CCC Annual Report

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Capturing and Suppressing Resonance in Steel Casting Mold Oscillation Systems

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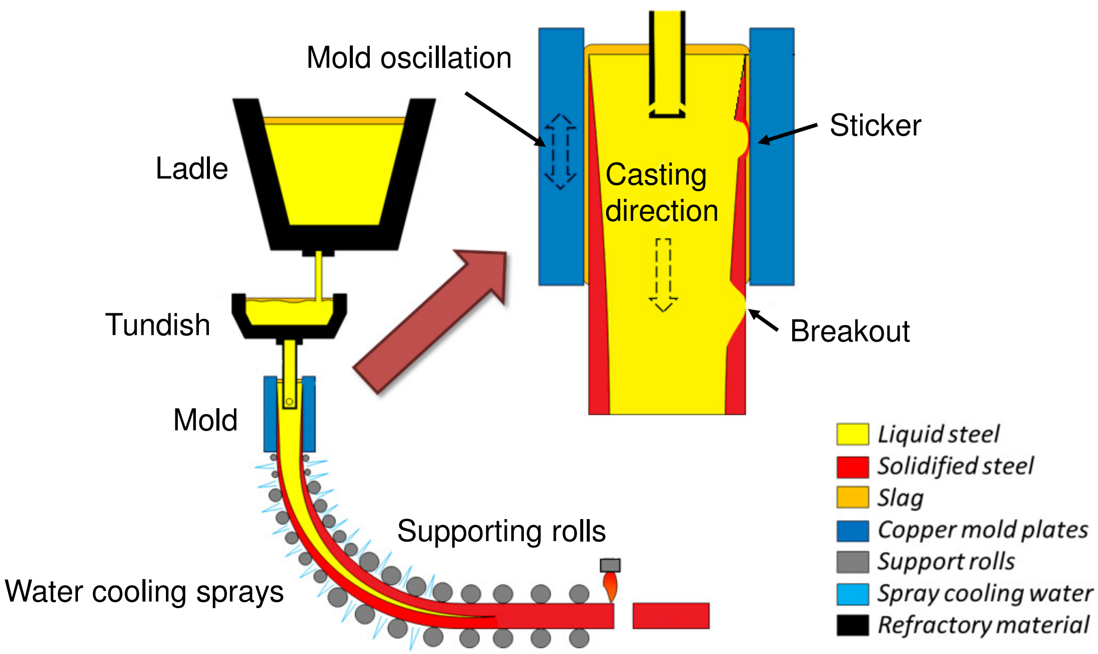
Continuous Steel Casting System Components



From CCC

Production Unit Resonance Problem and Project Objective

Pivot

Mold Table

Primary Beam

Position of hydraulic piston (not in picture) under the beam

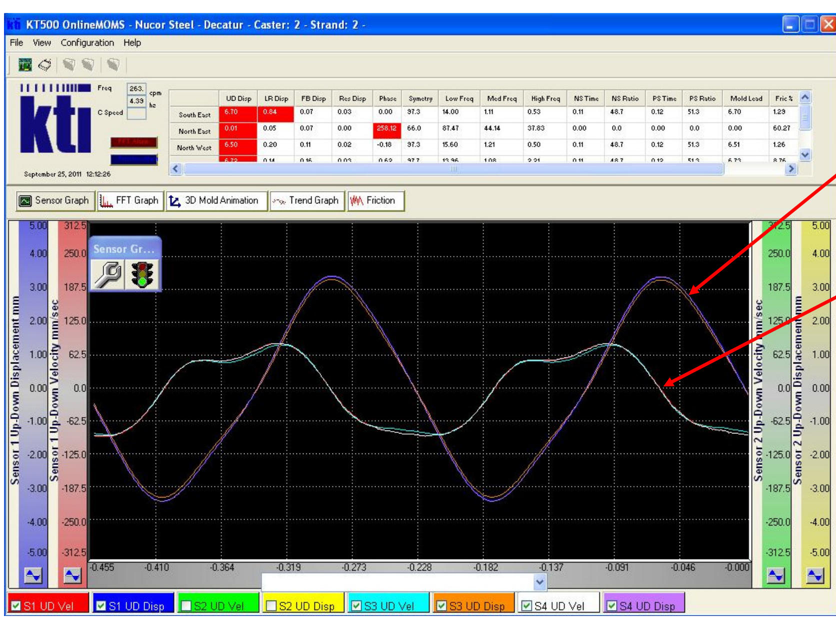
Problem:

1. The primary beam main resonance mode starts exhibiting excitation when the reference frequency approaches ***one-third*** of the resonance frequency
2. The mold displacement and velocity profiles distortion is found to be mainly caused by this resonance
3. The reason for the onset of beam resonance excitation has not been identified
4. Distortion has not been removed, operation in the desired frequency range has not been attained

<u>Specific Project Objective:</u> Model this mold oscillation system, simulate it, identify what causes excitation of the primary beam resonance, and eliminate the distortion

Position and Velocity Profile from Plant

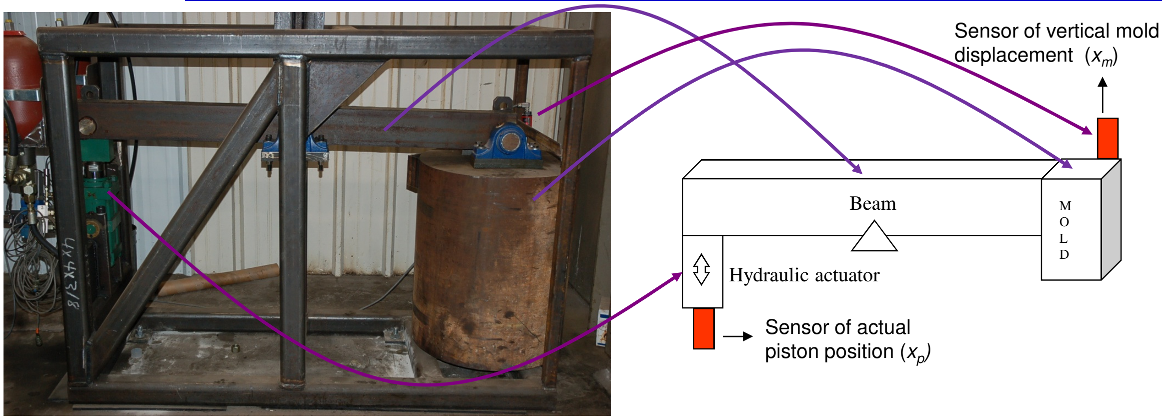
In going to higher frequencies, while reducing the oscillation amplitude, the velocity profile is found to become highly distorted.



Position profile

Velocity profile (time derivative of position profile)

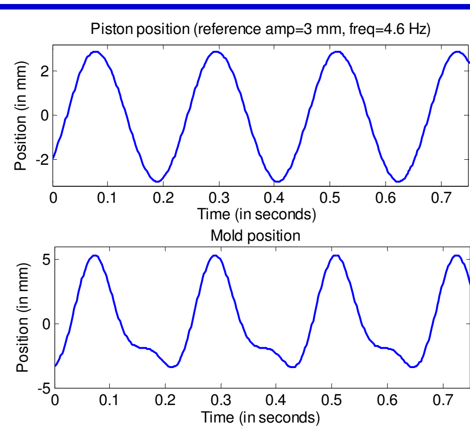
Hardware Testbed: Simplified Instrumented Layout



- Hardware testbed captures similar resonance problem
  - 1) Resonance frequency of beam – 9.2 Hz
  - 2) Reference input to the piston for tracking - sinusoid at 4.6Hz
  - 3) Mold position profile is highly distorted
- Hydraulic valve/actuator – Nonlinear behavior (same model as plant)

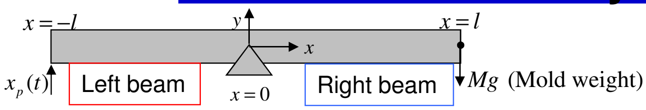
Note: Although the objective is a distortion-free mold velocity profile, we focus on piston and mold position profiles observed through sensor signals, since a distortion-free (pure sinusoidal) displacement guarantees a distortion-free velocity.

Hardware Testbed Validation: Experimental Data Exhibits Resonance Problem



- The desired position for the piston – reference input - is a sine wave of frequency f_r=4.6 Hz (half of 9.2 Hz - the resonance frequency of the beam) and amplitude 3 mm
- P controller with gain K=2 is used
- **Piston position signal looks almost perfect**
- **But large distortion (deviation from sinusoidal profile) at the mold end is observed**
- This happens when the frequency of the piston position reference is near a **submultiple (exact integer fraction) of the beam resonance frequency**

Software testbed: beam model: set of Timoshenko beam (linear) PDEs with boundary conditions



- Model of two beams coupled at the center via boundary conditions
- Mold dynamics is one of the boundary conditions

1D dynamic beam transverse displacement PDE

$$m_b \frac{\partial^2 y_L}{\partial t^2} + \gamma_y \frac{\partial y_L}{\partial t} = \frac{\partial}{\partial x}\left(k' G a_b \left(\frac{\partial y_L}{\partial x} - \psi_L\right)\right) - m_b g$$

$$m_b \frac{\partial^2 y_R}{\partial t^2} + \gamma_y \frac{\partial y_R}{\partial t} = \frac{\partial}{\partial x}\left(k' G a_b \left(\frac{\partial y_R}{\partial x} - \psi_R\right)\right) - m_b g$$

1D dynamic beam bending angle PDE

$$\frac{I}{a_b} m_b \frac{\partial^2 \psi_L}{\partial t^2} + \gamma_\psi \frac{\partial \psi_L}{\partial t} = \frac{\partial}{\partial x}\left(EI \frac{\partial \psi_L}{\partial x}\right) + k' G a_b \left(\frac{\partial y_L}{\partial x} - \psi_L\right)$$

$$\frac{I}{a_b} m_b \frac{\partial^2 \psi_R}{\partial t^2} + \gamma_\psi \frac{\partial \psi_R}{\partial t} = \frac{\partial}{\partial x}\left(EI \frac{\partial \psi_R}{\partial x}\right) + k' G a_b \left(\frac{\partial y_R}{\partial x} - \psi_R\right)$$

Boundary conditions at $x=-l$

$$y_L(-l) = x_p(t)$$

$$EI \frac{\partial \psi_L(-l)}{\partial x} = 0$$

Boundary conditions at $x=0$

$$y_L(0) = 0 \quad y_R(0) = 0 \quad \psi_L(0) = \psi_R(0)$$

$$EI \frac{\partial \psi_R(0)}{\partial x} = EI \frac{\partial \psi_L(0)}{\partial x}$$

Boundary conditions at $x=l$

$$EI \frac{\partial \psi_R(l)}{\partial x} = 0$$

$$k' G a_b \left(\frac{\partial y_R(l)}{\partial x} - \psi_R(l)\right) + Mg + M \frac{\partial^2 y_R(l)}{\partial t^2} + \gamma_m \frac{\partial y_R(l)}{\partial t} = 0$$

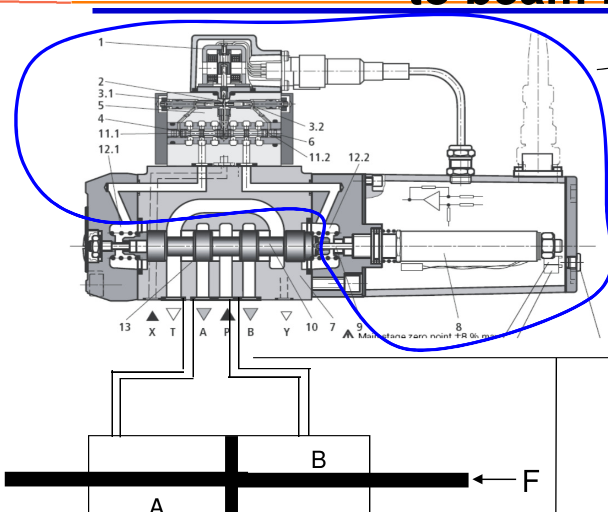
$E = 200Gpa$ Youngs modulus
$G = 82GPa$ Shear Modulus for steel
ρ=7870 Kg/m^3 - density of steel
$a_b = 0.0088 m^2$ cross section area of beam
$I = 2.2\times10^{-5} m^4$ Moment of inertia of beam
$m_b = 69 Kg/m$ Mass per unit length of beam
M=2250 Kgs mold mass
$k' = 0.83$ Shear constant
Beam width = 5.13'(hollow with thickness 0.94')
$l = 34.5'$
Beam breadth = 6' (hollow with thickness 0.38')
Damping coefficients $\gamma_L = \gamma_R = 10$ $\gamma = 10$Kg/sec

Testbed parameters

$y_R =$ linear displacement (right side of the beam)
$\psi_R =$ angular displacement (right side of the beam)
$y_L =$ linear displacement (left side of the fulcrum)
$\psi_L =$ angular displacement (left side of the beam)

PDE variables

Software testbed: hydraulic servo model: set of nonlinear ODEs and their coupling to beam PDEs



Electronic control of spool position
$\ddot{x}_s + 2\zeta_s \omega_s \dot{x}_s + \omega_s^2 x_s = \omega_s^2 u$.
Fast dynamics - hence not modelled in simulations

Turbulent Flow Equations for flow in chambers 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 spool position - positive when the spool moves to the right.
Its mean position is 0 with valve underlap gap of d on both sides.
ω_s , ζ_s - spool filter parameters,
q_A flow rate for champber A - positive when oil flows in to A
q_B flow rate for chamber B - positive when oil flows out of B
$\alpha_{A,B,s,t}$ – Chamber connected to A and B, source, tank

Pressure equations (P_A and P_B - pressure in A and B, respectively

$$\dot{P}_A = \frac{\beta}{\left(V_A + a_p(L + x_p)\right)}\left(q_A - a_p \dot{x}_p\right)$$

$$\dot{P}_B = \frac{\beta}{\left(V_B + a_p(L - x_p)\right)}\left(-q_B + a_p \dot{x}_p\right)$$

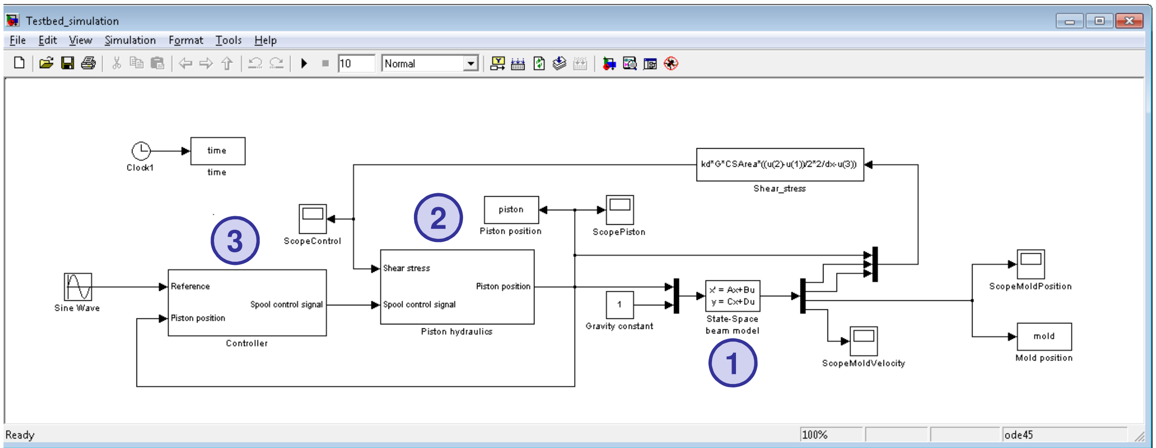
x_p – piston position - positive if it moves to right of midpoint of cylinder
V_A, V_B – static volume of chambers A and B
$a_p(L + x_p)$ – dynamic volume of chambers A and B
L – half the piston stroke length
a_p – surface area of piston

Piston dynamics showing the coupling with the beam

$$m_p \ddot{x}_p + b\dot{x}_p = (P_A - P_B)a_p - m_p g + k' G a_b \left(\frac{\partial y_L(-l)}{\partial x} - \psi_L(-l)\right)$$

The position of the piston is a boundary condition for the beam equations and the shear force in the beam acts on the piston, coupling the two models together

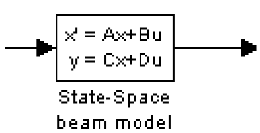
Software Testbed: Numerical Model – MATLAB Simulink Diagram



1. Finite-difference approximation (ODE) of Timoshenko beam equation (PDE)
2. Nonlinear piston hydraulics
3. Control algorithm (to be designed)

Software Testbed: Numerical Model – PDE Approximation

1 beam

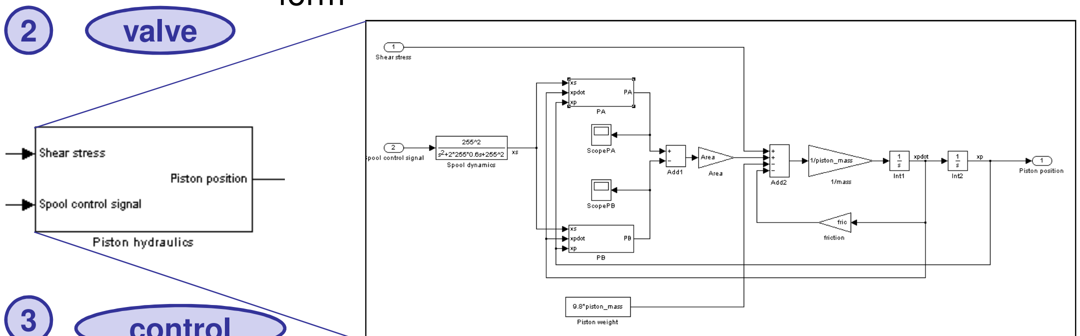


- Finite difference approximation – spatial derivatives in PDE are approximated by difference equations

$$\frac{\partial f}{\partial x} \approx \frac{f(x+\Delta x) - f(x-\Delta x)}{2\Delta x} \qquad \frac{\partial^2 f}{\partial x^2} \approx \frac{f(x+\Delta x) - 2f(x) + f(x-\Delta x)}{(\Delta x)^2}$$

- Result is a set of coupled time-ODEs implemented in state-space form.
- Nonlinear servo model ODE is implemented in state-space form

2 valve

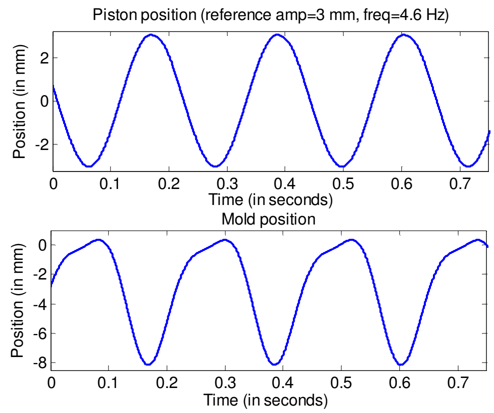


3 control

- Controller is implemented in state-space form

Software Testbed (Numerical Code) Validation

- Software (Matlab) program was written that computes analytical model response to inputs using numerical algorithms
- Parameters in the beam model were chosen to obtain a resonance frequency at 9.2 Hz
- Reference input to the piston for tracking – sinusoid at 4.6 Hz
- Proportional controller used with gain of 0.6
- **Mold position exhibits distortions similar to those of the testbed**
- Therefore, numerical model simulator can be used as a platform for understanding the testbed dynamics and for testing controllers
- We've got a tool for *in silico* experimentation - our own software testbed to play with!



Application of Mold Oscillation Model to Severstal Caster

- For Severstal caster, parameters needed for the model can't be directly measured due to complexity of assembly, which is similar to Nucor.
- **Change model parameters** for 9.2 Hz resonance at Nucor Decatur mold oscillation system to Severstal with an measured 5.0 Hz resonance frequency
- **Retune *k* factor** to adjust cross-sectional moment of inertia by factor of k^2. k is the value that maximizes the mold displacement magnitude response at the desired resonance. For Severstal, k is found to be 0.4823, yielding damped natural frequency 5.0042 Hz and resonance frequency 5.0025 Hz.
- **Adjustment of each individual model** for runtime (dicretization accuracy) versus resonance frequency to allow model to run in real time while still maintaining the required resonance.

Resonance frequency analysis

Set solution of two coupled beams equation as

$$y(x,t) = y(x)e^{j\omega t},\ \psi(x,t) = \psi(x)e^{j\omega t},\ x_p(t) = x_p e^{j\omega t}$$

Then, the corresponding equations for both the left and right beams have the form

$$-\omega^2 y + j\omega\frac{\gamma_y}{m_b}y = \frac{k'Ga_b}{m_b}\left(\frac{\partial^2 y}{\partial x^2} - \frac{\partial \psi}{\partial x}\right) - g,\ -\omega^2\psi + j\omega\frac{\gamma_\psi a_b}{Im_b}\psi = \frac{Ea_b}{m_b}\frac{\partial^2\psi}{\partial x^2} + \frac{k'Ga_b^2}{Im_b}\left(\frac{\partial y}{\partial x} - \psi\right)$$

or in matrix form

$$\begin{bmatrix} \frac{k'Ga_b}{m_b}r^2 + \left(\omega^2 - j\omega\frac{\gamma_y}{m_b}\right) & -\frac{k'Ga_b}{m_b}r \\ \frac{k'Ga_b^2}{Im_b}r & \frac{Ea_b}{m_b}r^2 + \left(\omega^2 - j\omega\frac{\gamma_\psi a_b}{Im_b} - \frac{k'Ga_b^2}{Im_b}\right) \end{bmatrix} \begin{bmatrix} y(x) \\ \psi(x) \end{bmatrix} = 0.$$

Equating determinant to zero yields equation below with four solutions: $\pm r_1$ and $\pm r_2$

$$r^4 + r^2\left(\omega^2\frac{m_b}{a_b}\left(\frac{1}{E} + \frac{1}{k'G}\right) - j\omega\left(\frac{\gamma_\psi}{EI} + \frac{\gamma_y}{k'Ga_b}\right)\right) + \frac{m_b^2}{k'Ga_b^2}\left(\omega^4 - j\omega^3\left(\frac{\gamma_\psi a_b}{Im_b} + \frac{\gamma_y}{m_b}\right)\right.$$
$$\left. -\omega^2\left(\frac{\gamma_y\gamma_\psi a_b^2}{Im_b^2} + \frac{k'Ga_b^2}{Im_b}\right) + j\omega\frac{\gamma_y k'Ga_b^2}{Im_b^2}\right) = 0.$$

General solution

Then, the general solution is

$$\begin{bmatrix} y(x) \\ \psi(x) \end{bmatrix} = C\begin{bmatrix} y \\ \psi \end{bmatrix}e^{rx} = \tilde{C}_1\begin{bmatrix} \tilde{y}_1 \\ \tilde{\psi}_1 \end{bmatrix}e^{r_1 x} + \tilde{C}_2\begin{bmatrix} \tilde{y}_2 \\ \tilde{\psi}_2 \end{bmatrix}e^{-r_1 x} + \tilde{C}_3\begin{bmatrix} \tilde{y}_3 \\ \tilde{\psi}_3 \end{bmatrix}e^{r_2 x} + \tilde{C}_4\begin{bmatrix} \tilde{y}_4 \\ \tilde{\psi}_4 \end{bmatrix}e^{-r_2 x},$$

with eigenvalues

$$\begin{bmatrix} \tilde{y}_i \\ \tilde{\psi}_i \end{bmatrix} = \begin{bmatrix} k'Ga_b r_i / m_b \\ \left(k'Ga_b r_i^2 + \left(m_b\omega^2 - j\omega\gamma_y\right)\right)/m_b \end{bmatrix}$$

Now we substitute this solution into boundary conditions below:

$$y_L(-l) = 0,\ EI\psi'_L(-l) = 0,\ y_L(0) = 0,\ y_R(0) = 0,$$
$$\psi_L(0) = \psi_R(0),\ EI\psi'_L(0) = EI\psi'_R(0),\ EI\psi'_R(l) = 0,$$
$$k'Ga_b\left(y'_R(l) - \psi_R(l)\right) - \frac{4M^2\omega^2 + \gamma_m^2}{4M}\psi_R(l) = 0.$$

Frequency response calculation

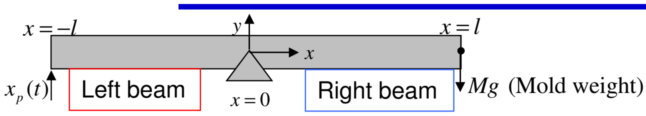
This substitution (of the general solution into the boundary conditions) yields

$$
\begin{aligned}
&C_{1L}y_1e^{-r_1l}+C_{2L}y_2e^{r_1l}+C_{3L}y_3e^{-r_2l}+C_{4L}y_4e^{r_2l}=x_p,\\
&C_{1L}r_1\psi_1e^{-r_1l}-r_2C_{2L}\psi_2e^{r_1l}+r_2C_{3L}y_3e^{-r_2l}-r_2C_{4L}y_4e^{r_2l}=0,\\
&C_{1L}y_1+C_{2L}y_2+C_{3L}y_3+C_{4L}y_4=0,\\
&C_{1R}y_1+C_{2R}y_2+C_{3R}y_3+C_{4R}y_4=0,\\
&C_{1L}y_1+C_{2L}y_2+C_{3L}y_3+C_{4L}y_4=C_{1L}y_1+C_{2L}y_2+C_{3L}y_3+C_{4L}y_4,\\
&C_{1L}r_1\psi_1-C_{2L}r_1\psi_2+C_{3L}r_2\psi_3-C_{4L}r_2\psi_4=C_{1R}r_1\psi_1-C_{2R}r_2\psi_2+C_{3R}r_2\psi_3-C_{4R}r_2\psi_4,\\
&C_{1L}r_1\psi_1e^{r_1l}-C_{2L}r_2\psi_2e^{-r_1l}+C_{3L}r_2\psi_3e^{r_2l}-C_{4L}r_2\psi_4e^{-r_2l}=C_{1R}r_1\psi_1e^{r_1l}-C_{2R}r_2\psi_2e^{-r_1l}\\
&\qquad+C_{3R}r_2\psi_3e^{r_2l}-C_{4R}r_2\psi_4e^{-r_2l},\\
&C_{1R}e^{r_1l}D_1+C_{2R}e^{-r_1l}D_2+C_{3R}e^{r_2l}D_3+C_{4R}e^{-r_2l}D_4=0,
\end{aligned}
$$

$$
\begin{aligned}
D_1&=k'Ga_b\left(r_1y_1-\psi_1\right)-y_1\left(M\omega^2+j\omega\gamma_m\right),\\
D_2&=-k'Ga_b\left(r_1y_2-\psi_2\right)-y_2\left(M\omega^2+j\omega\gamma_m\right),\\
D_3&=k'Ga_b\left(r_2y_3-\psi_3\right)-y_3\left(M\omega^2+j\omega\gamma_m\right),\\
D_4&=-k'Ga_b\left(r_2y_4-\psi_4\right)-y_4\left(M\omega^2+j\omega\gamma_m\right)
\end{aligned}
$$

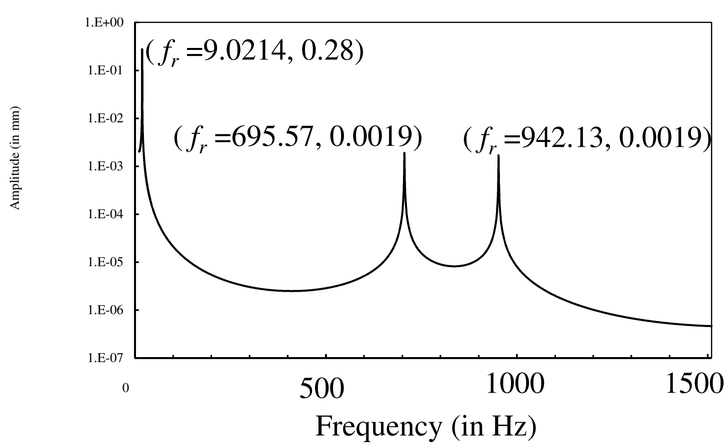
Factoring out all the coefficients *C* obtain matrix equation. Using tabled parameters solve for roots *r*.

Software testbed: frequency response



- the frequency response can then be obtained from the equation for the mold vertical displacement at the right end:

$$y_R(l)=C_{1R}\tilde{y}_1e^{r_1l}+C_{2R}\tilde{y}_2e^{-r_1l}+C_{3R}\tilde{y}_3e^{r_2l}+C_{4R}\tilde{y}_4e^{-r_2l}$$



Numerical modeling

Mold oscillation testbed is modeled by the second-order finite difference scheme

$$\partial^2 y/\partial t^2 = B_1\left(y(x-\Delta x) - 2y(x) + y(x+\Delta x)\right) + B_2\left(\psi(x-\Delta x) - \psi(x+\Delta x)\right) - D_1\,\partial y/\partial t - g$$

The "zero moment" boundary condition for the left end is

$$EI\frac{\partial \psi_L(-l)}{\partial x} \approx EI\frac{\psi_L(-l+\Delta x) - \psi_L(-l-\Delta x)}{2\Delta x} = 0,$$
$$\psi_L(-l-\Delta x) \approx \psi_L(-l+\Delta x),$$
$$\psi_L(-l) \approx \psi_L(-l+\Delta x).$$

Same method can be applied at the right end. For the boundary condition due to mold reaction force:

$$\frac{\partial^2 y_R(l)}{\partial t^2} \approx -\frac{\gamma_m}{M}\frac{\partial y_R(l)}{\partial t} - \frac{k'Ga_b}{M}\frac{y_R(l) - y_R(l-\Delta x)}{\Delta x} + \frac{k'Ga_b}{M}\psi_R(l-\Delta x) - g$$

For the boundary condition of equal moments at the hinge (Taylor expansion)

$$\psi(\Delta x) = \psi(0) + \Delta x\psi'(0) + \Delta x^2\psi''(0)/2 + \dots,$$
$$\psi(2\Delta x) = \psi(0) + 2\Delta x\psi'(0) + 2\Delta x^2\psi''(0) + \dots,$$

Since $\psi_L(0) = \psi_R(0)$ the angular displacement is

$$\psi_{L,R}(0) \approx -\frac{\psi_L(-2\Delta x)}{6} + \frac{2}{3}\left(\psi_L(-\Delta x) + \psi_R(\Delta x)\right) - \frac{\psi_R(2\Delta x)}{6}$$

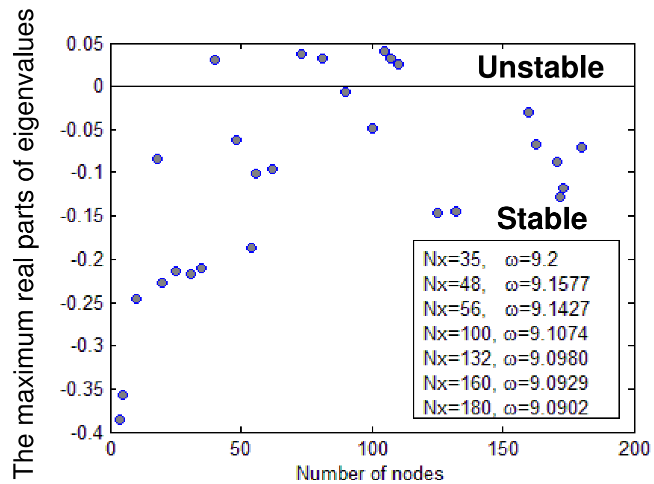
Mold Oscillation System Parameters

Parameter	Variable	Nominal value		Units
		Nucor-Steel	Severstal	
Mass of beam per unit length	m_b	69.256	33.402	kg/m
Area of cross section of beam	a_b	0.0088	0.0042	m^2
Shear modulus	G	$7.7 \cdot 10^{10}$	$7.7 \cdot 10^{10}$	Pa
Modulus of elasticity	E	$2 \cdot 10^{11}$	$2 \cdot 10^{11}$	Pa
Moment of inertia of the beam cross-section	I	$2.2085 \cdot 10^{-5}$	$0.5137 \cdot 10^{-5}$	m^2
Beam transverse displacement	γ_y	10	10	kg/(m·sec)
Beam angular displacement	γ_ψ	10	10	kg/(m·sec)
Mold damping	γ_m	1	1	kg/(m·sec)
Shape of the cross-section factor	k'	0.83	0.83	
Mold mass	M	2250	2250	kg
Half of beam length	l	0.88	0.88	m

Numerical Model Stability

The maximum real parts of eigenvalues vs. discretization nodes number:

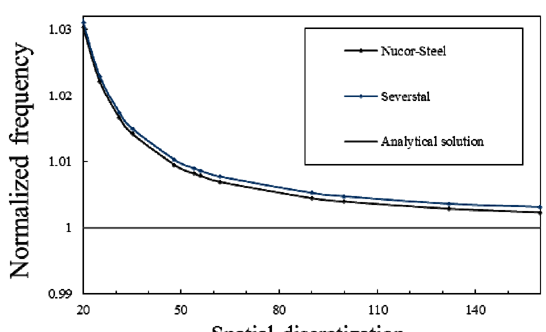
$$X_t = AX + Bu,$$
$$Y = CX + Du.$$



- When the real part is negative, the system is stable
- For most discretization nodes number, the system is stable

Software Testbed: Resonance Frequency Dependence on Spatial Discretization Step Size

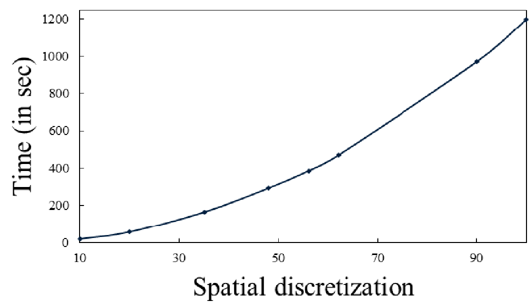
Δx	Nucor-Steel		Severstal	
	Damped Nat. Frequency, Hz	Resonance Frequency, Hz	Damped Nat. Frequency, Hz	Resonance Frequency, Hz
l/20	9.2953	9.2948	5.1559	5.1552
l/35	9.1493	9.1489	5.0753	5.0746
l/48	9.1071	9.1067	5.0520	5.0512
l/56	9.0923	9.0919	5.0438	5.0431
l/100	9.0574	9.0570	5.0245	5.0237
l/132	9.0477	9.0473	5.0188	5.0180
l/160	9.0427	9.0424	5.0164	5.0156
Analytical solution	9.0213	9.0214	5.0042	5.0025



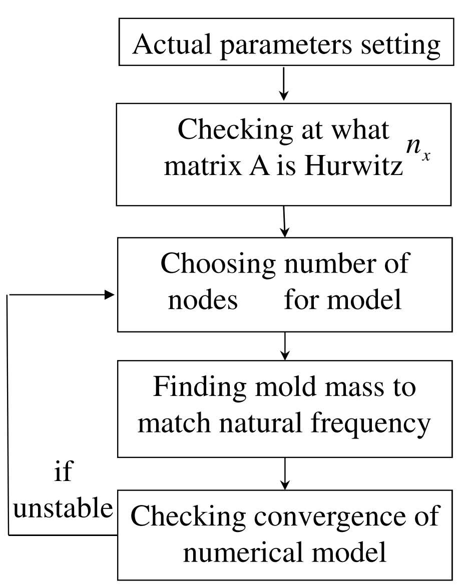
Normalized frequency is ratio of simulated resonance and analytical solution

Runtime Dependence on Spatial Discretization Step Size

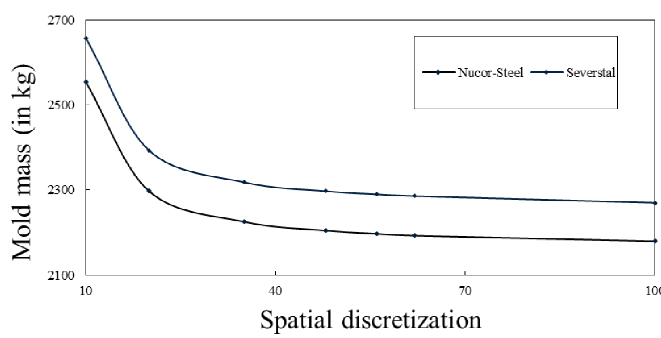
Δx	Nucor-Steel		Severstal	
	Damped Nat. Frequency, Hz	Resonance Frequency, Hz	Damped Nat. Frequency, Hz	Resonance Frequency, Hz
l/20	9.2953	9.2948	5.1559	5.1552
l/35	9.1493	9.1489	5.0753	5.0746
l/48	9.1071	9.1067	5.0520	5.0512
l/56	9.0923	9.0919	5.0438	5.0431
l/100	9.0574	9.0570	5.0245	5.0237
l/132	9.0477	9.0473	5.0188	5.0180
l/160	9.0427	9.0424	5.0164	5.0156
Analytical solution	9.0213	9.0214	5.0042	5.0025



Real-time Virtual Testbed: Parameter Matching for the Desired Resonance Frequency and Runtime

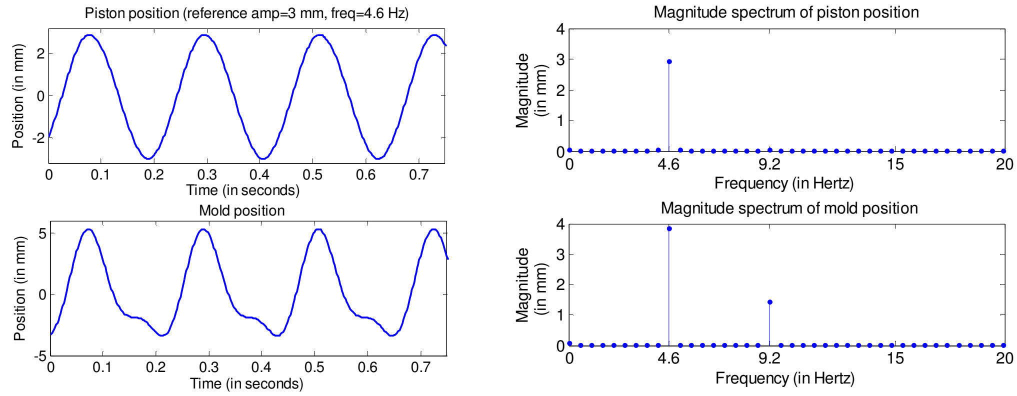


Δx	Nucor-Steel		Severstal	
	Mold mass, kg	Resonance Frequency, Hz	Mold mass, kg	Resonance Frequency, Hz
l/10	2553.0	9.2009	2658.0	5.0002
l /20	2296.9	9.2001	2392.5	5.0001
l /35	2225.2	9.2000	2318.0	5.0001
l /48	2204.7	9.2000	2297.0	5.0001
l /56	2197.0	9.2011	2289.6	5.0000
l /62	2192.7	9.2018	2285.5	5.0000
l /100	2179.6	9.2021	2270.0	5.0023



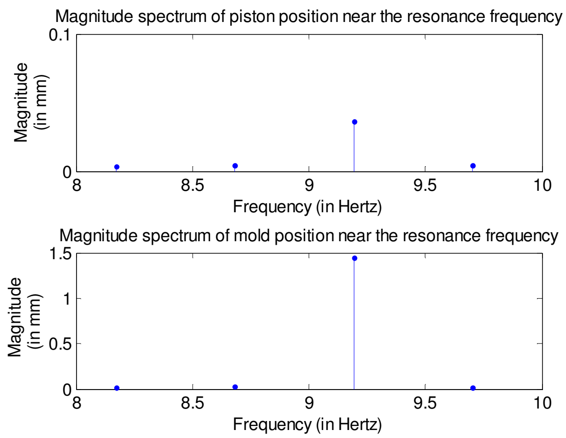
Hardware testbed performance diagnostics

The initial conjecture: nonlinear actuator dynamics produces harmonics of 4.6 Hz distorting the piston position, which in turn distorts the mold position



- The mold position is distorted, but *the piston position **looks perfect to the naked eye*** both in time domain and frequency domain (no visible harmonics).
- Since the actuator is visually observed to perform ideally, *the initial conjecture **looks wrong***

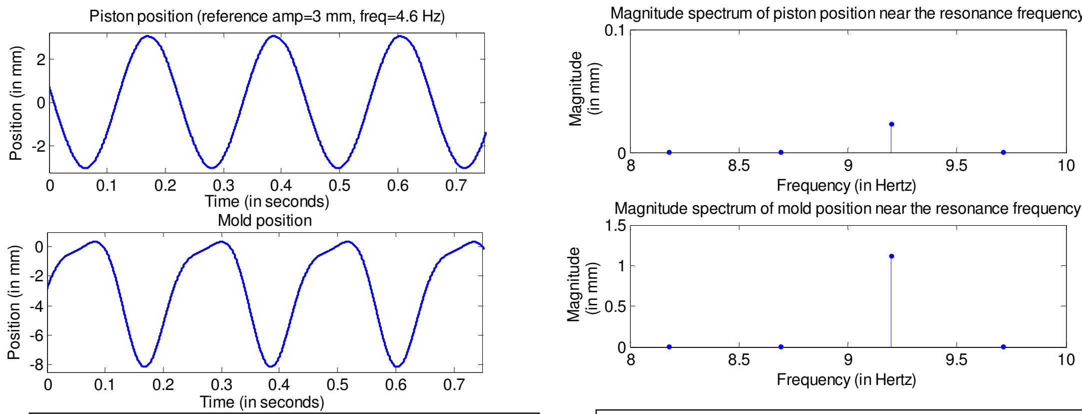
Catching the culprit with a magnifying glass!



- *However, zooming into the magnitude spectrum* of the piston position reveals a small peak at 9.2 Hz
- **Naked eye is easily misled!** Missed small *piston position distortion due to nonlinear actuator dynamics*
- Peak is small, but critical - at resonance frequency it is amplified by about 40 times to cause a large peak at the mold end!
- This matches with other experiments, in which a sinusoidal piston position amplitude at 9.2 Hz is amplified by the beam by a factor of 30 at the mold end

Conclusion: The small peak in piston position being at resonance frequency excites the beam resonance and creates the large peak in the mold position. *All previous attempts at solving this problem failed because the distortions were thought to start in the beam. In fact, however, they start in the piston, and are just amplified by the beam.*

Source of Distortion (Revisited)

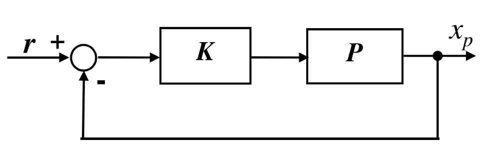


Simulation result: piston and mold position with piston reference at 4.6 Hz

Simulation result: magnitude spectra of piston and mold position near resonance frequency

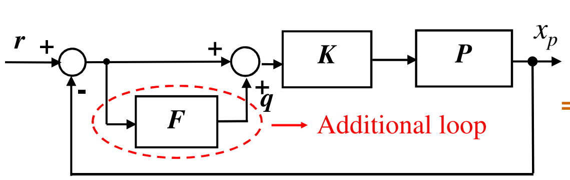
- Piston position looks perfect but has small peak at 9.2 Hz in magnitude spectrum generated by the nonlinear servo model
- This small peak is amplified by beam causing significant distortion in the mold position
- Software testbed generates the resonance problem and can be used for initial testing of potential solutions (modeling is NOT necessary for the solution approach developed)

Solution Technique–Loop Augmentation



- x_p tracks reference sinusoid r of frequency f_r well
- but has a small undesired sinusoidal component at frequency nf_r

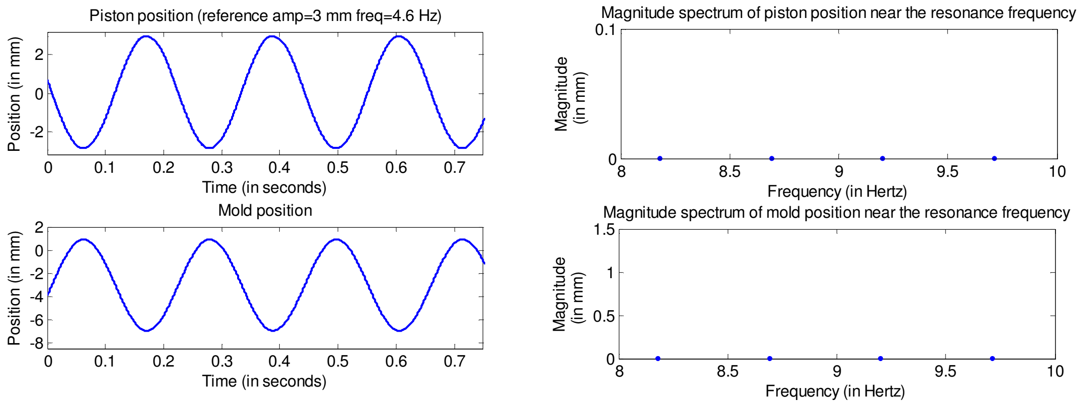
x_p represents the displacement of piston



- augment closed loop with controller F as shown
- then x_p continues to track r well, but has no undesired sinusoid at frequency nf_r

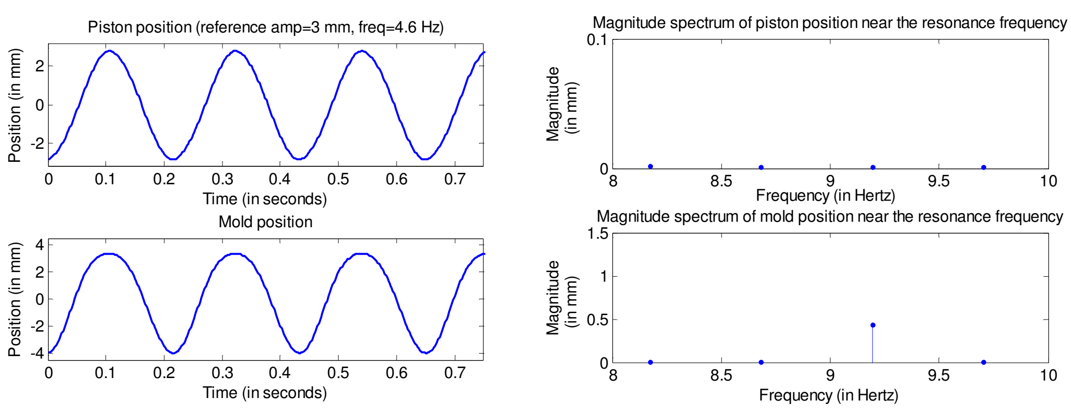
Software Testbed: Simulation Results with the Filter-Nucor

- Controller developed is applied to the model
- Internal model controller takes the form of filter F with ζ=0.1 and $2\omega_r$=$2\cdot 2\pi f_r$ =2π x 9.2 rad/sec
- Controller K=0.6 and reference sinusoid – amplitude 3 mm and frequency f_r= 4.6 Hz
- Distortions in mold position profile completely eliminated

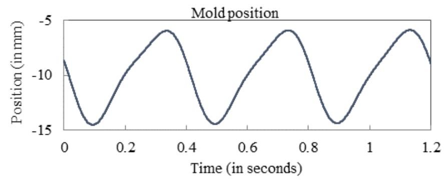


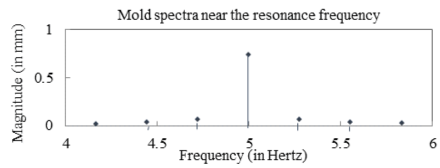
Digital Controller Implementation: Hardware Testbed-Nucor

- Controller developed is applied to hardware testbed to remove harmonic at 9.2 Hz from the piston position
- Filter F introduced with ζ =0.1 and $2\omega_r$=2π x 9.2 rad/sec
- Controller K=2, reference sinusoid – amplitude 3 mm and frequency 4.6 Hz



Mold Position Distortion of Severstal Model

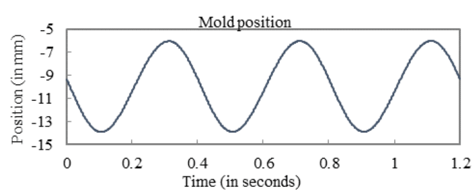


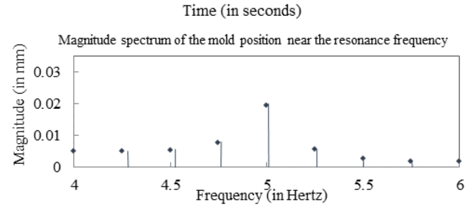


Simulation result: mold position with piston reference at 2.5 Hz

- Input at the frequency 2.5 Hz leads to the distortion in the mold displacement profile due to small amplitude sinusoid at the resonance frequency of the Severstal plant in the input from the actuator

Application of the Control Law to Severstal Model

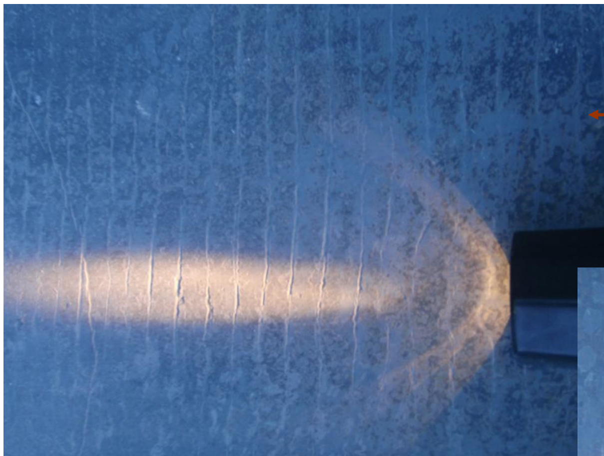




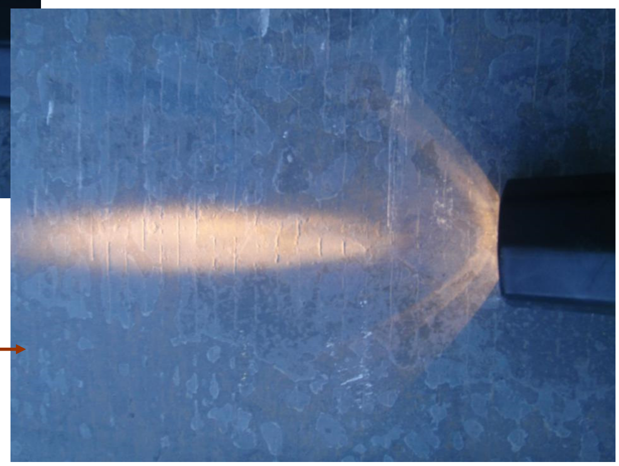
Simulation result: mold position with piston reference at 2.5 Hz

- the resonance magnitude reduced by 40 times and the mold position distortion is eliminated

Effect of Project Results on Casting Product Quality



Casting at a lower frequency of oscillation for the mold enforced by simple proportional controllers



Casting at a higher frequency of oscillation for the mold enabled by the proposed controller - shallower surface marks and hence less likely to turn into surface cracks

Conclusions (Past)

- The mold velocity distortion problem at Nucor Steel, Decatur has been solved for a hinged beam-type mold oscillation system by implementing a new feedback control law based on the disturbance model. Software testbed helped!
- The new control law eliminates periodic mold velocity disturbances, such as those caused by excitation of system natural frequencies by actuator nonlinearity, without the need for system model
- **This improves the surface quality of the steel being produced and also enables the production of other crack sensitive grades of steel**
- The controller is now a permanent feature of caster operation

Conclusions (New)

- An adjustable software simulation testbed has been developed that permits recreating the distortion problem in both thin and thick continuous casters
- This testbed retains the exact resonance frequency matching for a broad range of computing capabilities through simple one-parameter adjustment
- **As a result, the software testbed developed can be run in real time and connected to the production control system for mold oscillation controller testing and debugging, without the need for building the hardware testbed**

Acknowledgments

- Continuous Casting Consortium Members (ABB, AK Steel, ArcelorMittal, Baosteel, JFE Steel Corp., Magnesita Refractories, Nippon Steel and Sumitomo Metal Corp., Nucor Steel, Postech/ Posco, SSAB, ANSYS/ Fluent)
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