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Modeling and Control of Mold Oscillation

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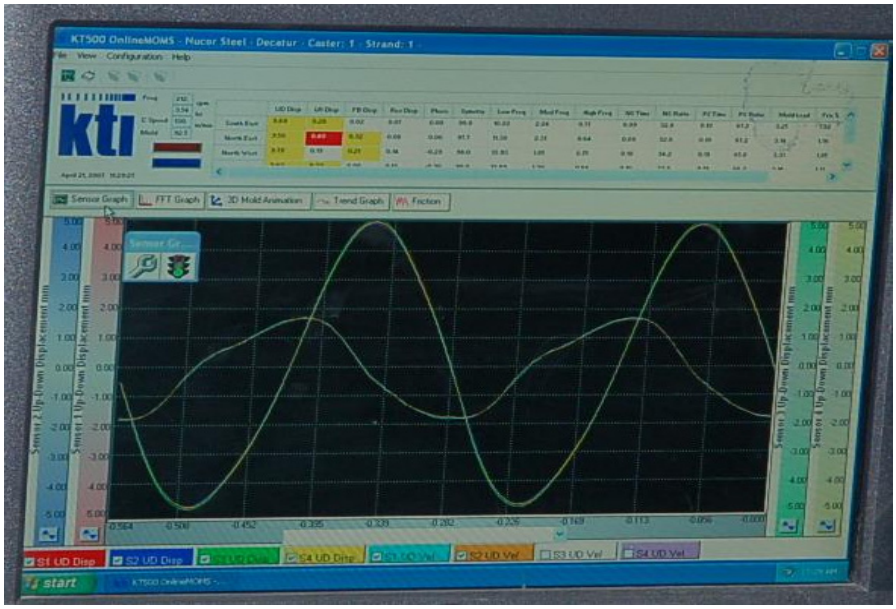
Mold oscillation system at NUCOR

System Objective: Provide vertical oscillatory mold motion characterized by distortion-free pure sinusoidal mold displacement and velocity profiles

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

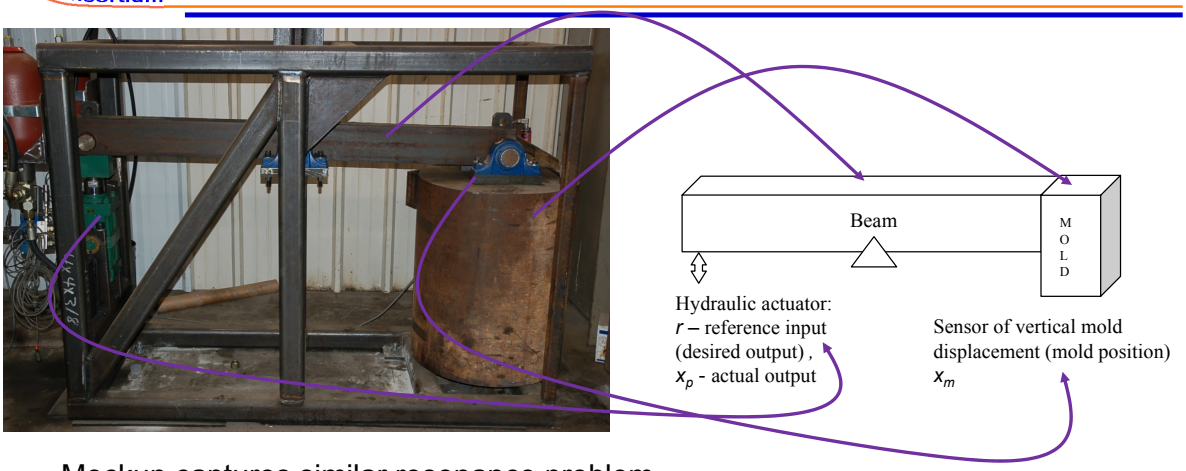
Project Objective: Model this mold oscillation system, simulate the problem, identify the source of disturbance, and control it

Screenshot of mold displacement and velocity profile (note distortions in velocity)



GOAL – Eliminate the distortions in velocity profile

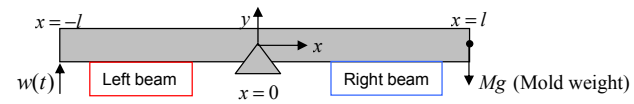
Labscale simplified mockup of oscillator



- Mockup captures similar resonance problem
  - 1) resonant frequency = 9.2Hz
  - 2) input at 4.6Hz excites 9.2Hz
- Hydraulic valve/actuator – Nonlinear behavior (same model as plant)

Note: Focus on position signals, since a distortion free pure sinusoidal position signal also guarantees a distortion free velocity signal.

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

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

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

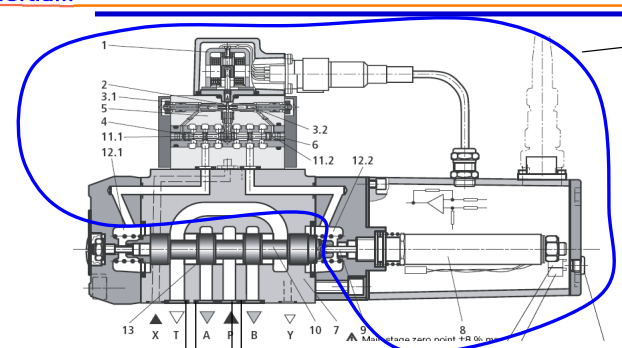
$$y_R(0)=0\qquad EI\frac{\partial \psi_R(0)}{\partial x}=M\qquad \psi_R(0)=S$$

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

Mockup 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
$I=1.7668\times10^{-5}m^4$ 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')	
Damping coefficients $\gamma_L=\gamma_R=10$	$\gamma=2\text{Kg/sec}$

Hydraulic servo valve and actuator model



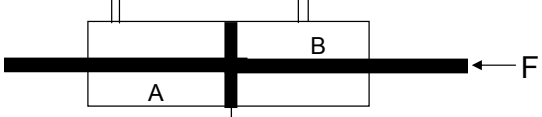
Electronic control of spool position
Fast dynamics - Hence not modelled in simulations

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



Actuator dynamic equations
$m\ddot{x}=(P_A-P_B)A+F-(fric)\dot{x}$

Pressure equations

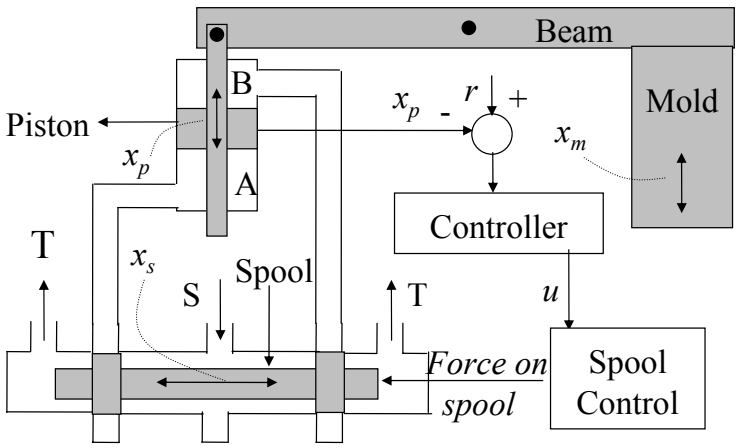
$$\dot{P}_A=\frac{\beta}{(V_A+A(L+x))}(q_A-A\dot{x})$$

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

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

Set of Nonlinear Differential Equations

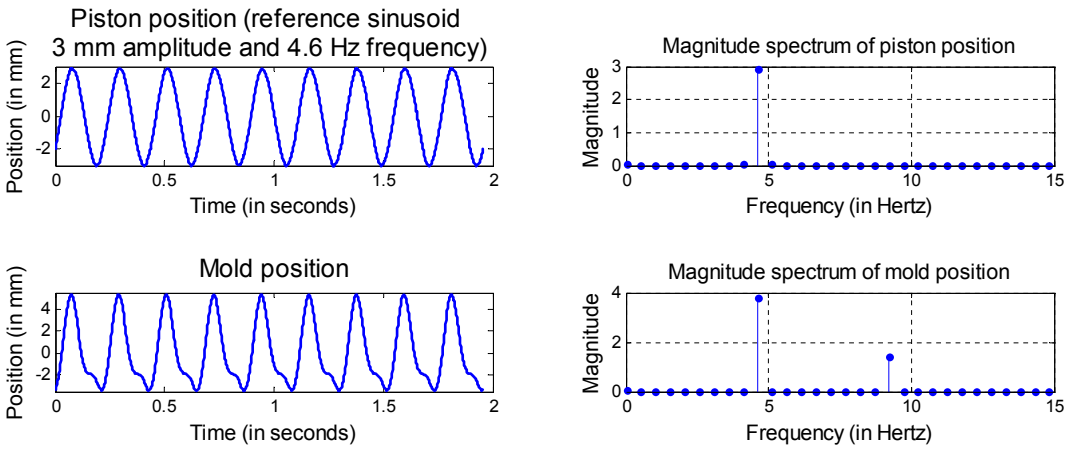
Working of the servo-beam structure



Important signals:
x_p - Actual piston position; x_m - Mold position
r – Desired piston position (reference)

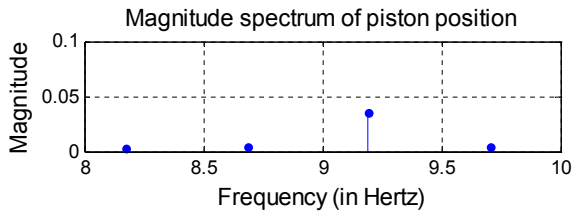
- Spool moves to left – S (pressure supply) connected to chamber A; T (tank) connected to chamber B; piston pushed up
- Spool moves to right – S (pressure supply) connected to chamber B; T (tank) connected to chamber A; piston pushed down
- Controller uses error between r (desired piston position) and x_P (actual piston position) to generate control signal u that moves spool suitably to get desired piston motion
- Typically proportional controller is used. $u=K(r-x_p)$

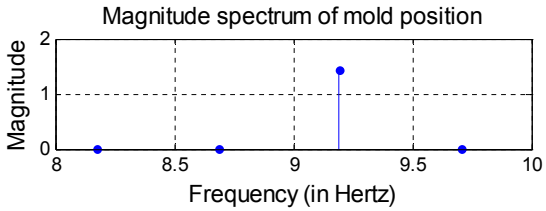
Experimental data from mockup exhibiting resonance problem



- The desired position or "reference" for the actuator is chosen to be a sine wave of frequency 4.6 Hz (half the resonance frequency) and amplitude 3 mm
- P controller with gain $K=2$ is used
- Piston position profile looks ok, but mold position profile is distorted
- Piston position apparently has no harmonics while mold position has
- Electrohydraulic servo - seems to perform ideally

Source of distortion

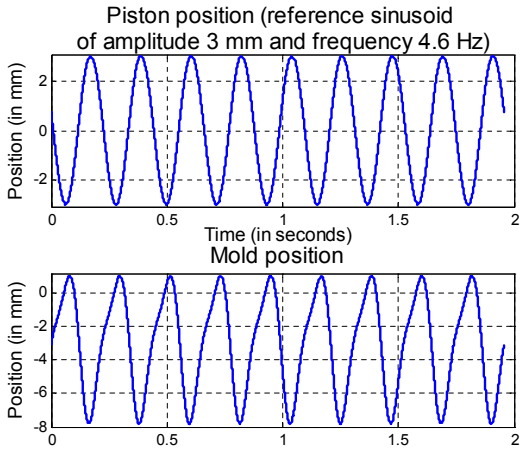


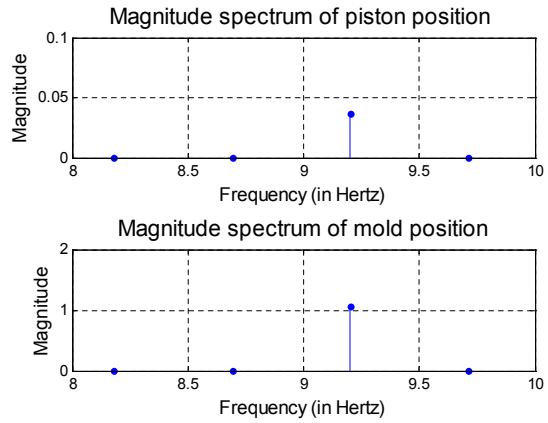


- But zooming into the magnitude spectrum of the piston position reveals a small peak at 9.2 Hz
- Caused by nonlinear servo dynamics
- Small peak is amplified by about 40 times to cause the large peak at the mold end
- This matches well with the resonance excitation experiment in which a sinusoidal piston position amplitude of 0.05 mm at 9.2 Hz is amplified by the beam by a factor of 30 at the mold end

Major source of distortion – peak due to nonlinear servo

Simulation results



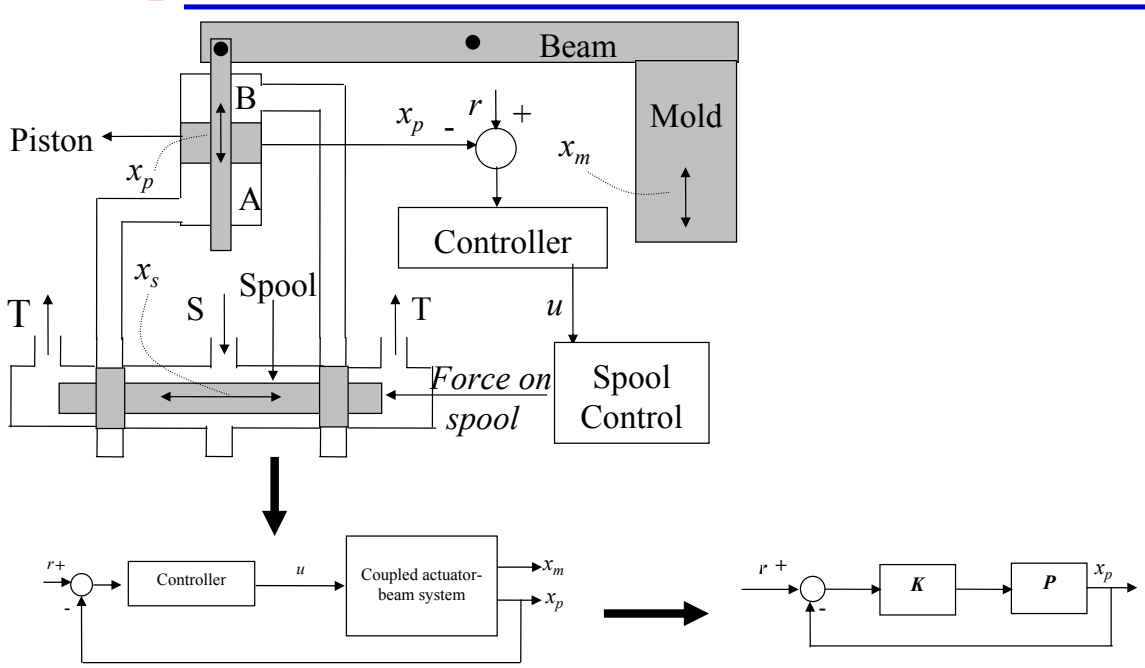


- Simulation of servo-beam model using nominal parameters, same reference
- Like the mockup, predicted piston position looks ok but has small peak at 9.2 Hz in magnitude spectrum generated by the nonlinear servo-beam model
- Again, like the mockup, the small peak is amplified by beam causing significant distortion in the mold position

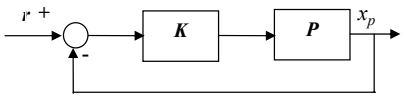
Problem statement

Design *a* filtering technique that removes the small-amplitude harmonic near 9.2 Hz from the piston position without affecting the already achieved good tracking at other desired frequencies.

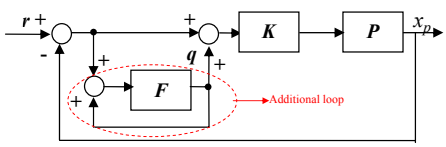
Block diagram representation of the pre-existing stable servo system



Solution for mockup using Internal model based controller (IMC)



x_p tracks *r* well, but has a small undesired sinusoidal component at frequency w

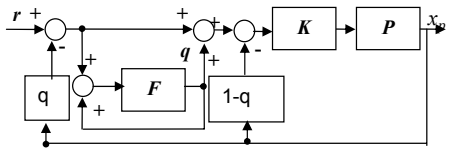


Augmenting a closed loop with a filter *F* as shown ensures that x_p continues tracking *r* well and has no undesired sinusoid at frequency w

Transfer function of the filter	$F(s) = \dfrac{2ews}{s^2 + 2ews + w^2}$	$0 < e \ll 1$ (damping coefficient) s - Laplace variable w - frequency to be removed

Generalizing the IMC solution

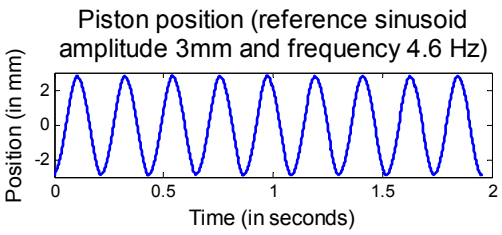
- The additional component *F* is a simple internal model based filter. It internally generates a sinusoid at frequency 'w' to cancel the unwanted sinusoid of the same frequency
- Before augmenting the closed loop with the filter *F*, the response of the servo at frequency 'w' must be measured and this measured quantity must satisfy a technical condition
- In the mockup and the simulations, this condition can be satisfied
- In case the condition cannot be satisfied (for example on the caster), we have developed modified loop topology to ensure that this approach can still be used:

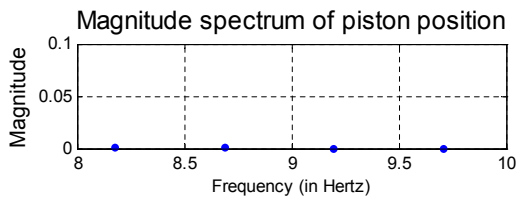


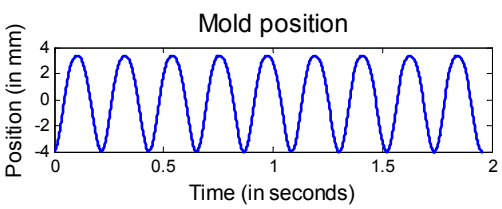
V. Natarajan and J. Bentsman, '*Robust Rejection of Sinusoids in Stable Nonlinearly Perturbed Unmodelled Linear Systems: Theory and Application to Servo*', Proceedings of the American Control Conference 2011, San Fransisco, pp. 3289-3294.

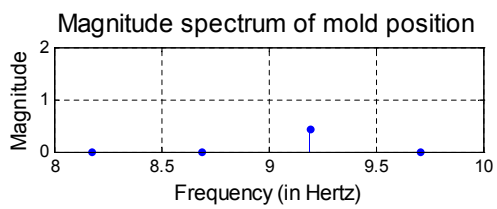
Validation of IMC on the mockup

- Proposed IMC is applied to mockup to remove harmonic at 9.2 Hz from the piston position
- Filter *F* introduced with e=0.1 and w=2π X 9.2 rad/sec
- Controller *K*=2, large enough for gain condition to be satisfied
- Reference sinusoid – amplitude 3 mm and frequency 4.6 Hz







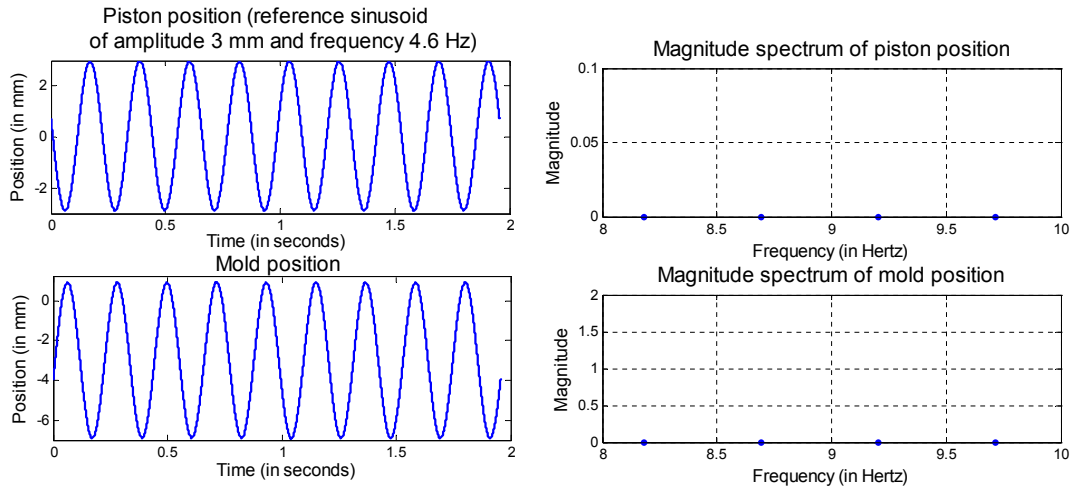


Observations

- The distortions in the mold position are significantly reduced
- The small peak at 9.2 Hz is eliminated from the magnitude spectrum of the piston position signal
- As a consequence the corresponding peak in the mold position reduced from 1.45 mm to 0.45 mm
- The residual peak of 0.45 mm caused by other structural non-idealities including contact at hinges that are ignored, but can be considered if required
- Major distortion removed
- The satisfactory tracking at 4.6 Hz is preserved

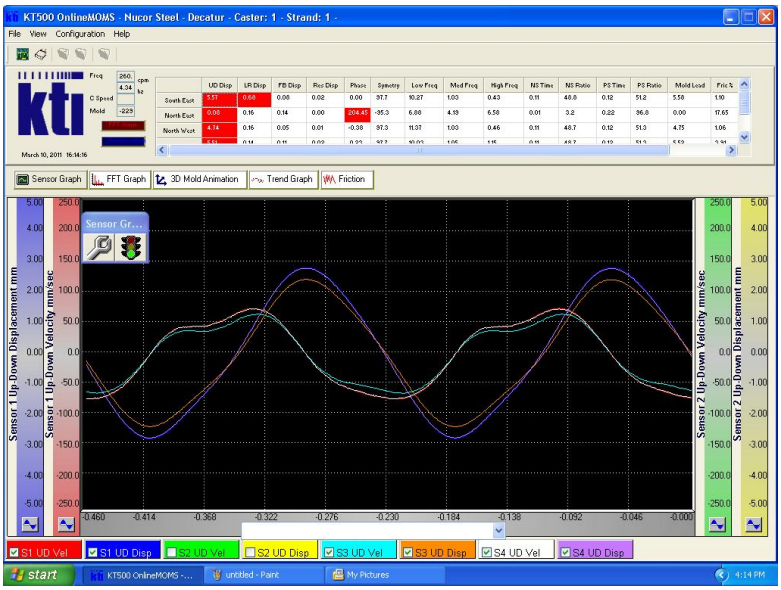
Validation of IMC on the computational model of the mockup

- Proposed IMC is applied to servo-beam model
- Filter *F* introduced with e=0.1 and w=2π X 9.2
- Controller *K*=0.6 and reference sinusoid – amplitude 3 mm and frequency 4.6 Hz
- Distortions in mold position profile completely eliminated



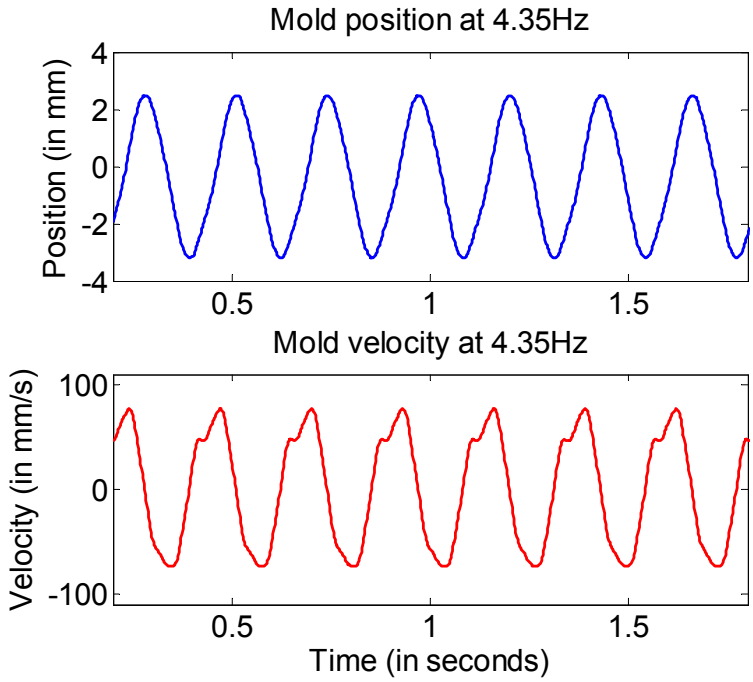
Experiments at the caster

- Experiments similar to those performed on the mockup are repeated at the caster
- The desired sinusoidal motion of the mold is input to the hydraulic actuator
- At certain frequencies, it is noted that the mold velocity profile is distorted
- In all the experimental data shown, the frequency of the desired sinusoid is 4.35 Hz and amplitude is 3 mm

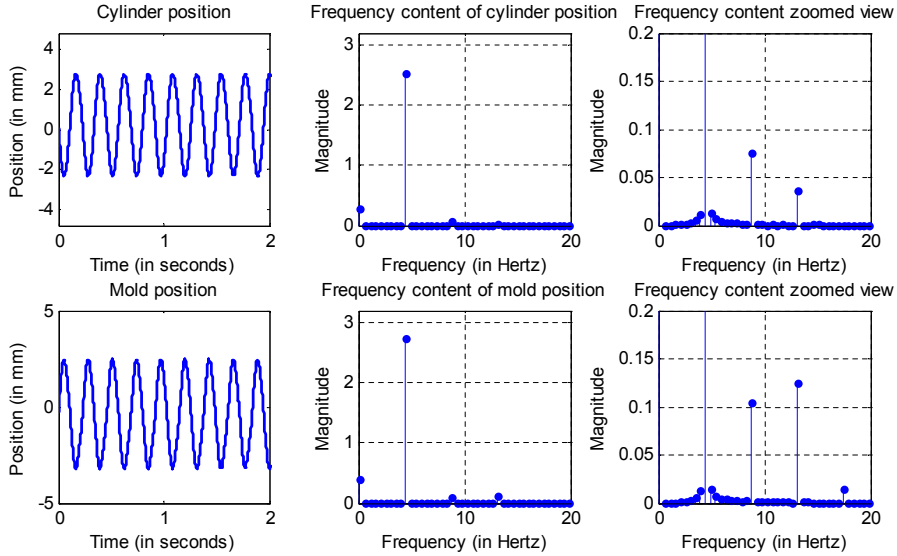


Screenshot of Mold data from accelerometers

Mold data from position sensors (velocity data obtained via differentiation and averaging)



Position data without IMC

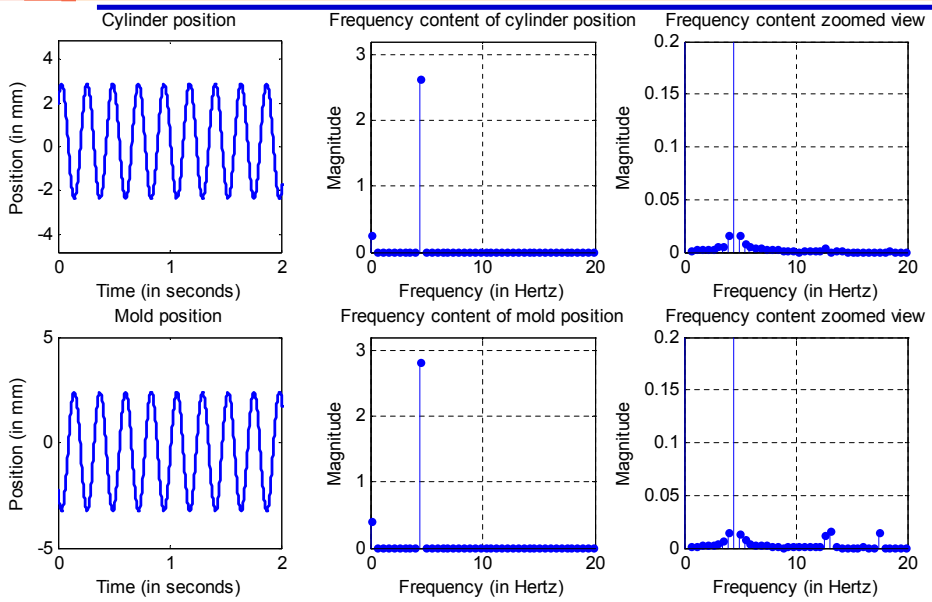


•In the frequency content plots peaks are located at 4.35 Hz, 8.7 Hz and 13.05 Hz while the resonance is at about 12.5 Hz

•So we are not looking at a purely resonance problem

•We need to remove the peaks at both 8.7 Hz and 13.05 Hz

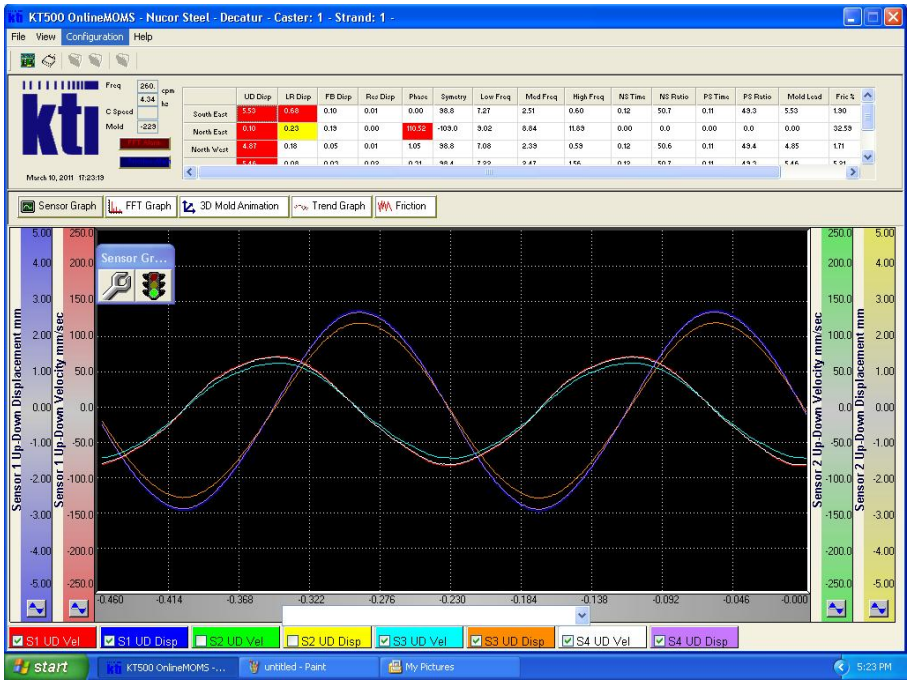
Position data using IMC



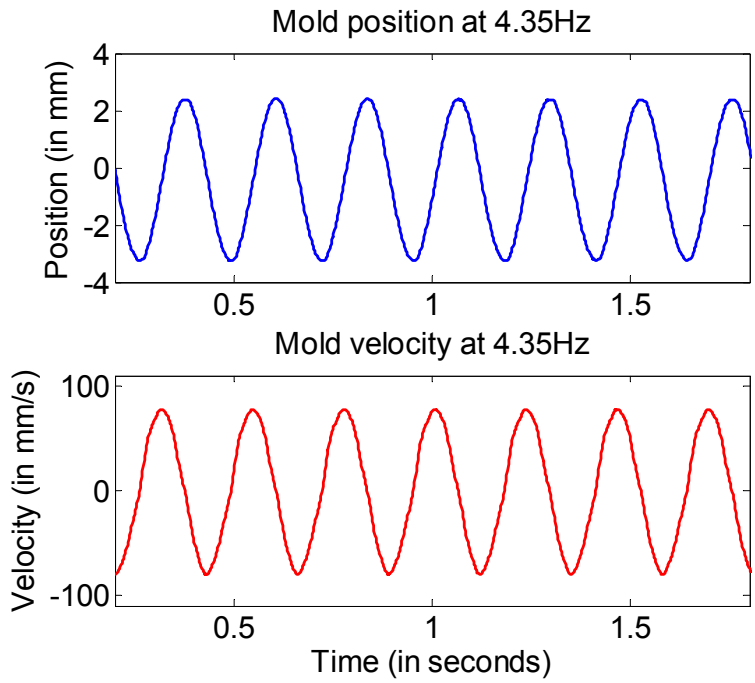
In the frequency content plots peaks near 8.7 Hz and 13.05 Hz are removed in the piston position signal thereby improving the mold position and velocity profiles

We do this by implementing one filter for each frequency and using the modified loop topology (not necessary at the mockup)

Screenshot of Mold data from accelerometers with IMC



Mold data from position sensors using IMC



Conclusions

- The cause of distortions in the mold position signal in the mockup was identified to be resonant amplification of nonlinear servo disturbance
- A filtering procedure was developed to solve this problem
- The procedure was successfully tested on a computational model and with experiments on the mockup
- Experiments at the caster confirmed that the major source of distortion in the mockup and the caster are the same
- The filtering approach was demonstrated successfully at the caster and the distortion in the velocity profile at a fixed frequency (4.35 Hz) was eliminated
- The software code was modified so that when the frequencies change by small amounts (4.35 +/- 0.05 Hz), the velocity profiles remain undistorted

Future work

- Code development to ensure distortion free velocity profile when the frequency is changed by large amounts has been completed and will be tested at the caster
- Finish porting the new IMC control software, automating and implementing the procedure at the caster, and test the system in commercial practice. (planned for the near future)
- To ensure stability of the approach, it is necessary to understand the variations in the data used to design the filter. Establishing bounds on the variations is a ongoing process
- Evaluate the system performance, to account for and extract mold friction information from the measured behavior of the operating caster / oscillator / control system, (requires the help of a model of mold dynamics).

Acknowledgements

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