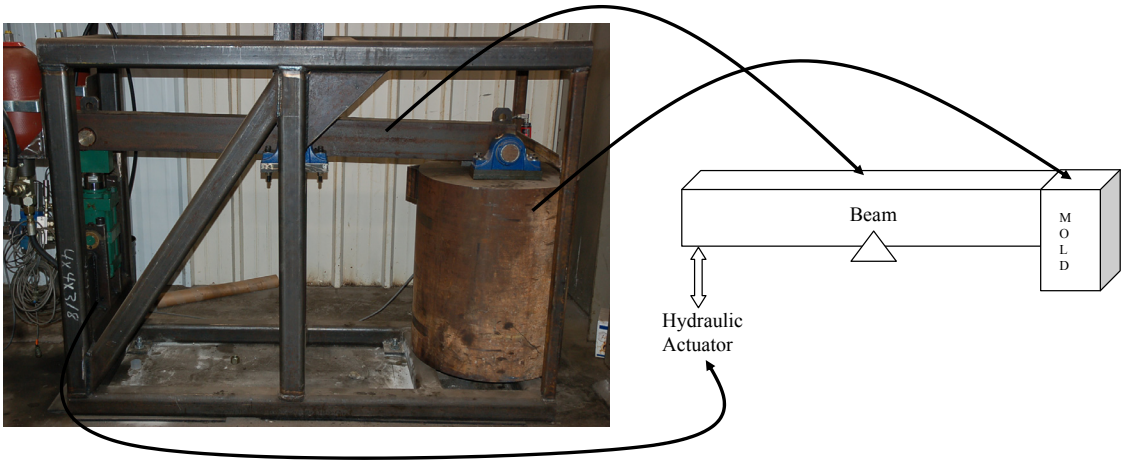
Mold displacement and velocity



Problem:
1. Resonance mode of primary beam gets excited when the actuator oscillates at ***one-third*** the resonance frequency
2. This unwanted resonance distorts the mold displacement and velocity profile

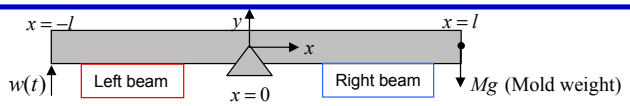
Objective: Model this mold oscillation system, simulate it, identify the source of disturbance and control it

Simplified mock-up



- Mock-up captures similar resonance problem
  - 1) resonant frequency = 9.6Hz
  - 2) input at 4.8Hz excites 9.6Hz
- Beam is modeled using Timoshenko beam model
- Heavy Mold, significant dynamics accounted for in Boundary conditions
- Hydraulic actuator – Nonlinear behavior (same model as plant)
- Beam and actuator coupled using boundary condition

Timoshenko Beam Model



1D dynamic beam transverse displacement PDE

$$\frac{\partial}{\partial x}\left(k'GA\left(\frac{\partial y_L}{\partial x}-\psi_L\right)\right)-mg-\gamma_L\frac{\partial y_L}{\partial t}-m\frac{\partial^2 y_L}{\partial t^2}=0$$

$$\frac{\partial}{\partial x}\left(k'GA\left(\frac{\partial y_R}{\partial x}-\psi_R\right)\right)-mg-\gamma_R\frac{\partial y_R}{\partial t}-m\frac{\partial^2 y_R}{\partial t^2}=0$$

1D dynamic beam bending angle PDE

$$\frac{\partial}{\partial x}\left(EI\frac{\partial \psi_L}{\partial x}\right)+k'GA\left(\frac{\partial y_L}{\partial x}-\psi_L\right)-\gamma_L\frac{\partial \psi_L}{\partial t}-k^2m\frac{\partial^2 \psi_L}{\partial t^2}=0$$

$$\frac{\partial}{\partial x}\left(EI\frac{\partial \psi_R}{\partial x}\right)+k'GA\left(\frac{\partial y_R}{\partial x}-\psi_R\right)-\gamma_R\frac{\partial \psi_R}{\partial t}-k^2m\frac{\partial^2 \psi_R}{\partial t^2}=0$$

Boundary conditions

$$y_L(-l)=w(t)\quad EI\frac{\partial \psi_L(-l)}{\partial x}=0$$

$$y_L(0)=0\quad EI\frac{\partial \psi_L(0)}{\partial x}=M\quad \psi_L(0)=S$$

$$y_R(0)=0\quad EI\frac{\partial \psi_R(0)}{\partial x}=M\quad \psi_R(0)=S$$

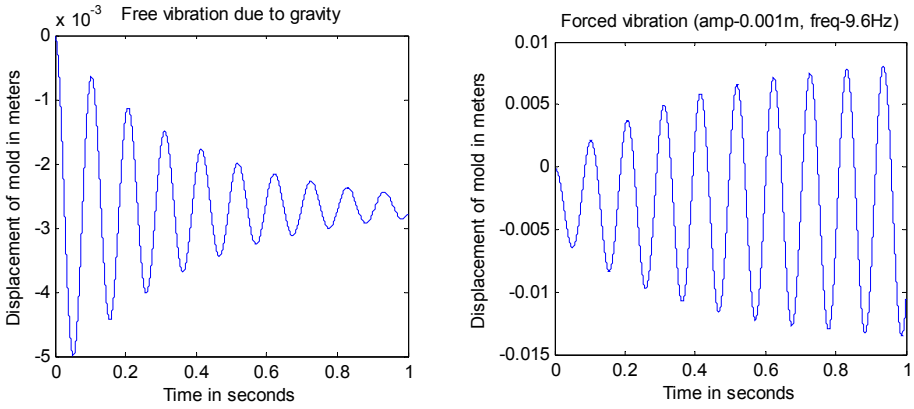
$$k'GA\left(\frac{\partial y_R(l)}{\partial x}-\psi_R(l)\right)+Mg+M\ddot{y}_R(l)+\gamma\frac{\partial y_R(l)}{\partial t}=0\quad EI\frac{\partial \psi_R(l)}{\partial x}=0$$

Mock-up parameters

$E=200Gpa$ Youngs modulus	$G=82GPa$ Shear Modulus for steel
ρ=7870 Kg/m^3 - density of steel	$Area=0.0088m^2$ cross section area of beam
$I=2.2\times10^{-5}m^4$ Moment of inertia of beam	$m=69Kg/m$ Mass per unit length of beam
M=2250 Kgs	$k'=0.83$ Shear constant
Beam width = 5.13'(hollow with thickness 0.94')	$l=34.5'$
Beam breadth = 6' (hollow with thickness 0.38')	

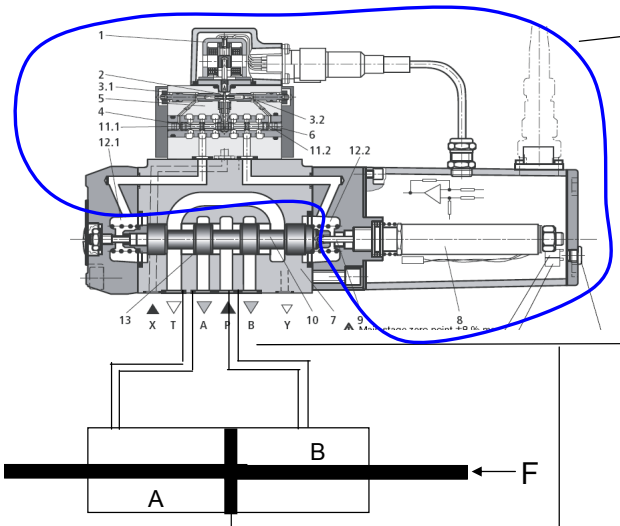
Solve these 4 Partial differential equations simultaneously

Simulated displacement histories



- Simulation of the four partial differential equations with the actual parameters indicates the beam resonance frequency is 10.6 Hz
- Simulation after changing moment of inertia slightly (factor of 1.15) gives a resonance frequency of 9.6Hz
- Experimental resonance frequency is 9.6Hz

Hydraulic Actuator



Electronic control of spool position
Fast dynamics - Hence not modelled in simulations

Turbulent Flow Equations for flow in A and B

$$q_A = \begin{cases} c(d-x_s)\sqrt{P_s-P_A} & x_s < -d \\ c(d-x_s)\sqrt{P_s-P_A} - c(x_s+d)\sqrt{P_A-P_t} & -d < x_s < d \\ -c(x_s+d)\sqrt{P_A-P_t} & x_s > d \end{cases}$$

$$q_B = \begin{cases} c(d-x_s)\sqrt{P_B-P_t} & x_s < -d \\ c(d-x_s)\sqrt{P_B-P_t} - c(x_s+d)\sqrt{P_s-P_B} & -d < x_s < d \\ -c(x_s+d)\sqrt{P_s-P_B} & x_s > d \end{cases}$$

x_s is positive when the spool moves to the right. Its mean position is 0 with valve underlap gap of d on both sides.
q_A is positive when oil flows in to chamber A
q_B is positive when oil flows out of chamber B
q, P – Volume flow rate and Pressure
$\alpha_{A,B,s,t}$ – Chamber connected to A and B, source, tank

Pressure equations

$$\dot{P}_A = \frac{\beta}{(V_A + A(L+w))}(q_A - A\dot{w})$$

$$\dot{P}_B = \frac{\beta}{(V_B + A(L-w))}(-q_B + A\dot{w})$$

w – positive if it moves to right of midpoint of cylinder
V_A, V_B – Static volume of chambers A and B
$A(L+w)$ – dynamic volume of chamber A and B
L – Half the piston stroke length
A – Surface area of piston

Actuator dynamic equations

$$m\ddot{w} = (P_A - P_B)A + F - (fric)\dot{w}$$

w - Actuator displacement

Set of Nonlinear Differential Equations

Hydraulic Actuator contd.

Force acting on actuator due to beam

$$F = -k'GA\left(\frac{\partial y_L}{\partial x} - \psi_L\right)_{x=-l}$$

Control law that determines spool position

Control law

$$x_s = f\left(w, w_{des}\right) = k_p\left(w - w_{des}\right) + k_I \int_0^t \left(w - w_{des}\right) dt$$

State space form of actuator equations

$$\dot{x}_1 = x_2$$

$$\dot{x}_2 = -(fric)x_2 + (x_3 - x_4)\frac{A}{m} - k'GA\left(\frac{\partial u_L}{\partial x} - \psi_L\right)_{x=-l}$$

$$\dot{x}_3 = \frac{\beta}{\left(V_A + A(L + x_1)\right)}\left(q_A - Ax_2\right)$$

$$\dot{x}_4 = \frac{\beta}{\left(V_B + A(L - x_1)\right)}\left(-q_B + Ax_2\right)$$

$$q_A = \begin{cases} c(d - x_s)\sqrt{P_s - x_3} & x_s < -d \\ c(d - x_s)\sqrt{P_s - x_3} - c(x_s + d)\sqrt{x_3 - P_t} & -d < x_s < d \\ -c(x_s + d)\sqrt{x_3 - P_t} & x_s > d \end{cases}$$

$$q_B = \begin{cases} c(d - x_s)\sqrt{x_4 - P_t} & x_s < -d \\ c(d - x_s)\sqrt{x_4 - P_t} - c(x_s + d)\sqrt{P_s - x_4} & -d < x_s < d \\ -c(x_s + d)\sqrt{P_s - x_4} & x_s > d \end{cases}$$

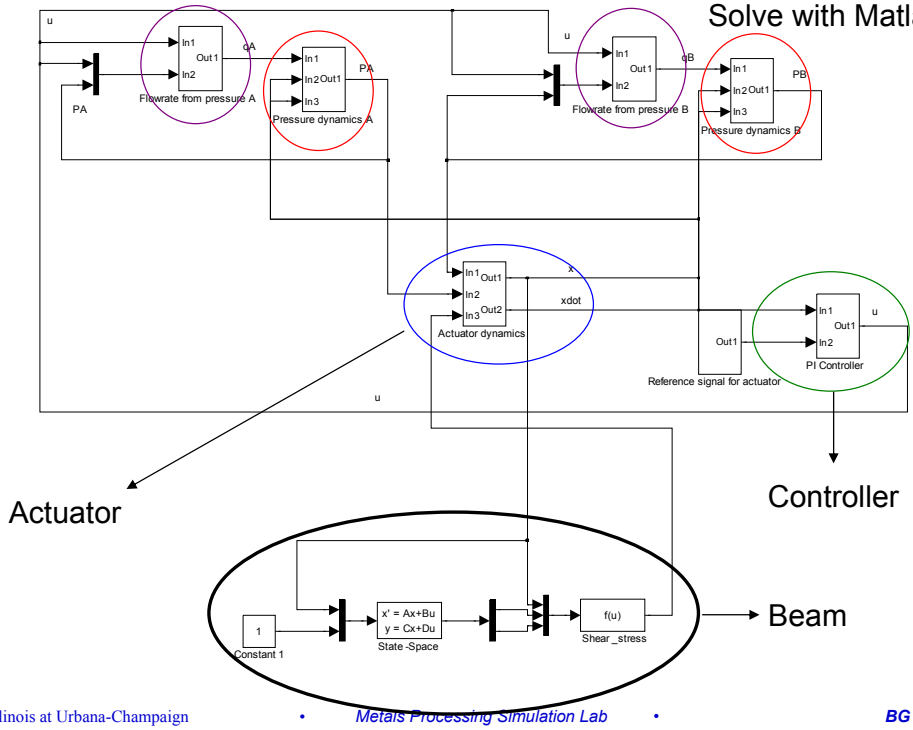
$$x_s = u$$

Parameters for mock-up

$d = 1.27\times10^{-6}\,m$
$A = 0.0046m^2$
$P_s = 3000Psi$
$P_t = 30Psi$
$c = 3\times10^{-4}$
$V_A = 2.875inch^3$
$V_B = 4.3inch^3$
$m = 8Kgs$
$\beta = 1500Mpa$
$L = 0.015m$
$fric = 1000N.s/m$

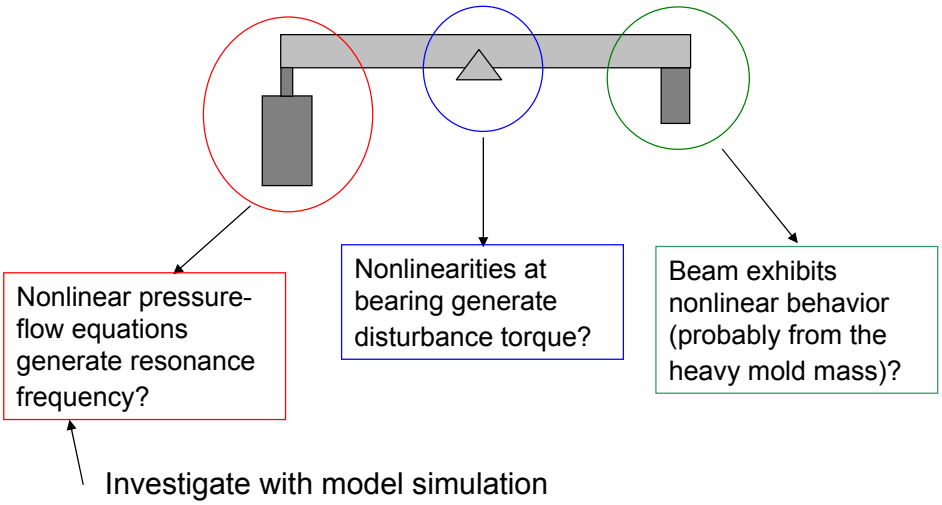
Flow diagram for coupled actuator-beam control simulation

Solve with Matlab Simulink

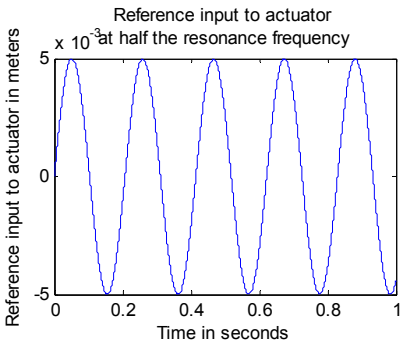


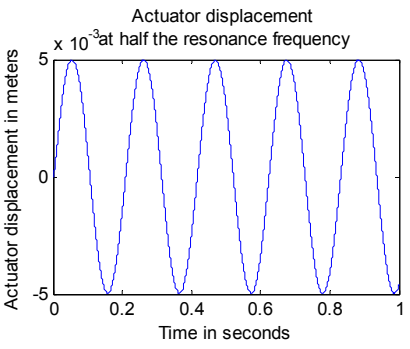
Possible causes of problem

When actuator is given a sinusoidal reference input at a frequency half the resonance frequency, the resonance frequency of beam gets excited.

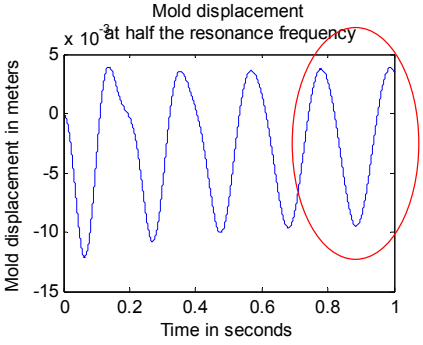


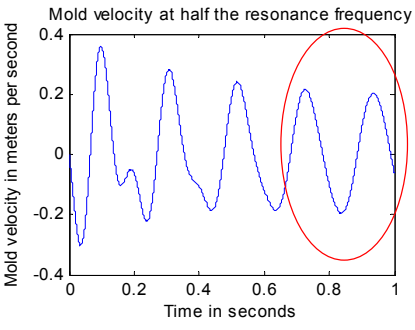
Mockup Simulation with well-tuned controller: Is nonlinear pressure-flow equation the cause of problem?





- Careful tuning of PI controller (k_P=1.6, k_I=0.2) ensures that the actuator output has small second harmonic content



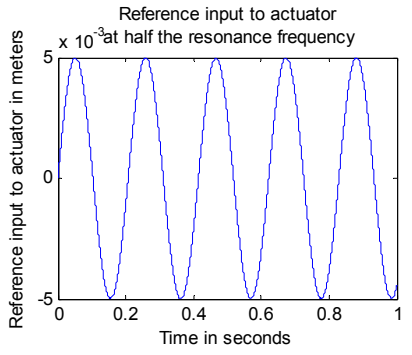


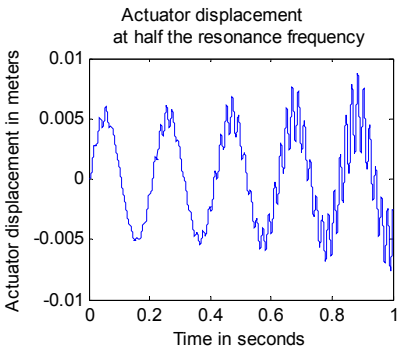
- Small distortions exhibited by mold displacement and velocity profiles
- Distortions less than those noticed in experiments
- Nonlinear pressure-flow equations may **not** be the source of trouble if controller is properly tuned.

Other sources of resonance frequency

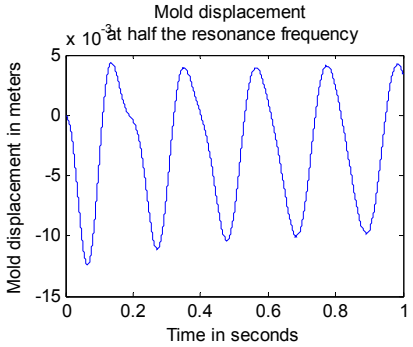
- Simulation indicates bearing friction torque is probably not the source of disturbance.
- Other types of bearing vibrations have not yet been considered.
- In case beam dynamics is identified to be the source of problem, a more complicated higher dimension beam model will be considered.
- Earlier reports on this problem by other groups indicate that the actuator is the source of resonance harmonic and the slow spool update rate could be a reason for this.
- Hence other unmodeled dynamics and nonlinearities not inherent in the actuator need to be considered

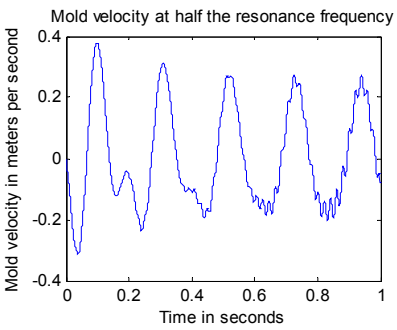
Mockup Simulation with delay in spool input





- With a delay of 0.005s introduced between the controller and the actuator, the actuator displacement starts exhibiting higher frequencies





- These frequencies give mild distortions in the mold displacement and get magnified in the velocity
- Hence similar unmodeled dynamics in the actuator could be the source of higher harmonics

Conclusions and future work

- Simulations suggest that nonlinear pressure-flow behavior of the actuator does not explain the resonance frequency.
- Additional nonlinearities and unmodeled dynamics such as delay might be responsible.
- Future experiments will be performed using the mockup to quantify these nonlinearities by measuring actuator pressure, & displacement and velocity at various points in the beam
- Improving controller design depends on the source of the disturbance.

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