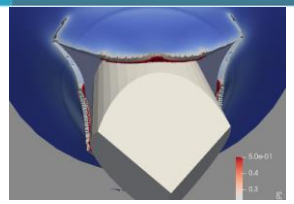
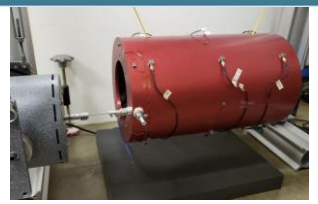
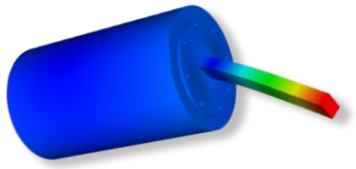
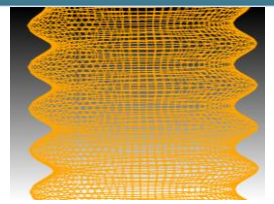
Quantifying the Effect of Non-Physical Parameters on the Nonlinear Dynamics of an Electromechanical Ratcheting Mechanism

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Rob Flicek, Scott Grutzik, Rob Kuether, Chris Schumann, Aabhas Singh, and Kumar Vemaganti

August 5th, 2025

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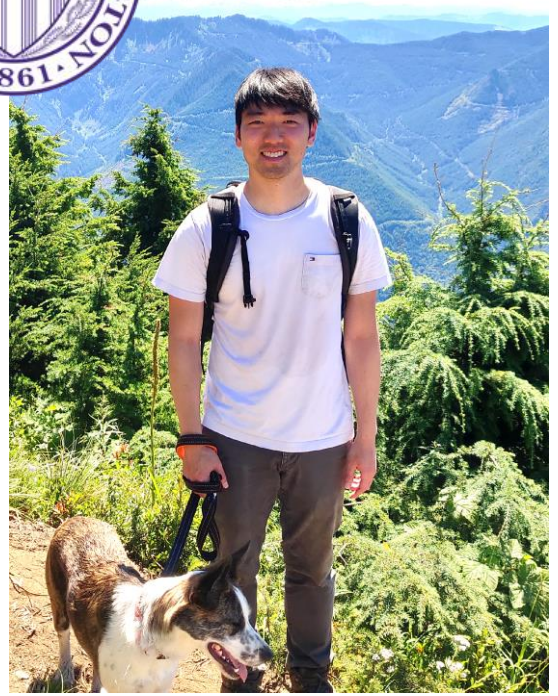

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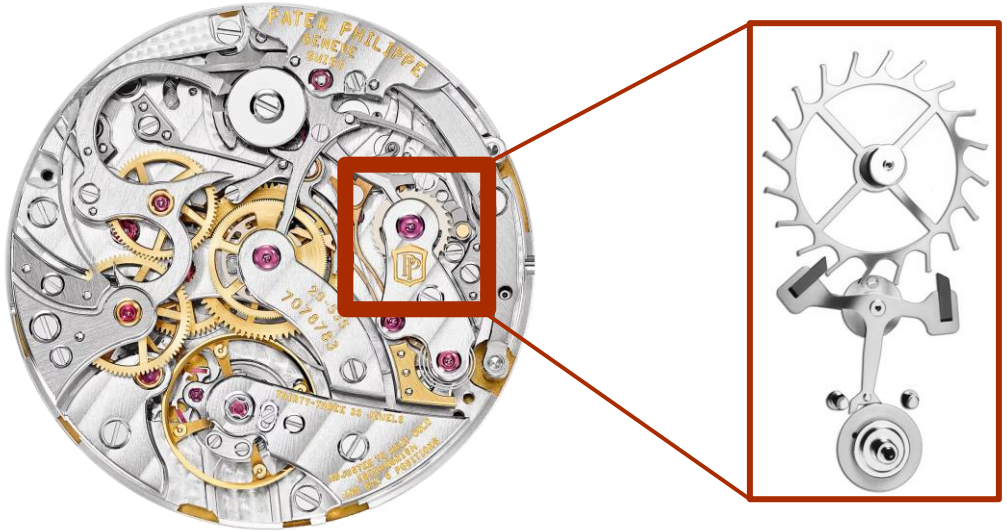
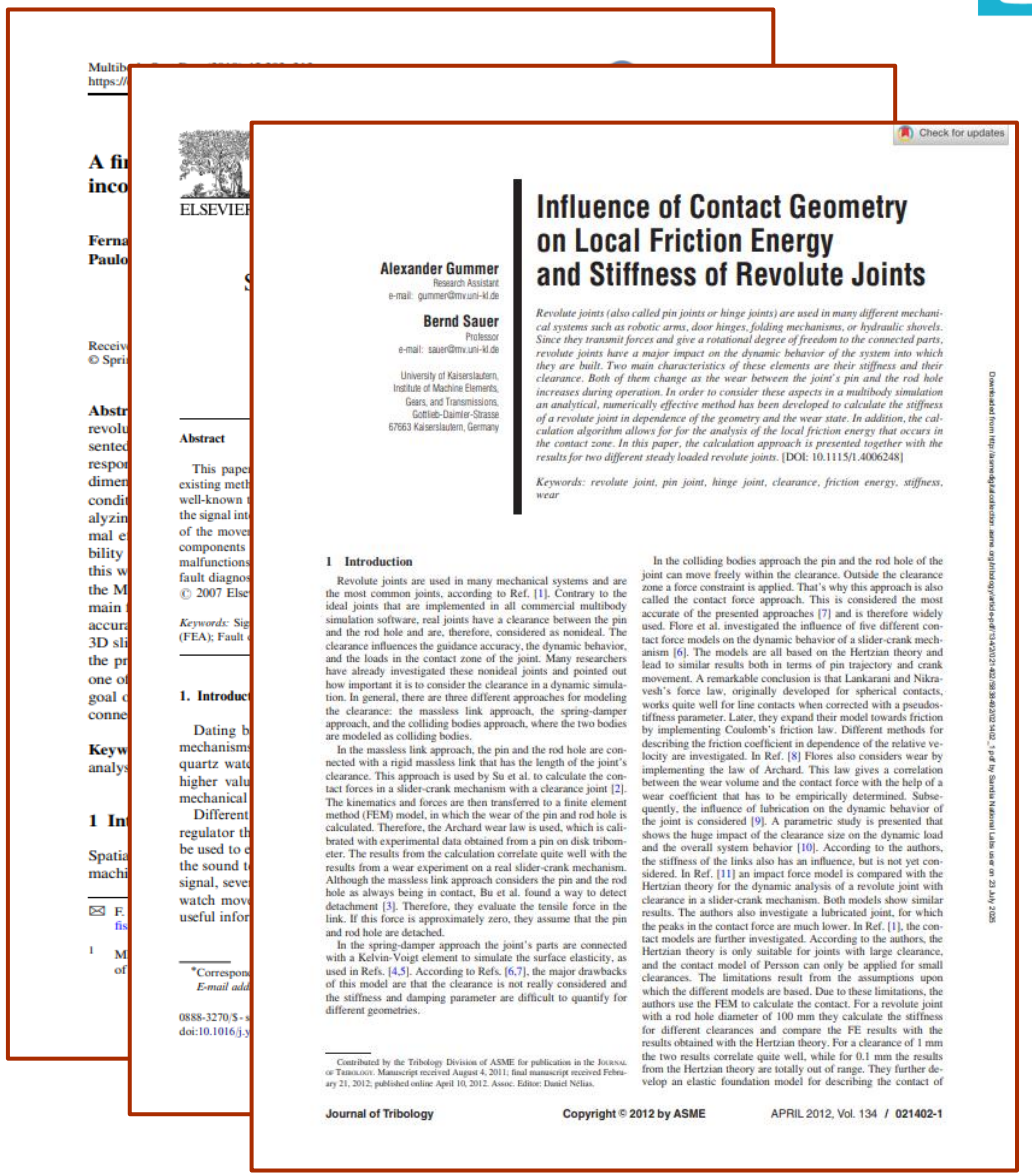
Background & Motivation

- Complex ratchet-like mechanisms exist in many commercial and aerospace applications.
- Previous work has looked at FEA's sensitivity to nonphysical parameters by analyzing a single component.
- This work seeks to quantify the effect that these parameters have on multiple components that interact with each other.





Influence of Contact Geometry on Local Friction Energy and Stiffness of Revolute Joints

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Abstract

Revolute joints (also called pin joints or hinge joints) are used in many different mechanical systems such as robotic arms, door hinges, folding mechanisms, or hydraulic shovels. Since they transmit forces and give a rotational degree of freedom to the connected parts, revolute joints have a major impact on the dynamic behavior of the system into which they are built. Two main characteristics of these elements are their stiffness and their clearance. Both of them change as the wear between the joint's pin and the rod hole increases during operation. In order to consider these aspects in a multibody simulation an analytical, numerically effective method has been developed to calculate the stiffness of a revolute joint in dependence of the geometry and the wear state. In addition, the calculation algorithm allows for for the analysis of the local friction energy that occurs in the contact zone. In this paper, the calculation approach is presented together with the results for two different steady loaded revolute joints. [DOI: 10.1115/1.4006248]

Keywords: revolute joint, pin joint, hinge joint, clearance, friction energy, stiffness, wear

1 Introduction

Revolute joints are used in many mechanical systems and are the most common joints, according to Ref. [1]. Contrary to the ideal joints that are implemented in all commercial multibody simulation software, real joints have a clearance between the pin and the rod hole and are, therefore, considered as nonideal. The clearance influences the guidance accuracy, the dynamic behavior, and the loads in the contact zone of the joint. Many researchers have already investigated these nonideal joints and pointed out how important it is to consider the clearance in a dynamic simulation. In general, there are three different approaches for modeling the clearance: the massless link approach, the spring-damper approach, and the colliding bodies approach, where the two bodies are modeled as colliding bodies.

In the massless link approach, the pin and the rod hole are connected with a rigid massless link that has the length of the joint's clearance. This approach is used by Su et al. to calculate the contact forces in a slider-crank mechanism with a clearance joint [2]. The kinematics and forces are then transferred to a finite element method (FEM) model, in which the wear of the pin and rod hole is calculated. Therefore, the Archard wear law is used, which is calibrated with experimental data obtained from a pin on disk tribometer. The results from the calculation correlate quite well with the results from a wear experiment on a real slider-crank mechanism. Although the massless link approach considers the pin and the rod hole as always being in contact, Bu et al. found a way to detect detachment [3]. Therefore, they evaluate the tensile force in the link. If this force is approximately zero, they assume that the pin and rod hole are detached.

In the spring-damper approach the joint's parts are connected with a Kelvin-Voigt element to simulate the surface elasticity, as used in Refs. [4,5]. According to Refs. [6,7], the major drawbacks of this model are that the clearance is not really considered and the stiffness and damping parameter are difficult to quantify for different geometries.

In the colliding bodies approach the pin and the rod hole of the joint can move freely within the clearance. Outside the clearance zone a force constraint is applied. That's why this approach is also called the contact force approach. This is considered the most accurate of the presented approaches [7] and is therefore widely used. Flore et al. investigated the influence of five different contact force models on the dynamic behavior of a slider-crank mechanism [6]. The models are all based on the Hertzian theory and lead to similar results both in terms of pin trajectory and crank movement. A remarkable conclusion is that Lankarani and Nikravesh's force law, originally developed for spherical contacts, works quite well for line contacts when corrected with a pseudostiffness parameter. Later, they expand their model towards friction by implementing Coulomb's friction law. Different methods for describing the friction coefficient in dependence of the relative velocity are investigated. In Ref. [8] Flores also considers wear by implementing the law of Archard. This law gives a correlation between the wear volume and the contact force with the help of a wear coefficient that has to be empirically determined. Subsequently, the influence of lubrication on the dynamic behavior of the joint is considered [9]. A parametric study is presented that shows the huge impact of the clearance size on the dynamic load and the overall system behavior [10]. According to the authors, the stiffness of the links also has an influence, but is not yet considered. In Ref. [11] an impact force model is compared with the Hertzian theory for the dynamic analysis of a revolute joint with clearance in a slider-crank mechanism. Both models show similar results. The authors also investigate a lubricated joint, for which the peaks in the contact force are much lower. In Ref. [1], the contact models are further investigated. According to the authors, the Hertzian theory is only suitable for joints with large clearance, and the contact model of Persson can only be applied for small clearances. The limitations result from the assumptions upon which the different models are based. Due to these limitations, the authors use the FEM to calculate the contact. For a revolute joint with a rod hole diameter of 100 mm they calculate the stiffness for different clearances and compare the FE results with the results obtained with the Hertzian theory. For a clearance of 1 mm the two results correlate quite well, while for 0.1 mm the results from the Hertzian theory are totally out of range. They further develop an elastic foundation model for describing the contact of

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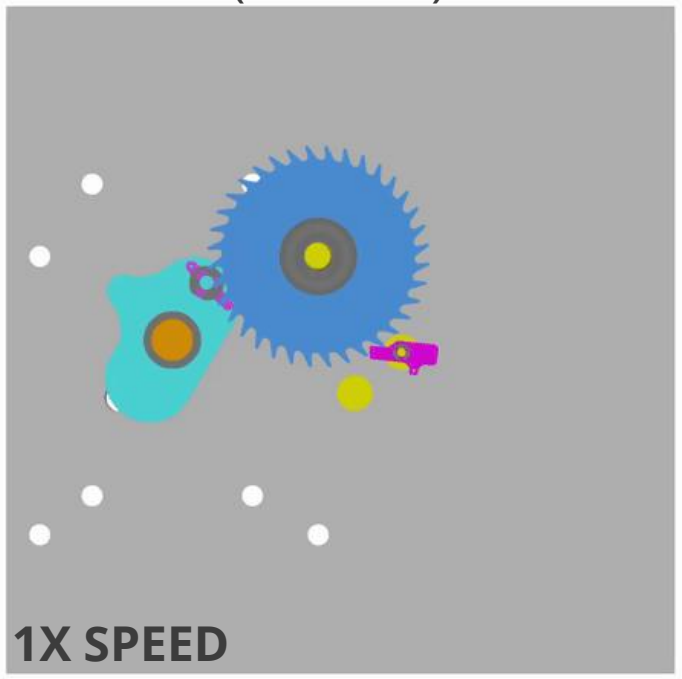
Image Credit: https://www.watchfinder.co.uk/articles/getting-technical-swiss-lever-escapement
https://www.patek.com/en/collection/movements/29-535

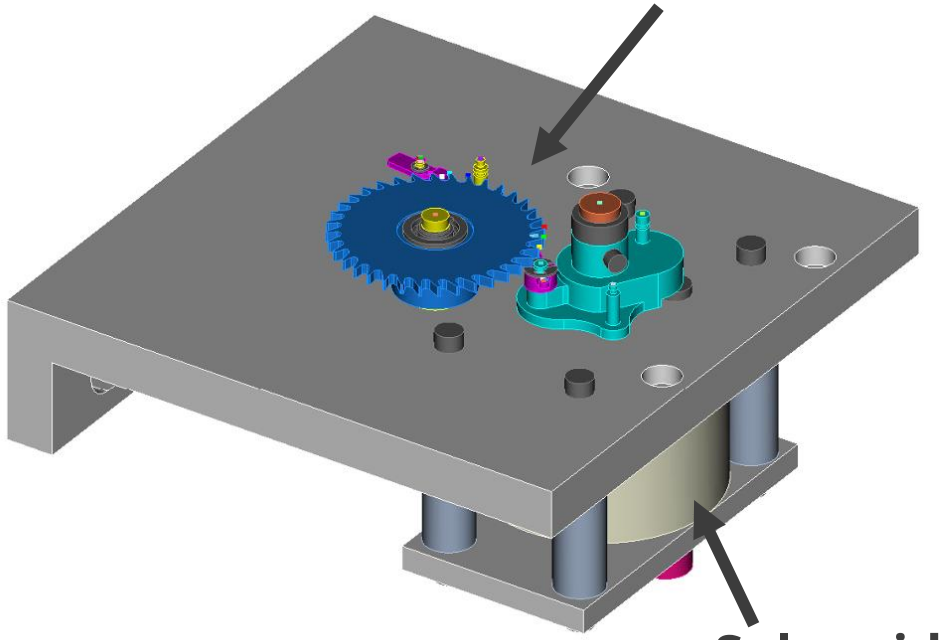
Ratcheting Mechanism Overview

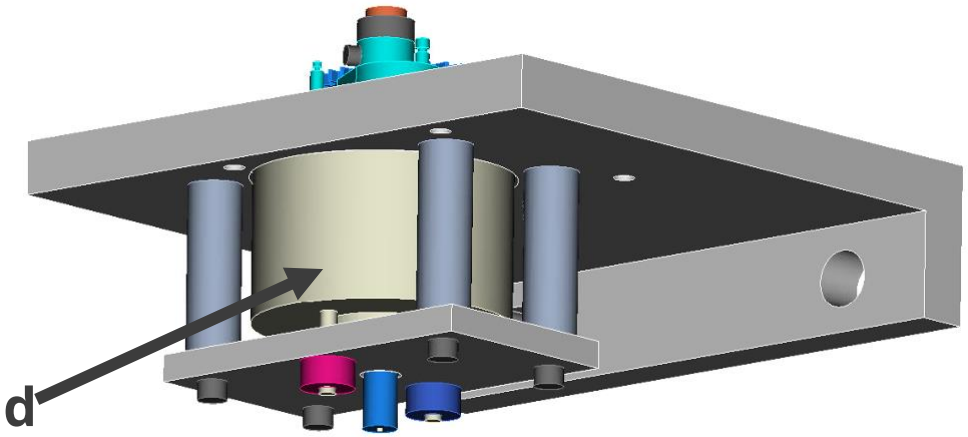
BASEBALL AND NICKELS FOR SCALE

https://en.wikipedia.org/wiki/Baseball_%28ball%29

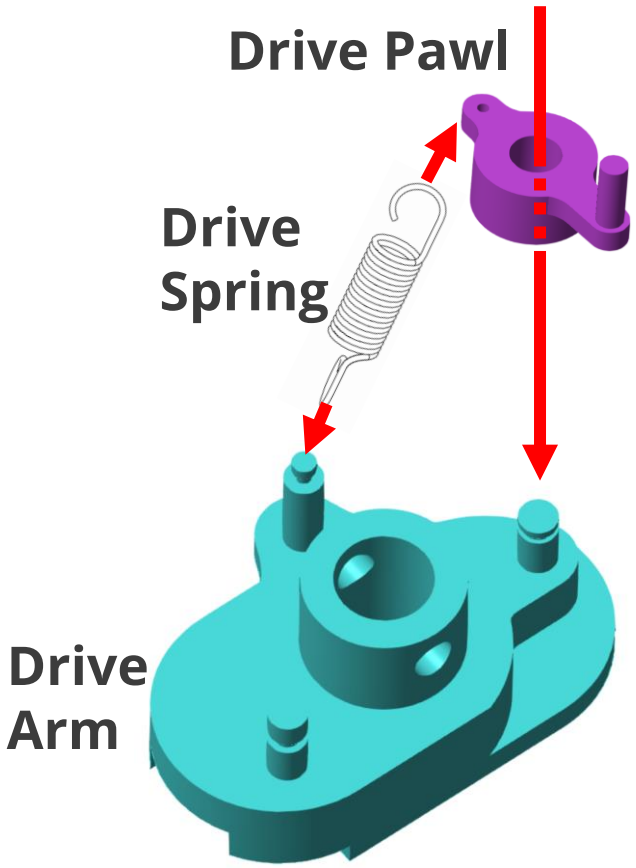
Multibody Simulation
(MATLAB)

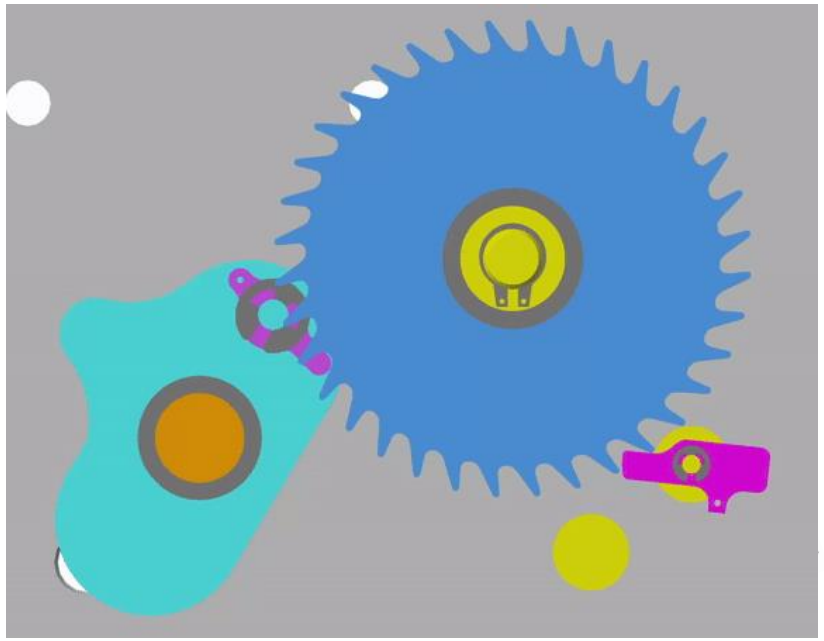


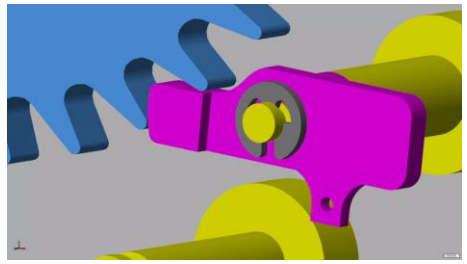


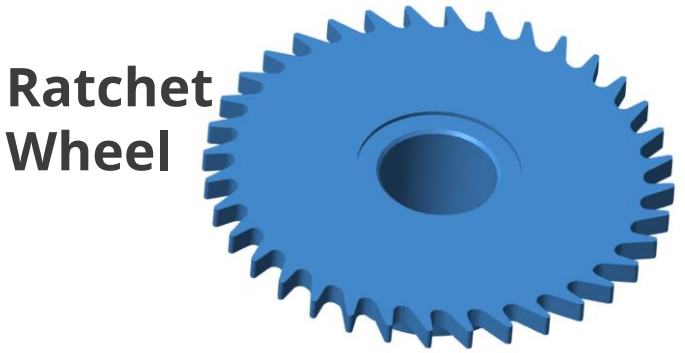


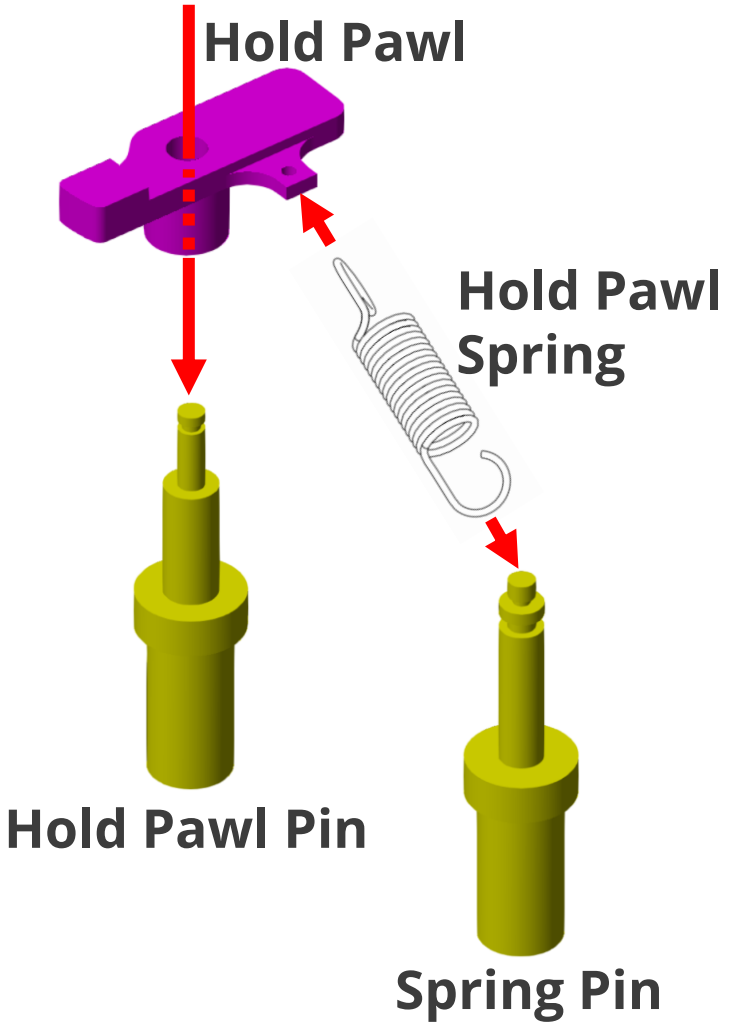
Ratcheting Mechanism Components



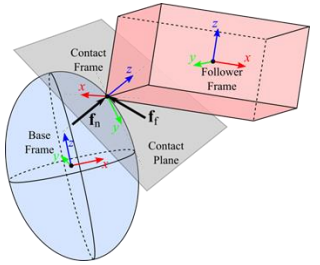


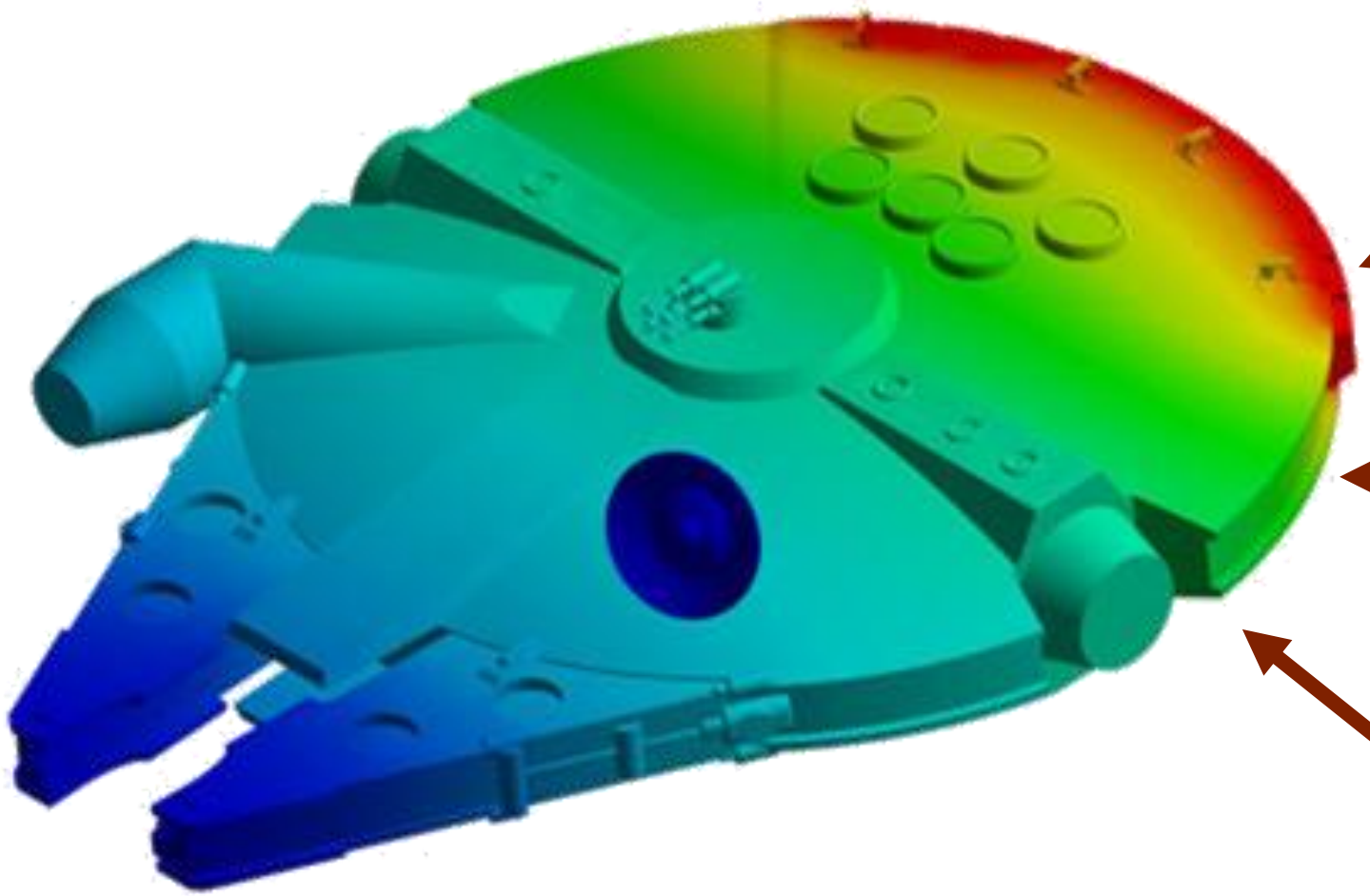






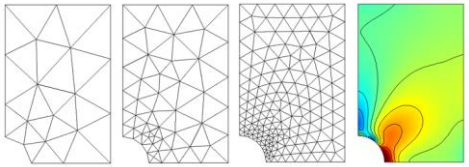
Finite Element Method





Momentum Balance Iterations

- Calculate and apply contact force to minimize penetration distance



Mesh Density

- Typically, a finer mesh results in better solution but increases computation cost

Processor Count

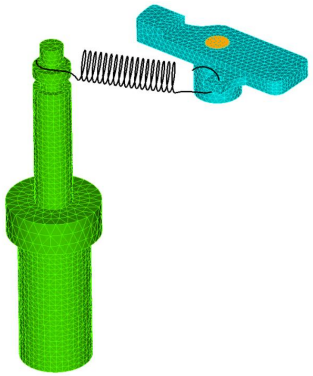
- Analysis is split up between multiple parallel processors to speed up processing

https://fetchcfd.com/view-project/2453-thermal-simulation-millennium-falcon
https://study.com/academy/lesson/what-is-momentum-definition-equation-units-principle.html
https://www.comsol.com/blogs/meshing-considerations-linear-static-problems

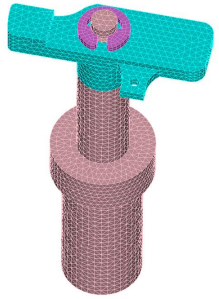
Project Overview

- **Conduct parameter studies on small submodels of the ratcheting mechanism**
- **Find out how changing parameters affect a ratcheting mechanism experiencing different excitation environments**

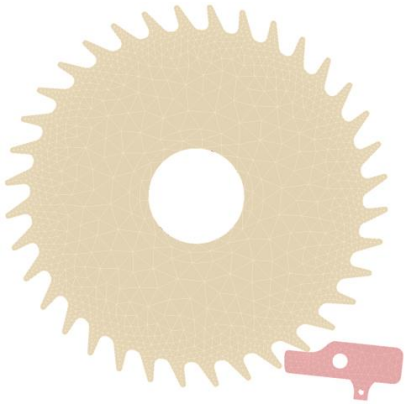
Submodels



Pin-spring-pawl



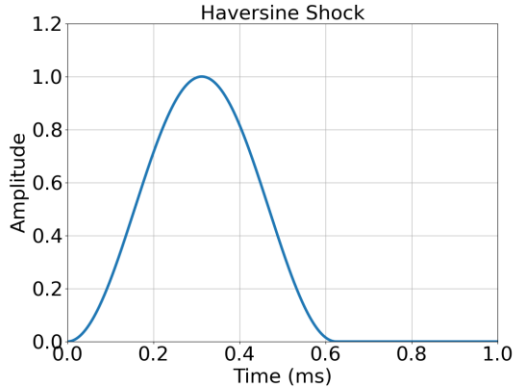
Pin-pawl



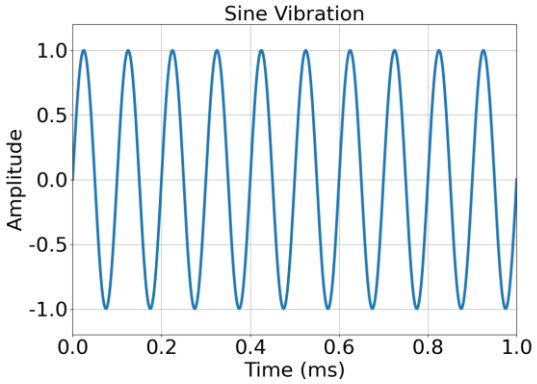
Pawl-gear

Momentum Balance Iterations: 1, 5, 10, 20, 50, 75, 100
Mesh Density: Fine, Nominal, Coarse, Very Coarse
Processor Count: 50%, 100%, 150%, 200%, 250%

Environments:



Haversine Shock

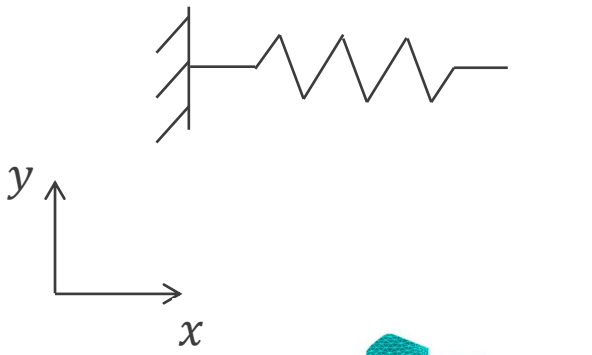


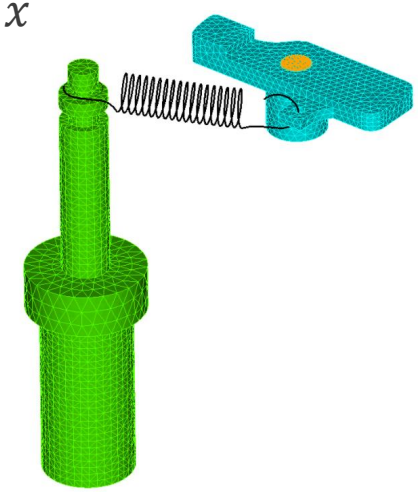
Sinusoidal Vibration

(Unit amplitude shown)

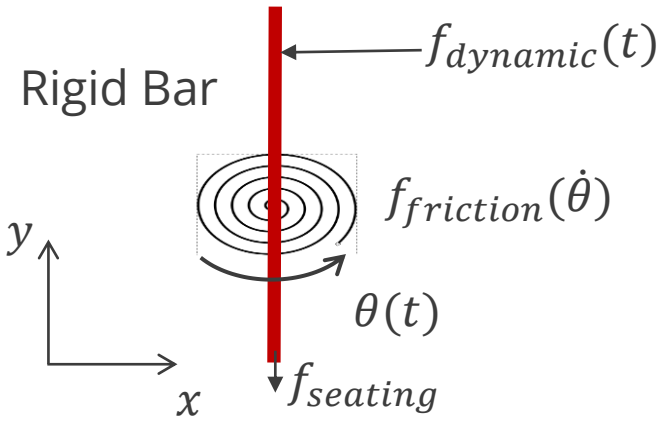
Idealized Model

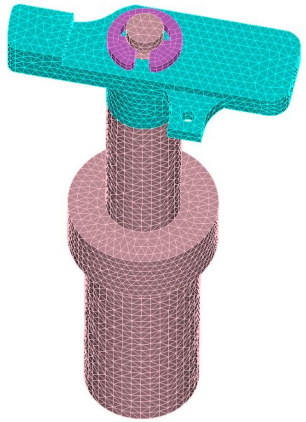
- **A simplified approximation of a system for quick and easy analysis**



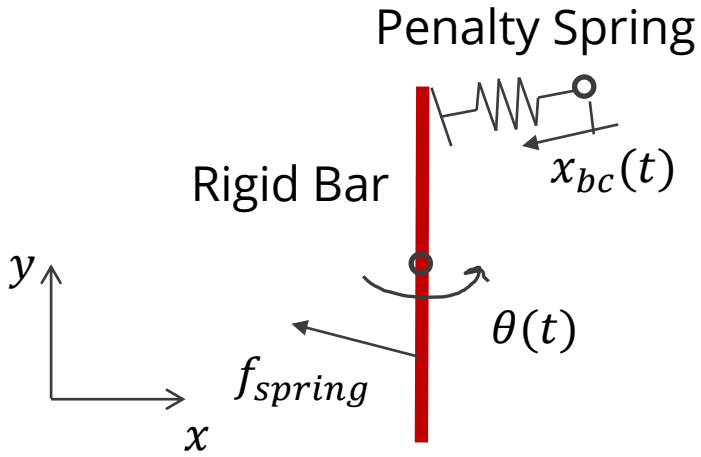


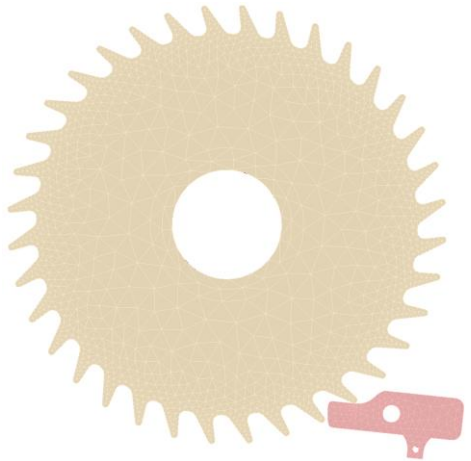
Pin-spring-pawl





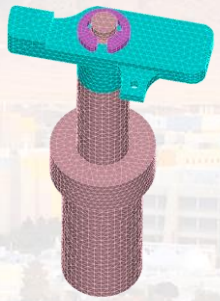
Pin-pawl





Pawl-gear

Pin-spring-pawl



Pin-pawl

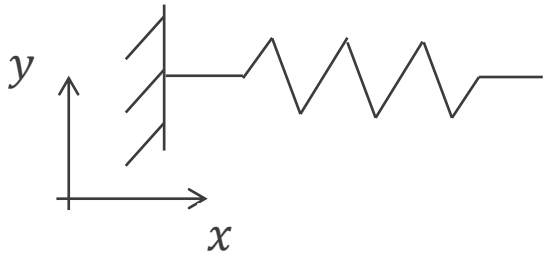
Pawl-gear

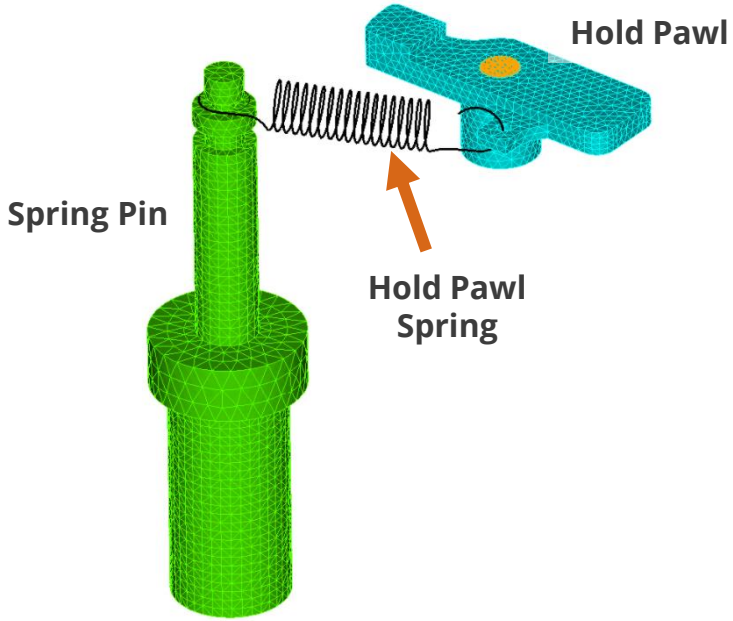
Submodel – Pin-Spring-Pawl

Pin-Spring-Pawl Submodel

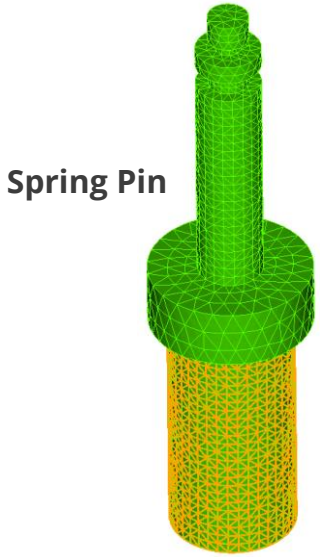
- **Boundary Conditions**
  - Base of Pin is fixed in all degree's of freedom
  - Center of pawl is displaced in the axial direction of the spring
- **Quantities of Interest**
  - Contact force on Hold Pawl and Spring Pin

Idealized Model:

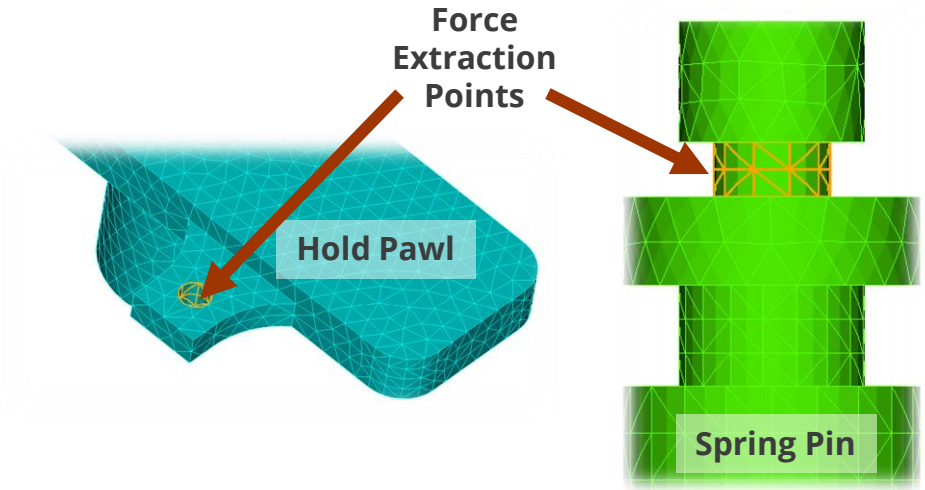




Prescribed displacement to the Hold Pawl in the axial direction of the spring



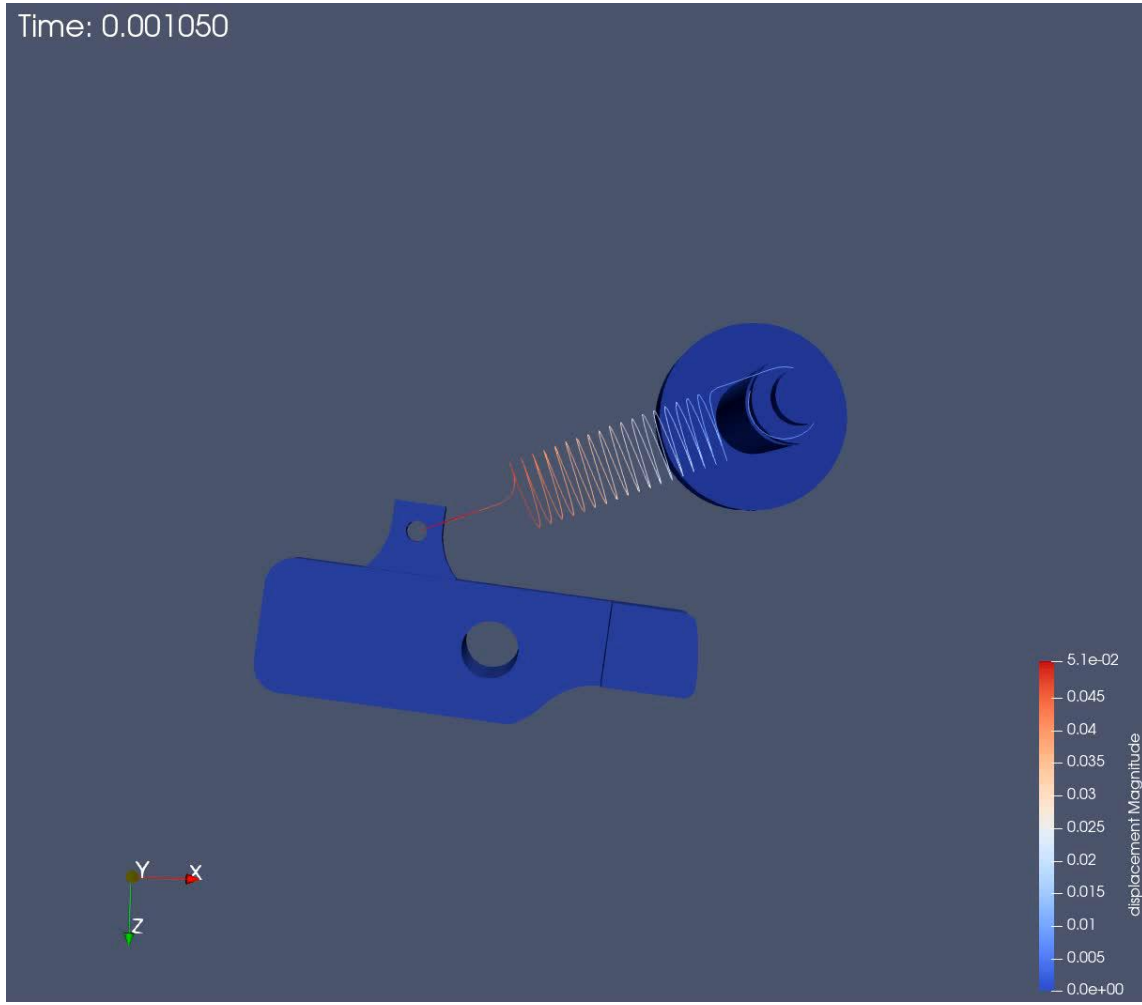
Base of Pin is fixed in all DOF's.



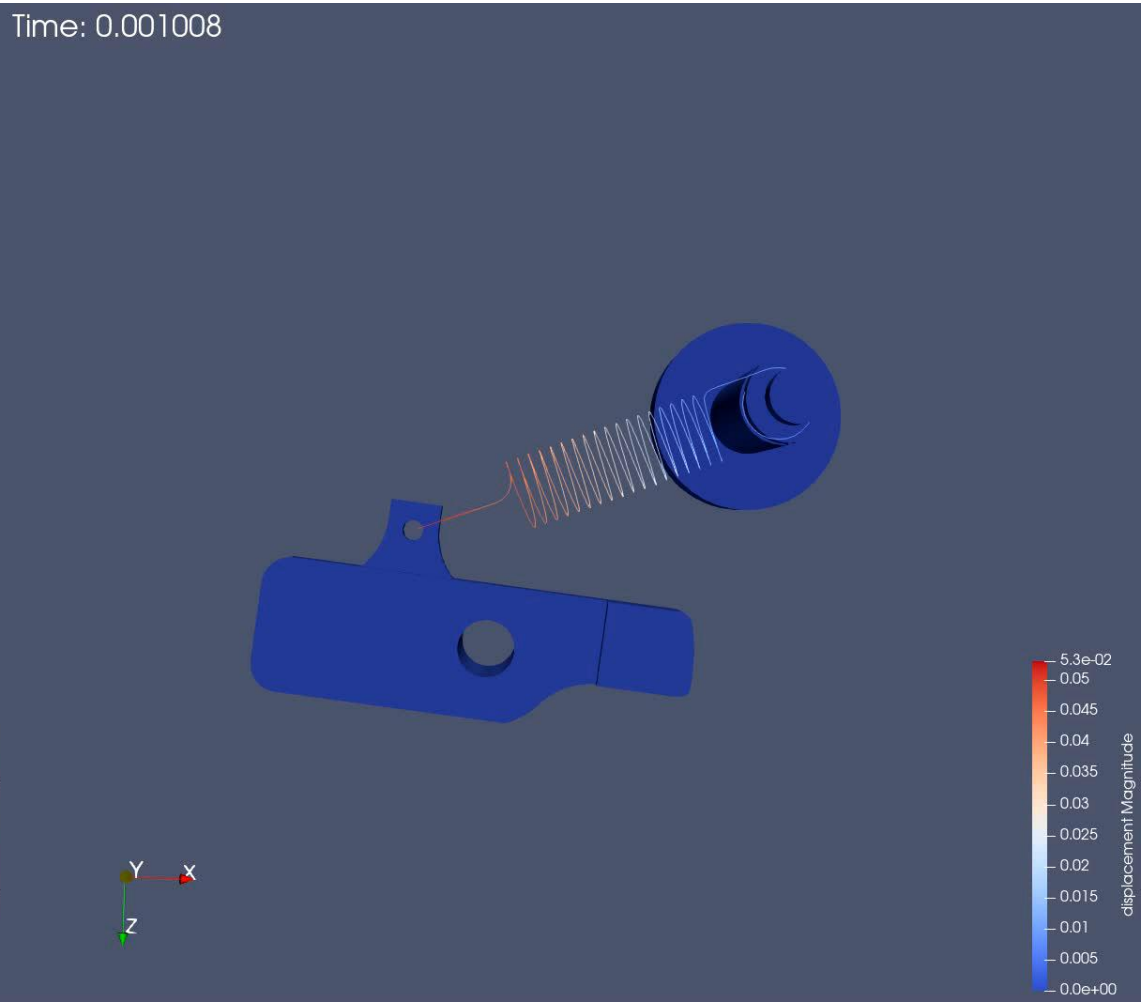
Force Extracted Where Spring Interacts with the Pin and Pawl

Pin-Spring-Pawl Environments

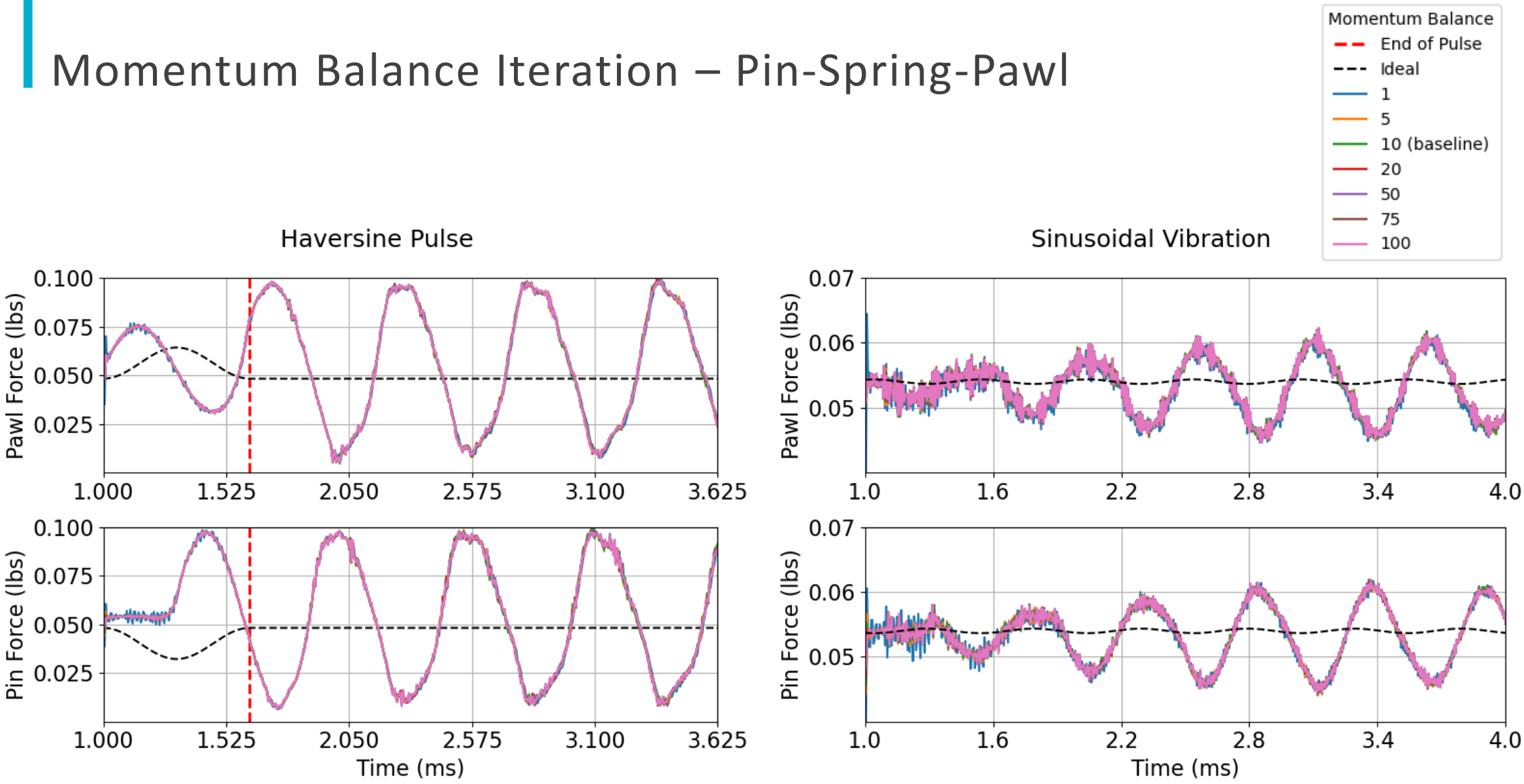
Haversine Shock



Sinusoidal Vibration

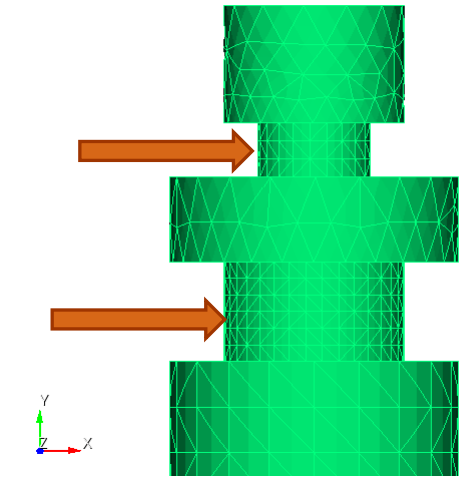
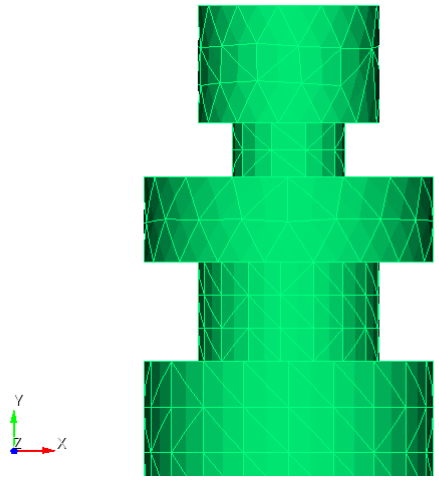
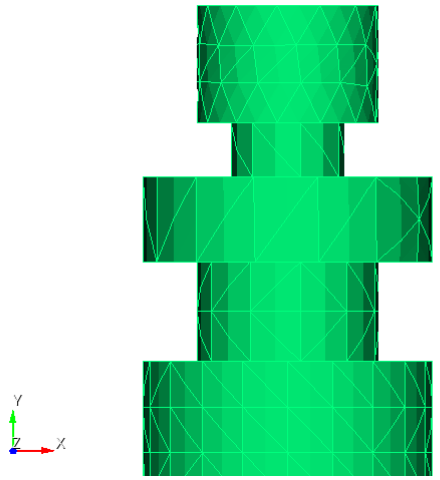
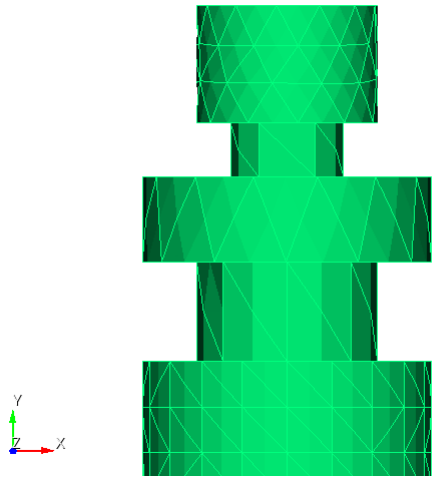
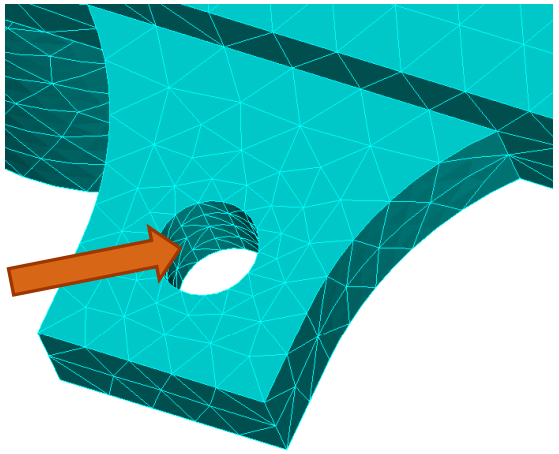
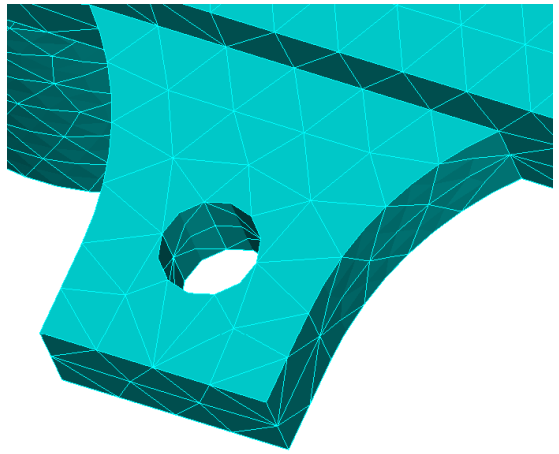
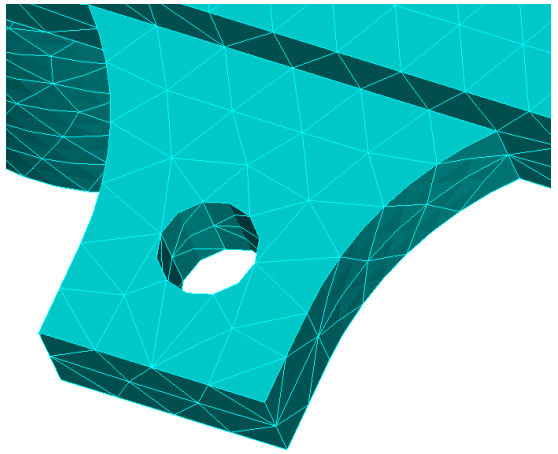
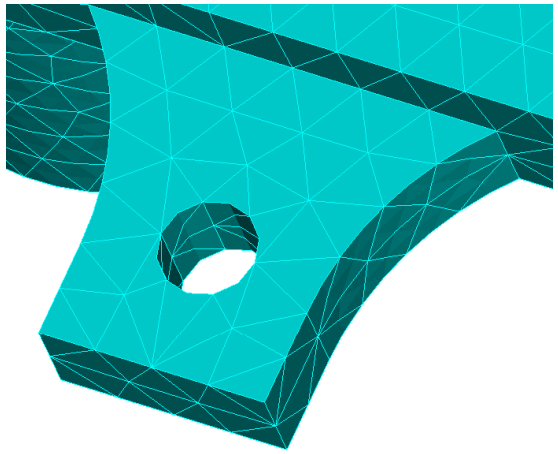


Momentum Balance Iteration – Pin-Spring-Pawl

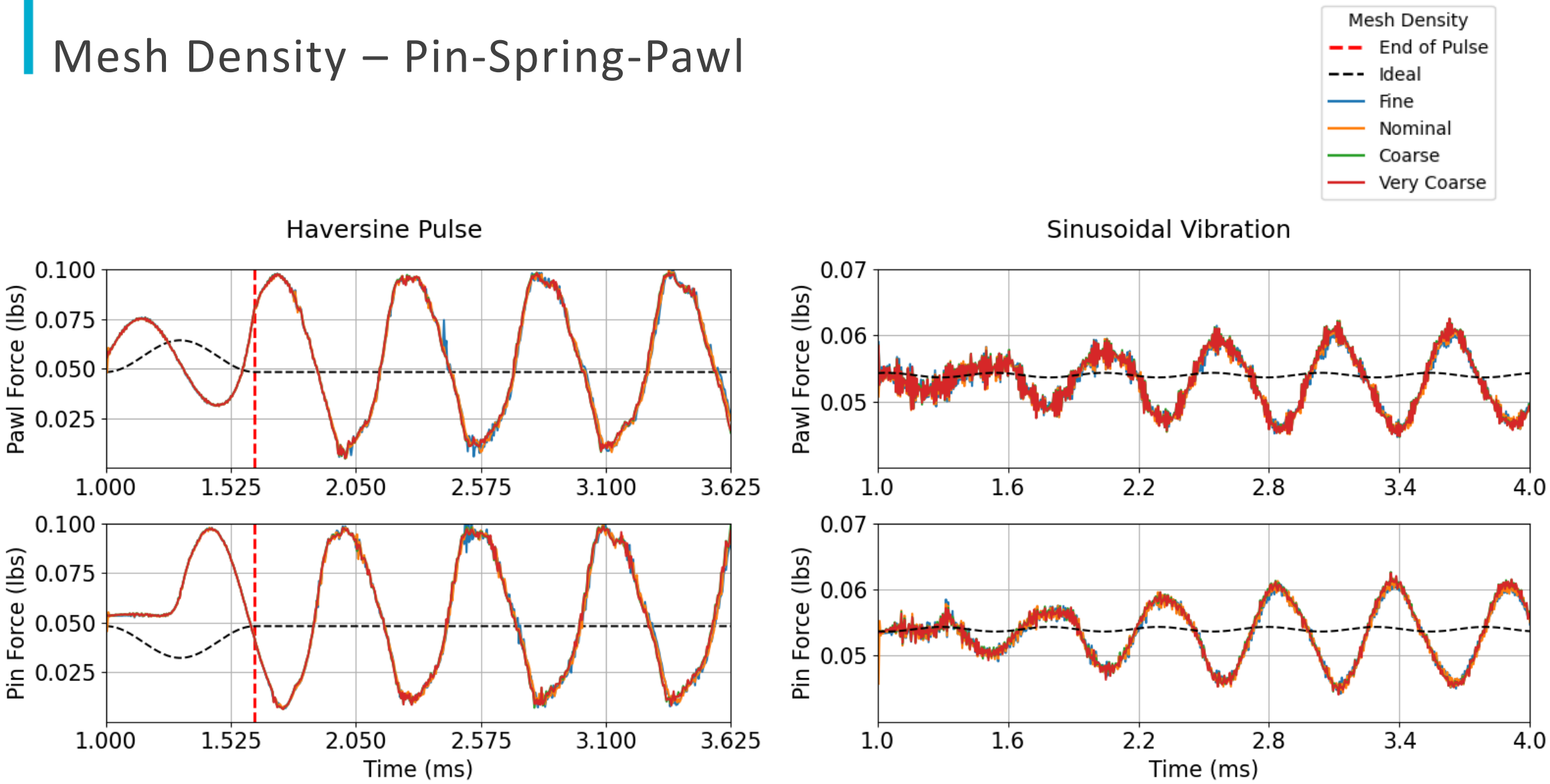


- Submodel is not sensitive to the number of momentum balance iterations used during analysis
- Ideal model needs adjustments to make accurate predictions of dynamics

Mesh Density – Pin-Spring-Pawl

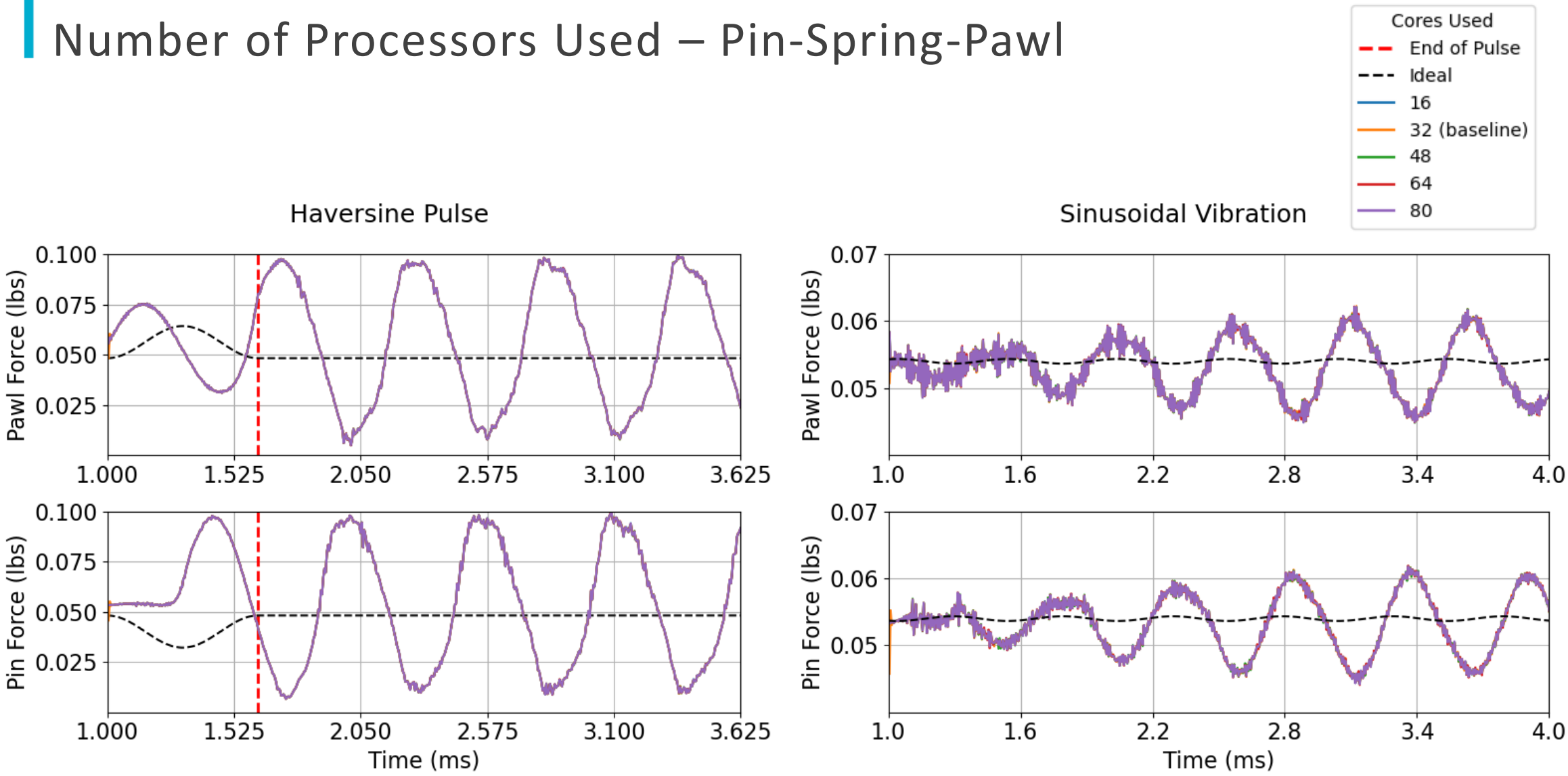
Fine	Nominal	Coarse	Very Coarse
			
			
35655 Elements	28827 Elements	27618 Elements	27410 Elements

Mesh Density – Pin-Spring-Pawl



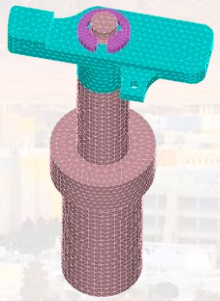
- Increasing or decreasing mesh density at the interface does not affect results
  - It could be possible to use larger elements to speed up simulation time

Number of Processors Used – Pin-Spring-Pawl



- Submodel is insensitive to the number of cores used

Pin-spring-pawl



Pin-pawl

Pawl-gear

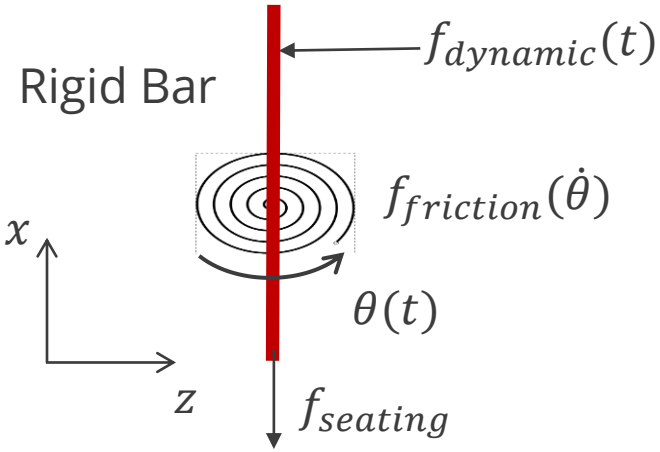
Submodel – Pin-Pawl

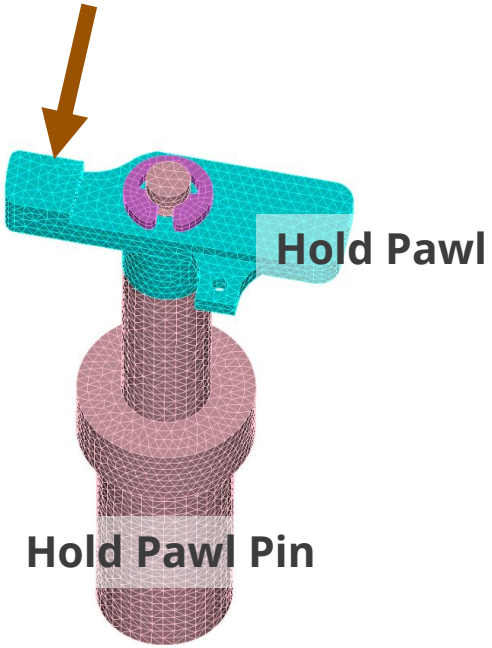
Pin-Pawl Submodel

Parts:
- Hold pawl
- Hold pawl pin

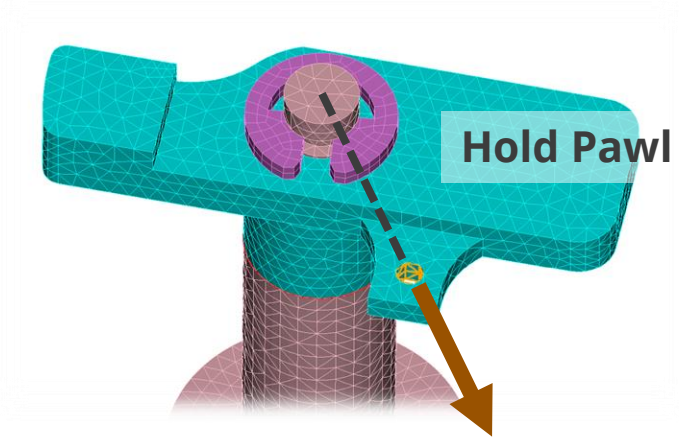
Quantities of interest:
- Contact force between pin and pawl
- Pawl rotation angle

Idealized Model:
- Center of mass and inertia from FE model

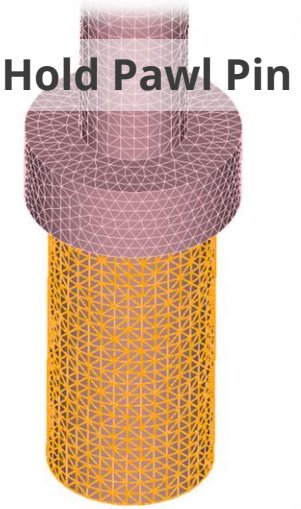




Pressure (shock or vibration) applied to side of pawl



Constant seating force applied in one direction



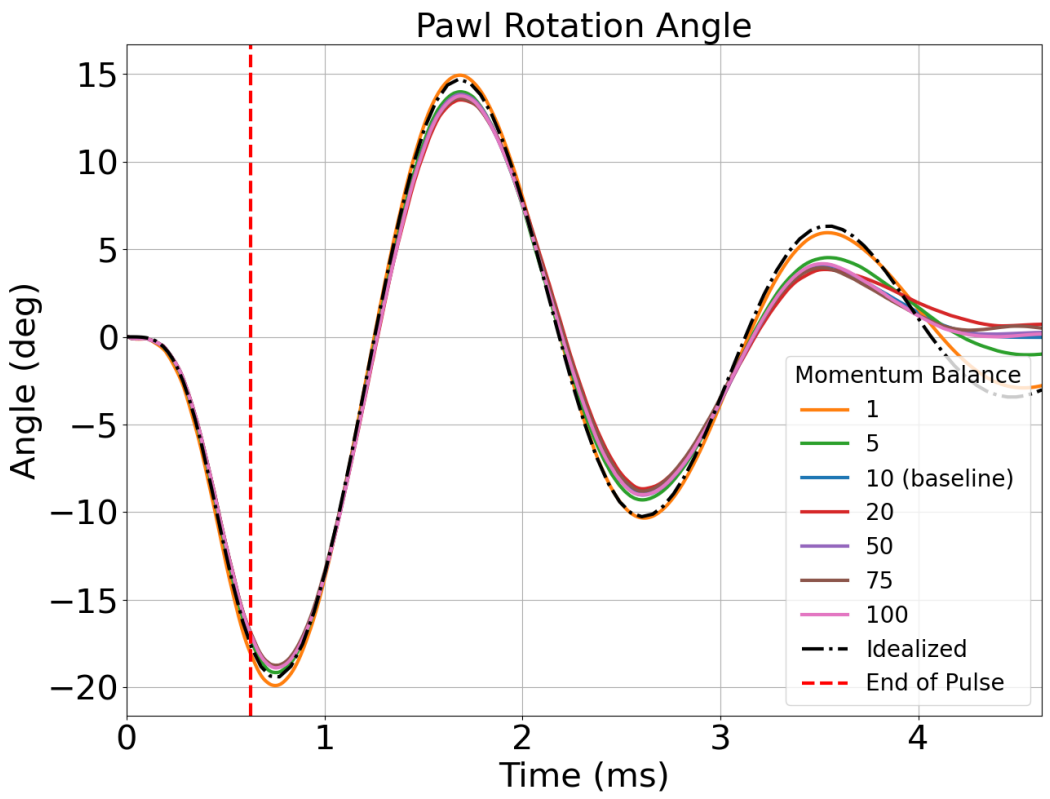
Base of pin is fixed in all DOF's

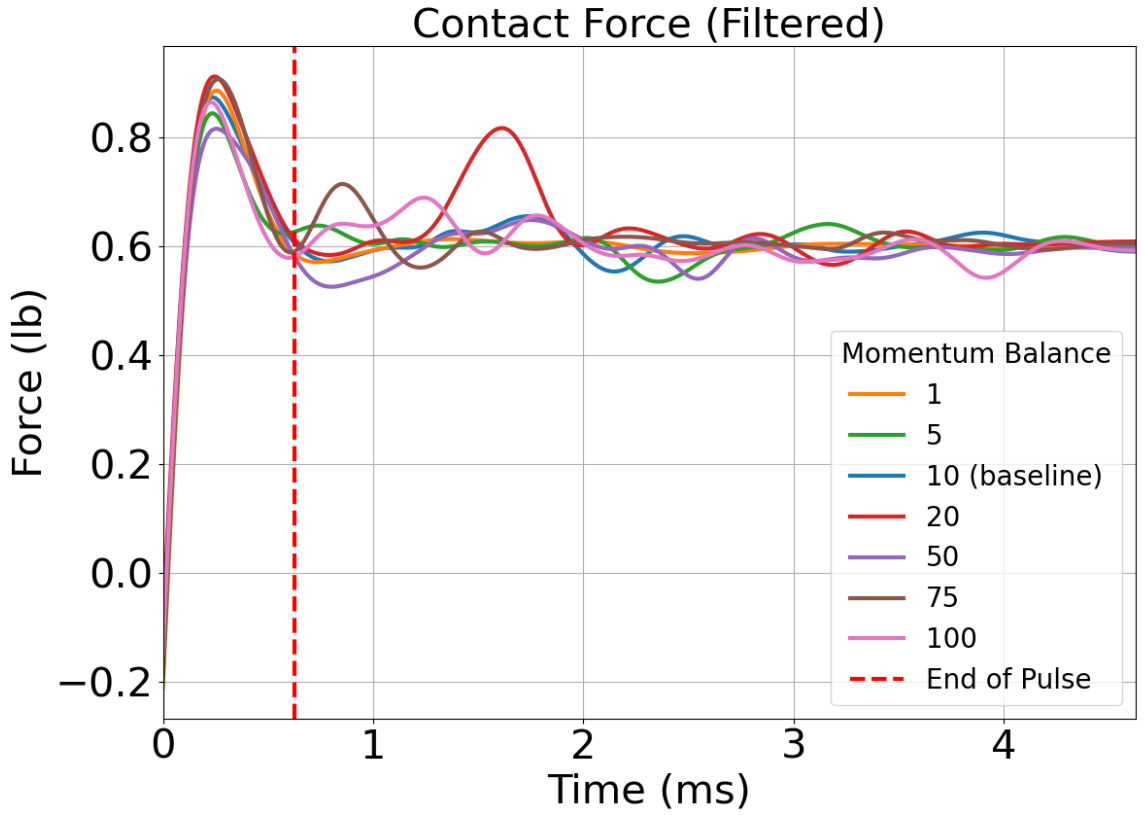
Pin-Pawl Environments

Haversine Shock

Sinusoidal Vibration

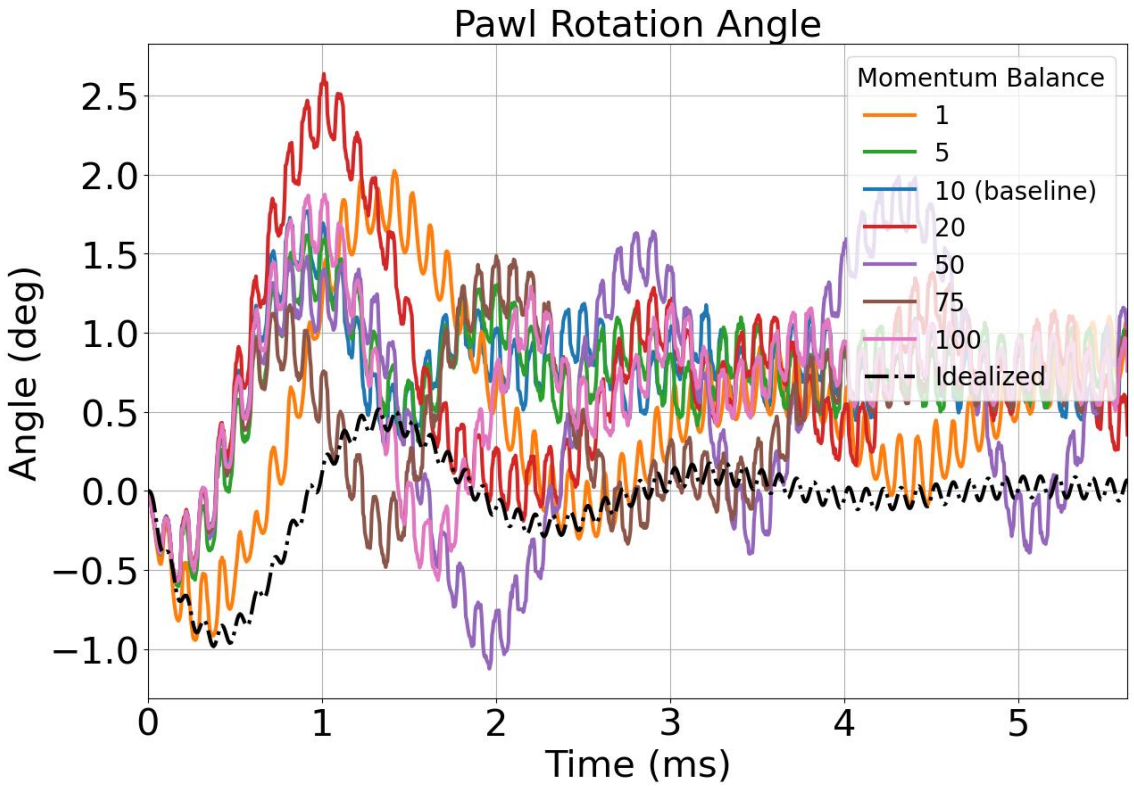
Momentum Balance Iteration – Pin-Pawl (Haversine Shock)

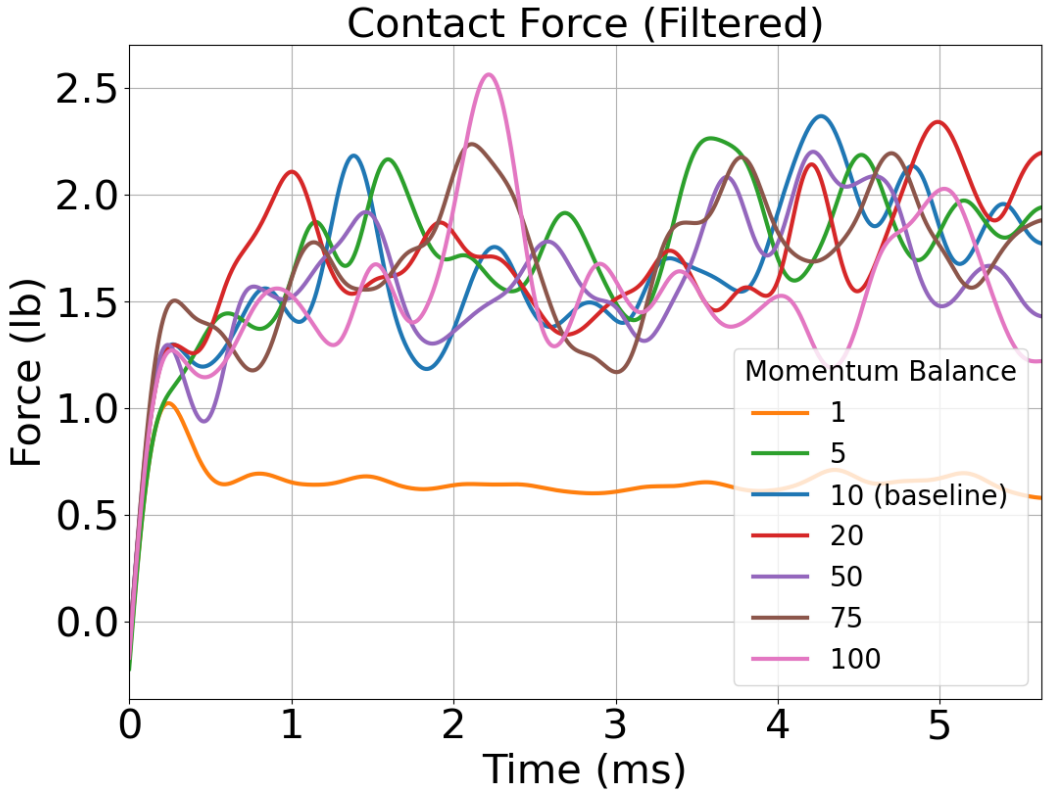




- Rotations agree with idealized model
- Rotations diverge after shock as contact and friction dictate motion
- Contact force is erratic

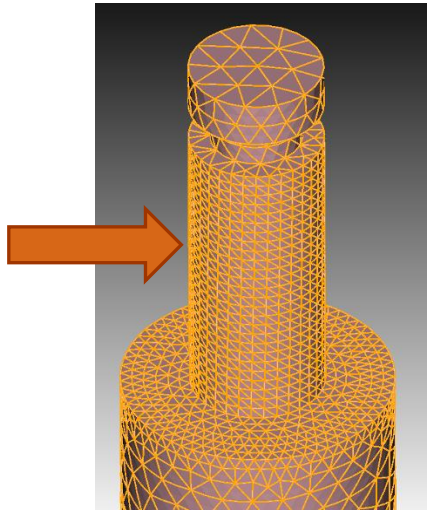
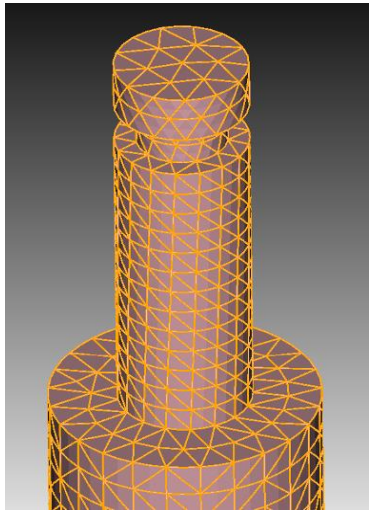
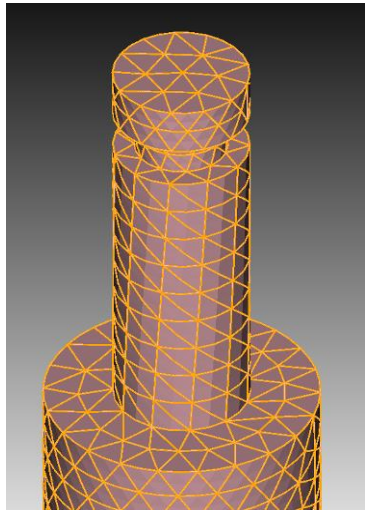
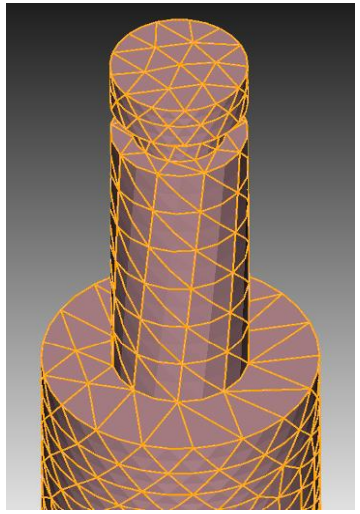
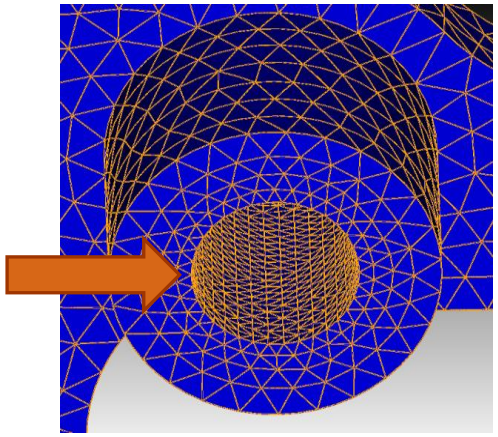
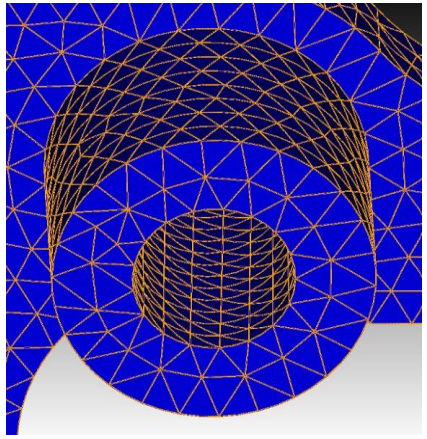
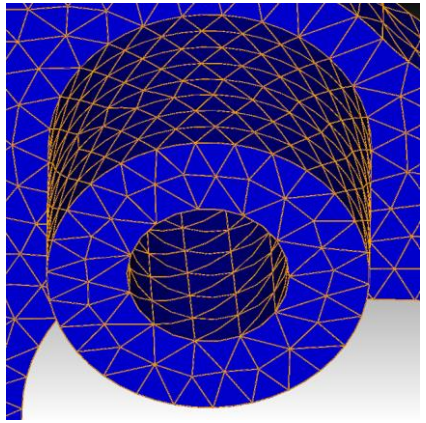
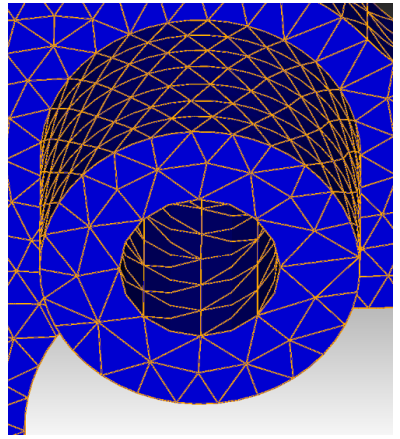
Momentum Balance Iteration – Pin-Pawl (Sinusoidal Vibration)



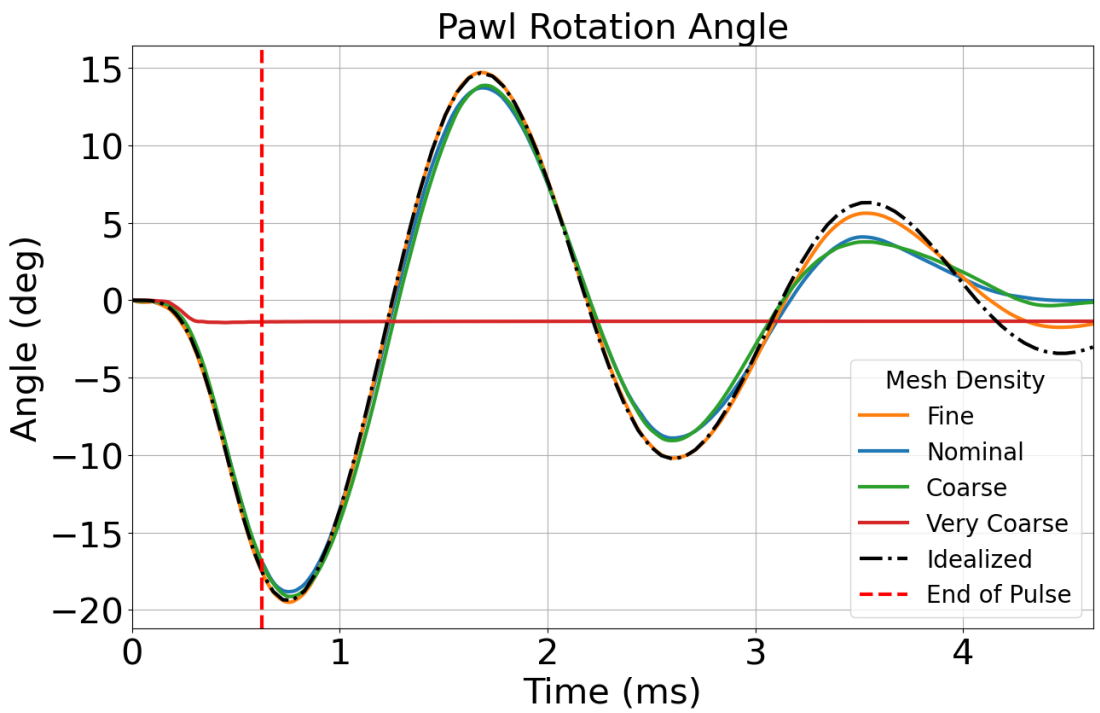


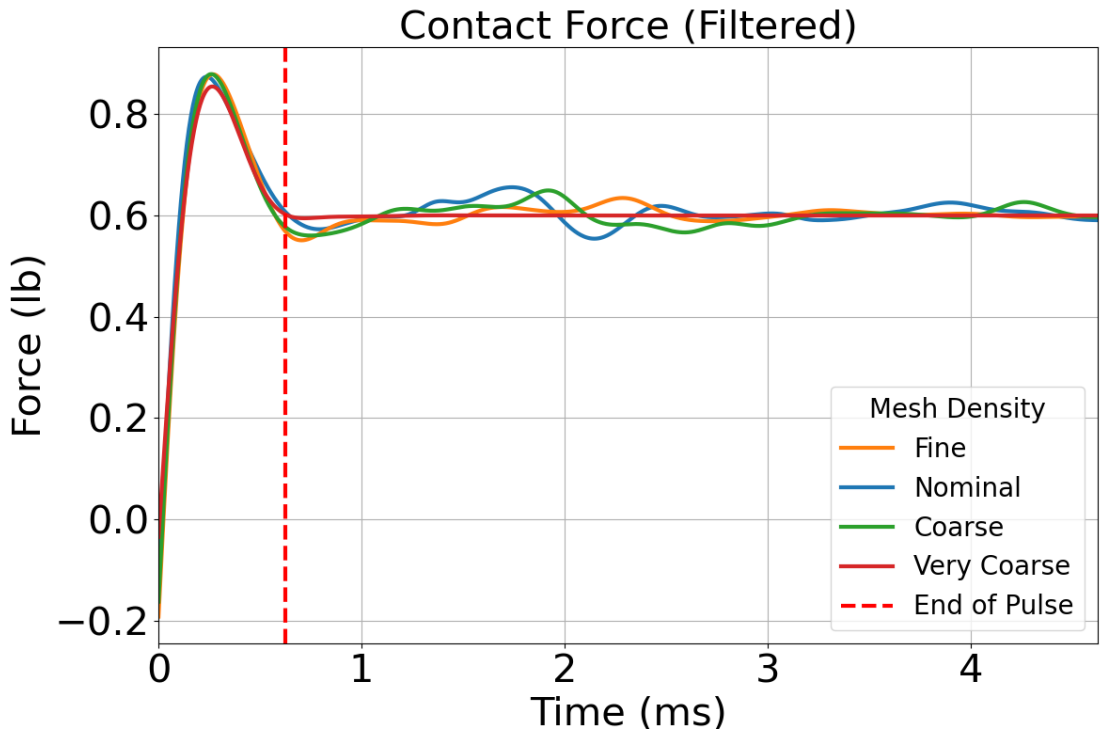
- Larger amplitude compared to idealized model, but otherwise similar profile
- Rotation and contact force are erratic
- Sensitive to momentum balance iteration

Mesh Density – Pin-Pawl

Fine	Nominal	Coarse	Very Coarse
			
			
65763 Elements	46310 Elements	45025 Elements	43909 Elements

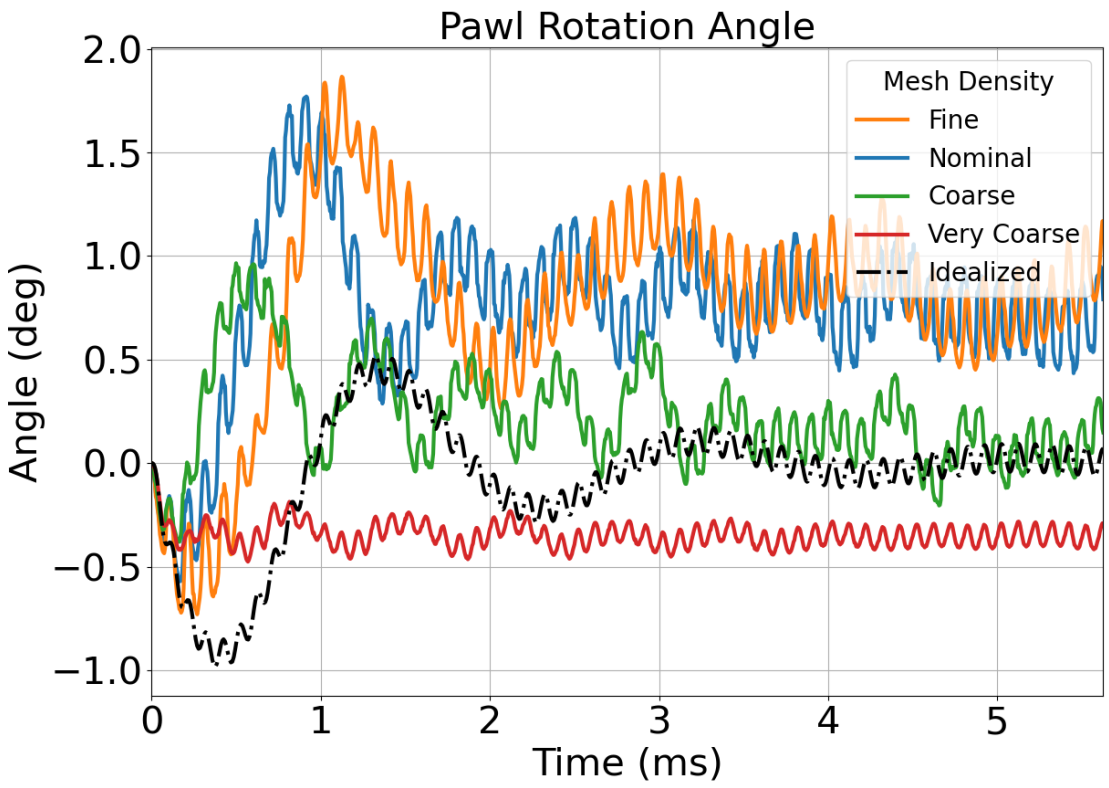
Mesh Density – Pin-Pawl (Haversine Shock)

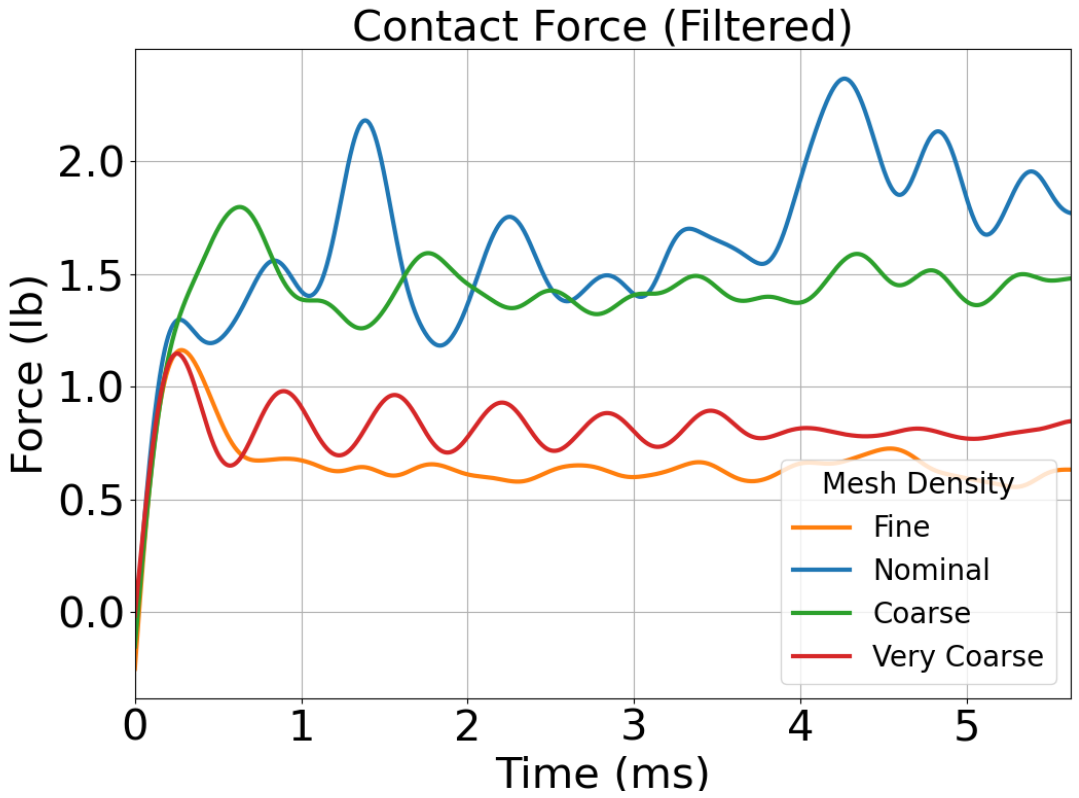




- Rotations roughly agree with idealized model
- Rotation lock-up for very coarse mesh density
- Rotations diverge after shock
- Contact force is erratic

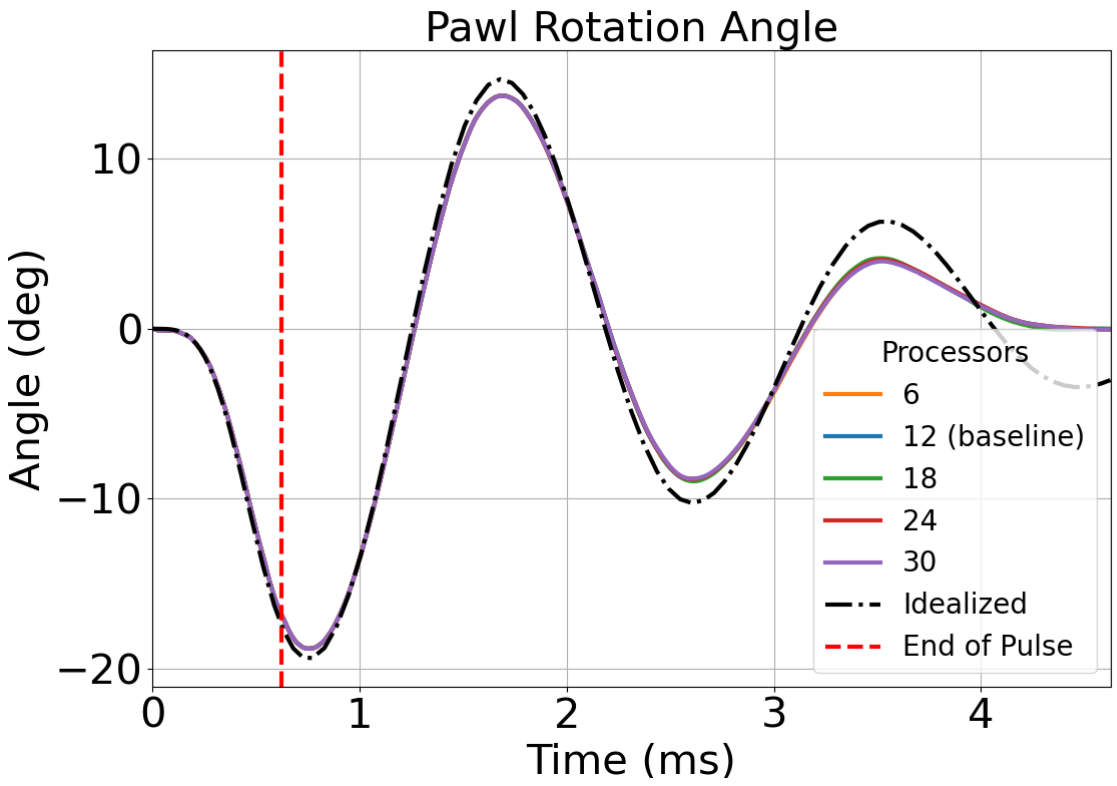
Mesh Density – Pin-Pawl (Sinusoidal Vibration)

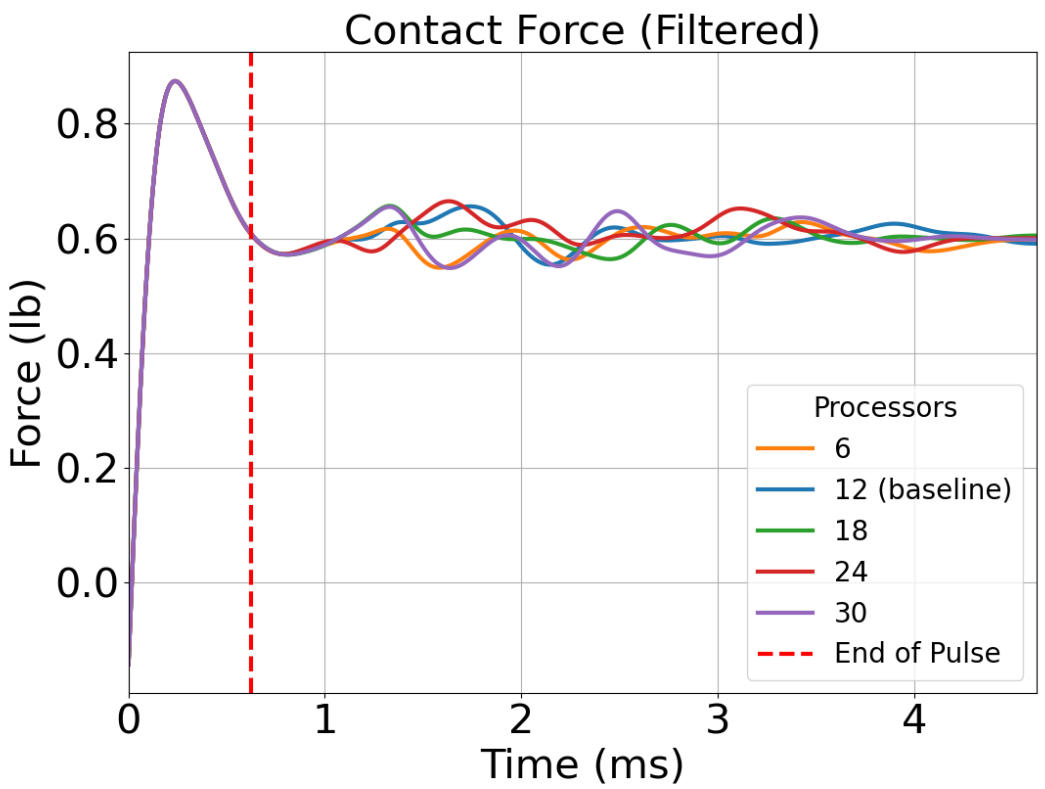




- Similar to momentum balance, overall rotation trend is similar but different in damping and amplitude to idealized model
- Less rotation allowed for very coarse mesh density (potentially a lock-up)
- More rotation allowed as element becomes finer
- Contact force does not converge

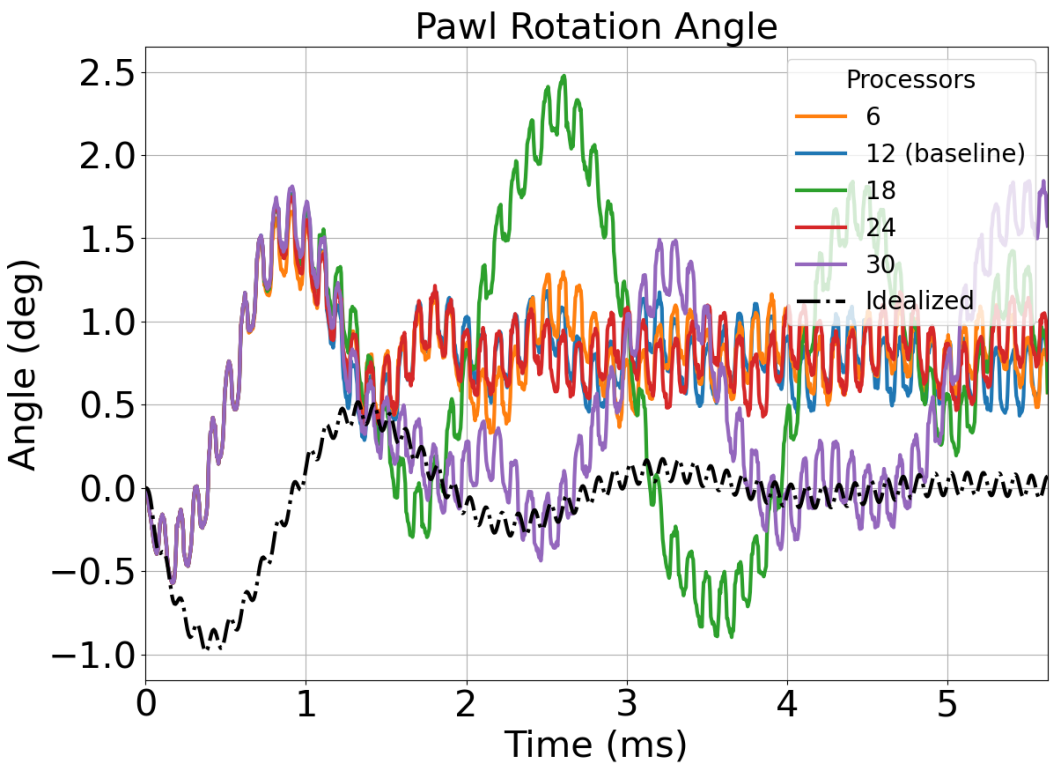
Number of Processors Used – Pin-Pawl (Haversine Shock)

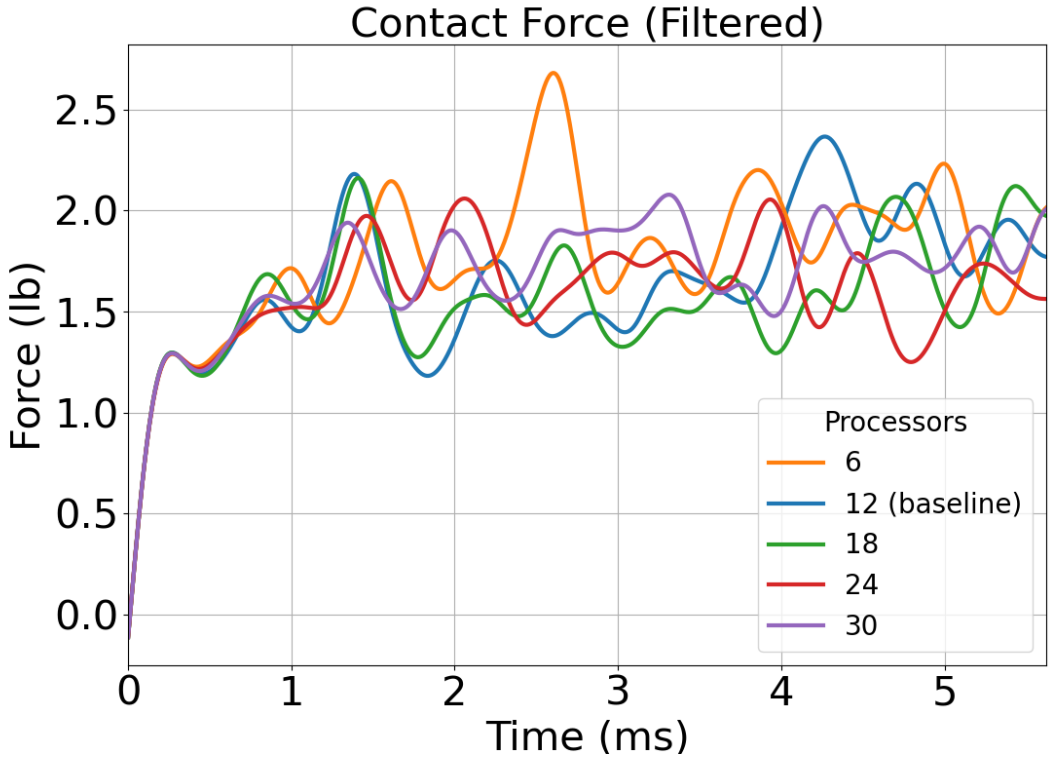




- Very small difference in rotation and larger difference in contact force after less than 1 ms
- Less sensitive to processor count

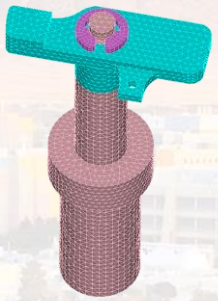
Number of Processors Used – Pin-Pawl (Sinusoidal Vibration)





- Similar to haversine, results diverge after less than 1 ms
- Difference is much larger than that of haversine shock
- Sensitive to processor count

Pin-spring-pawl



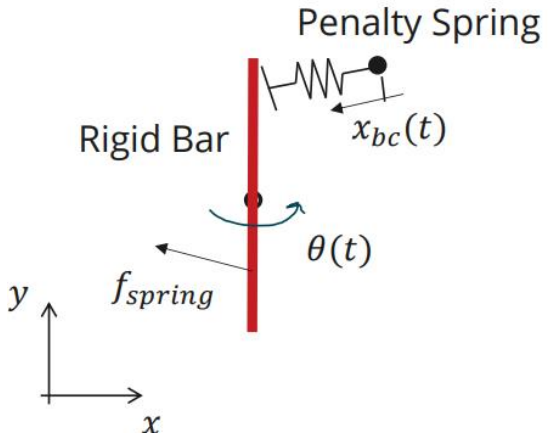
Pin-pawl

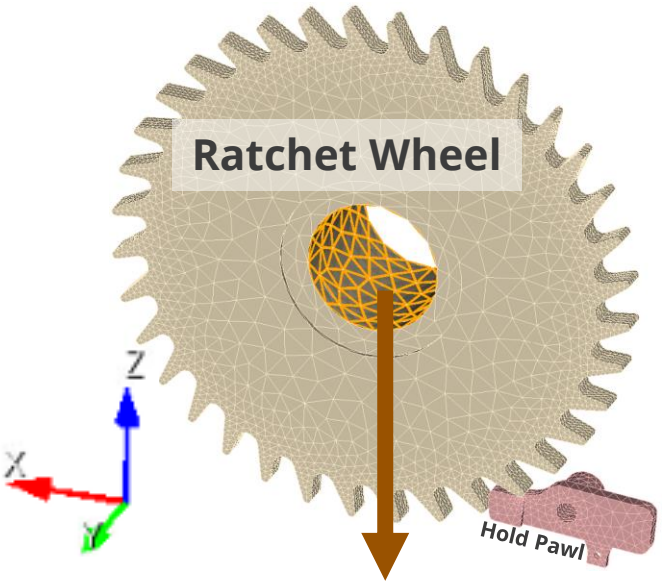
Pawl-gear

Submodel – Pawl-gear

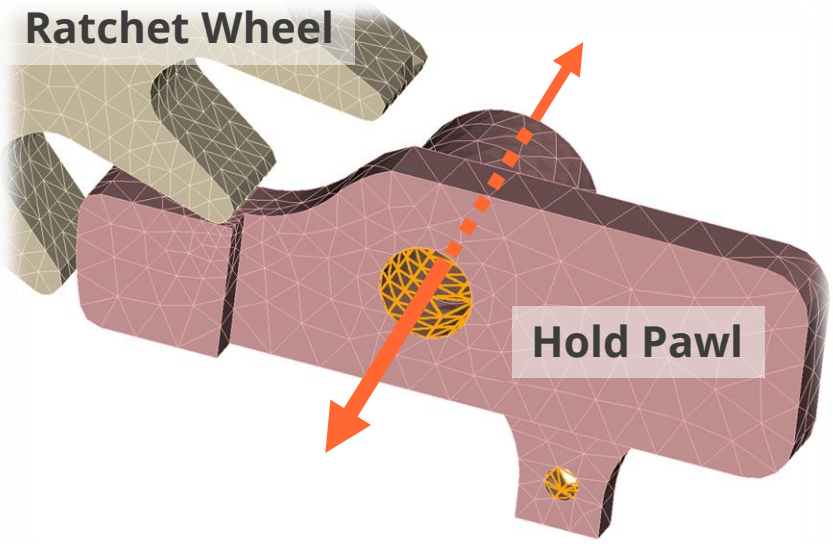
Pawl-Gear Submodel

- **Boundary Conditions**
  - Ratchet wheel is pushed and contacts hold pawl
  - Pseudo spring force rotates pawl into ratchet wheel
- **Quantity of Interest**
  - Part rotation of the hold pawl over time

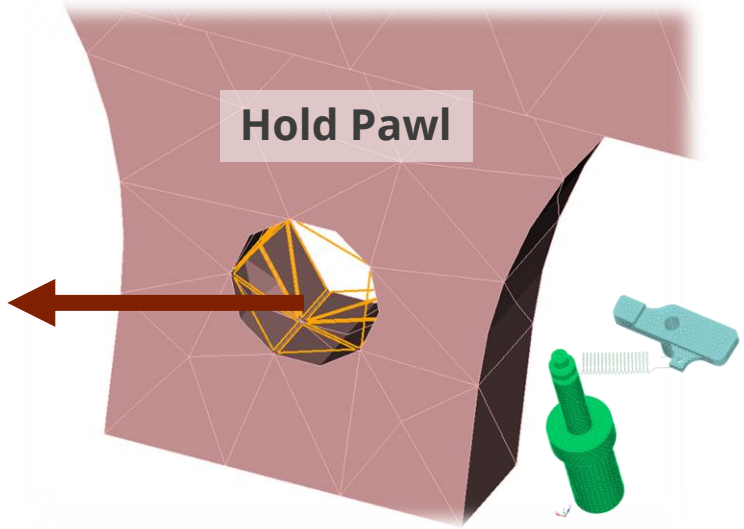




Prescribed displacement in Z direction (shock or vibration)

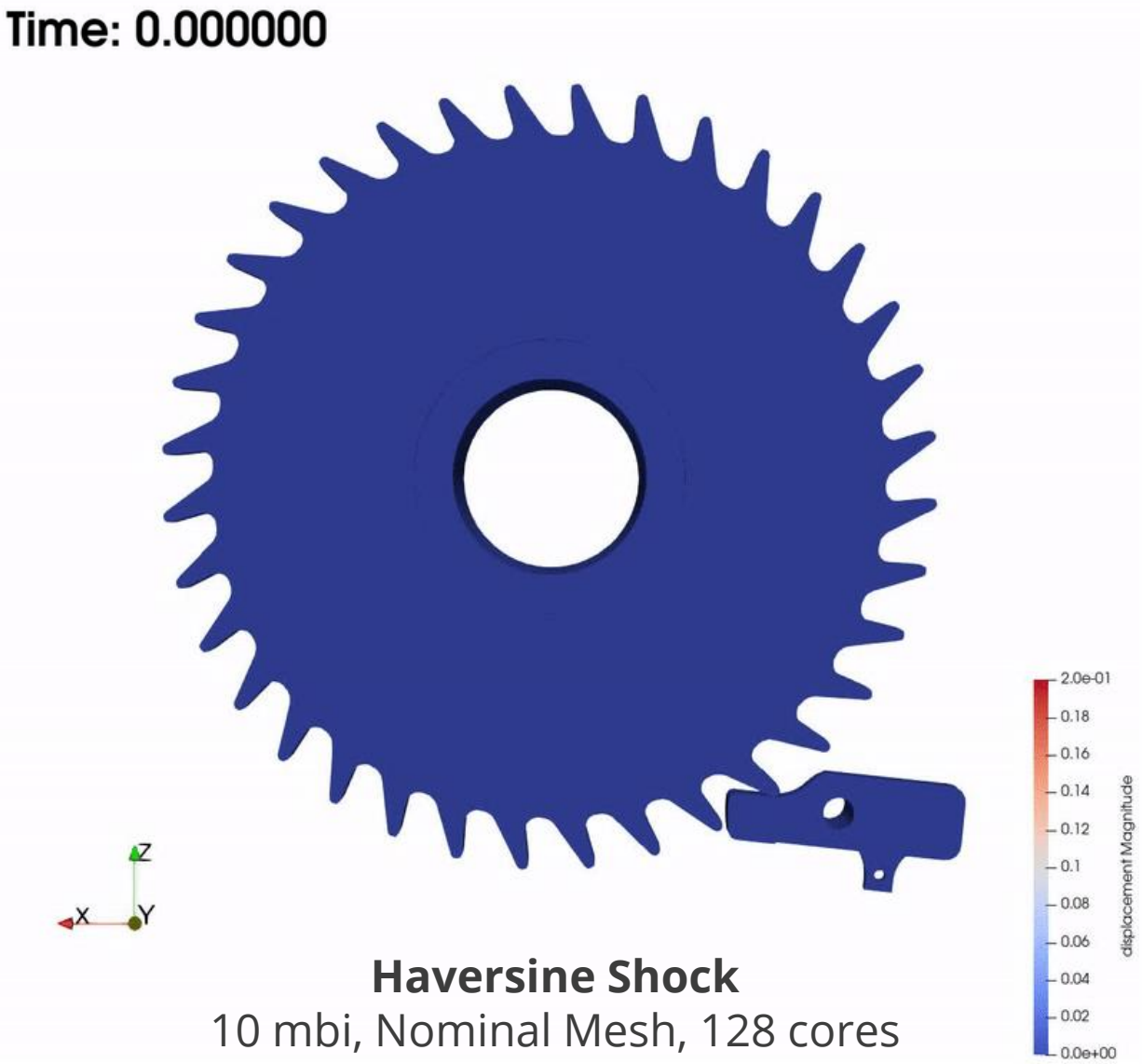


Pawl is fixed in radial and axial direction

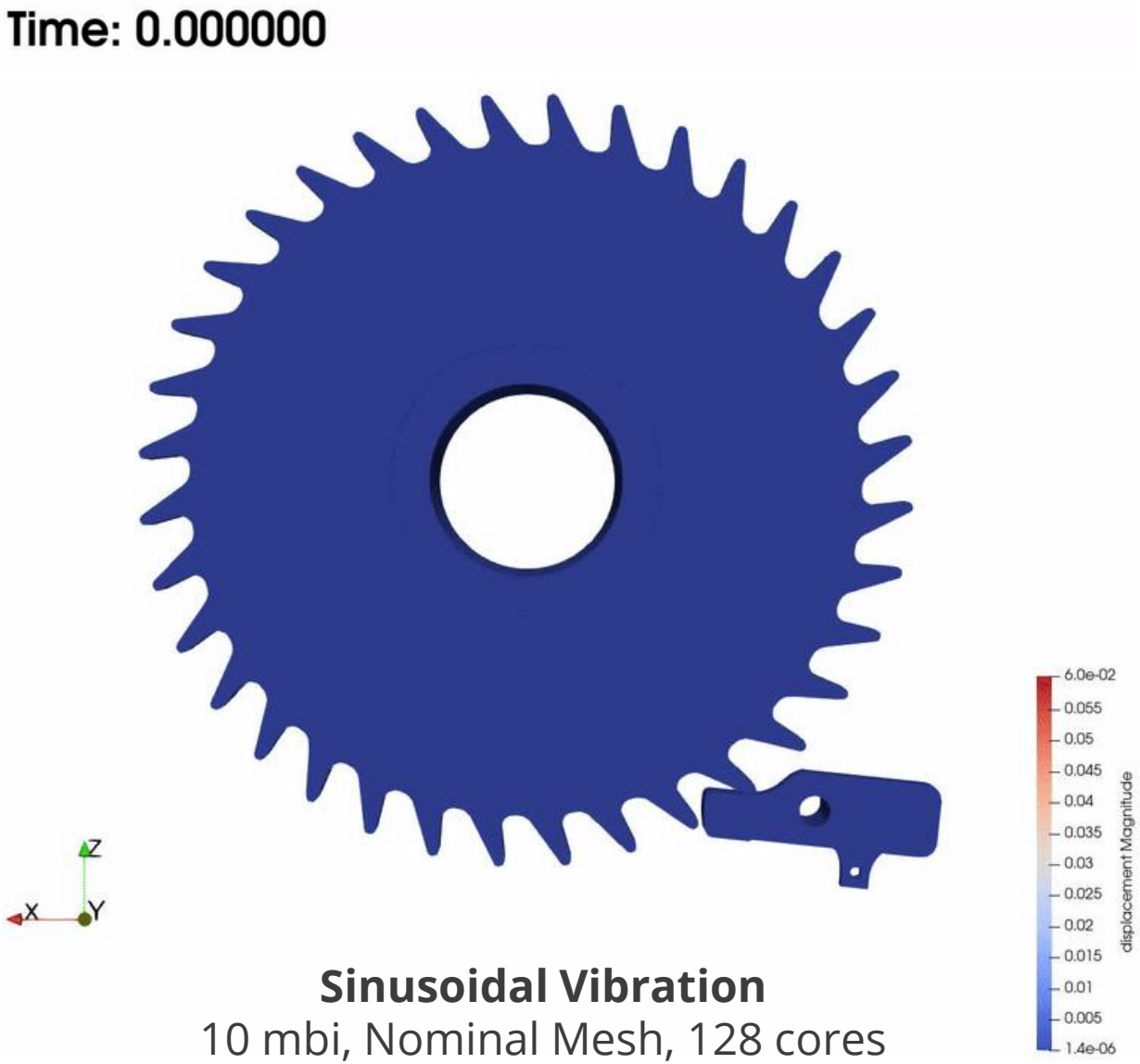


“Pseudo spring force“ along pawl spring direction

Pawl-Gear Submodel

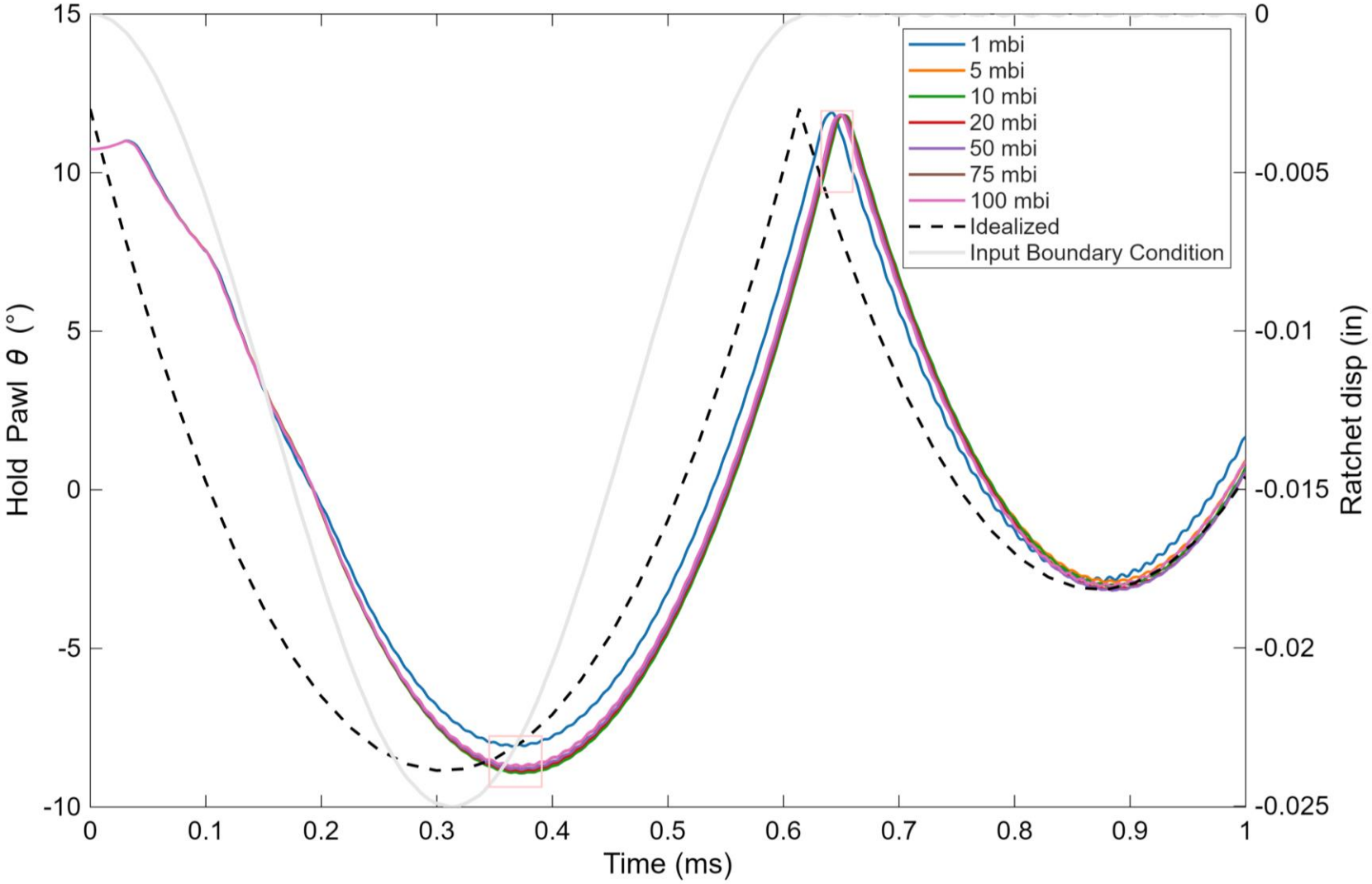


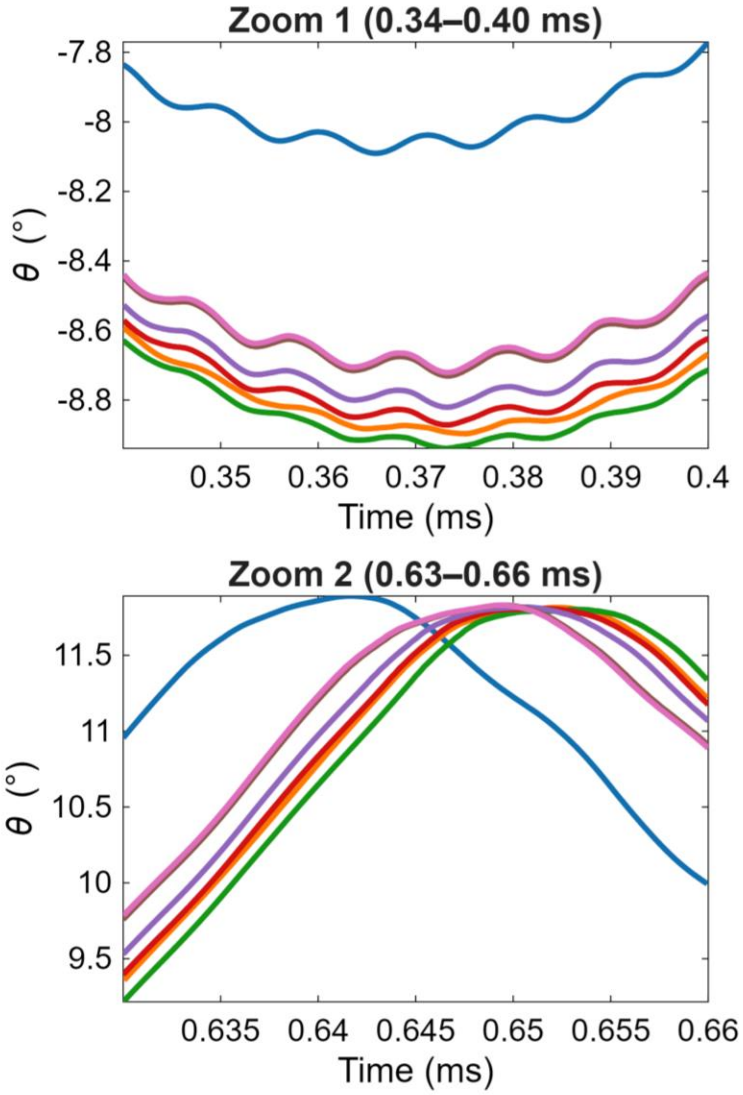
Haversine Shock
10 mbi, Nominal Mesh, 128 cores



Sinusoidal Vibration
10 mbi, Nominal Mesh, 128 cores

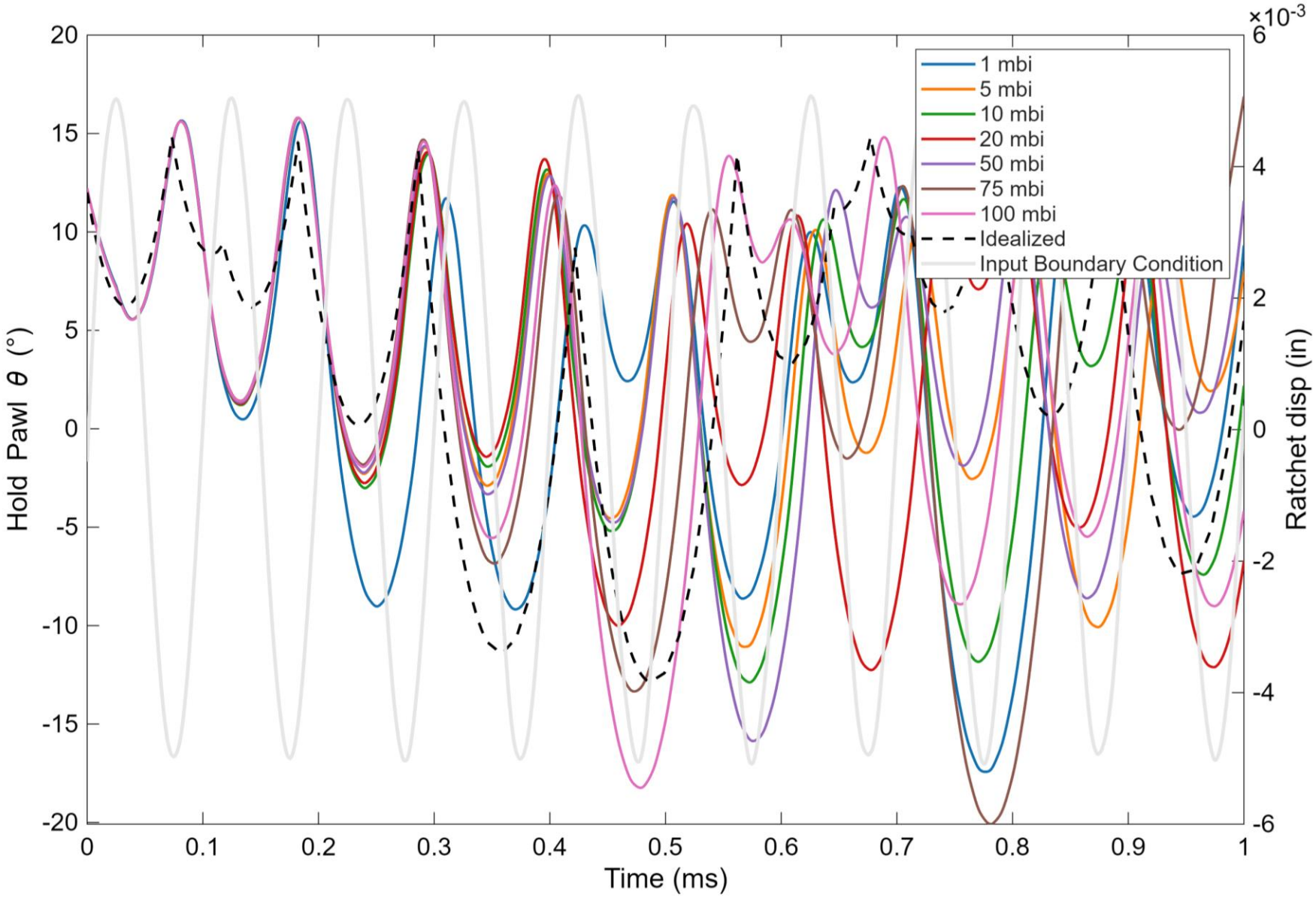
Momentum Balance Iteration – Pawl-Gear (Haversine Shock)





Haversine shock settings: 0.025 inch amplitude, 0.5 lbf pseudo force, 100% mesh density, 128 cores

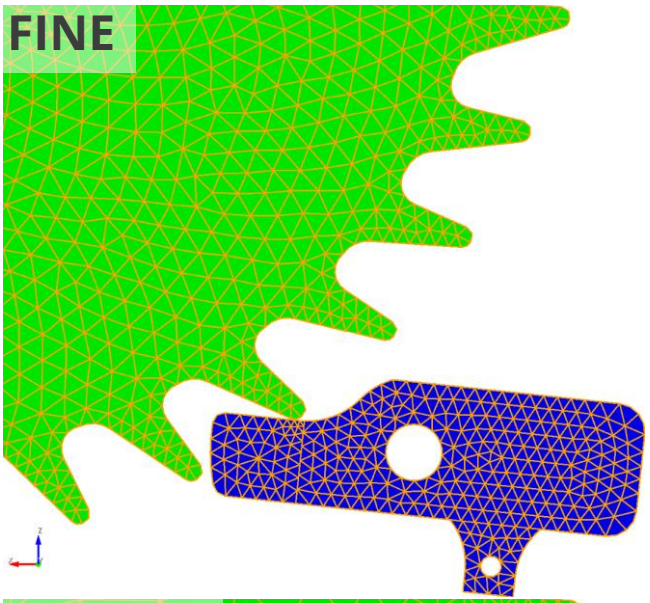
Momentum Balance Iteration – Pawl-Gear (Sinusoidal Vibration)

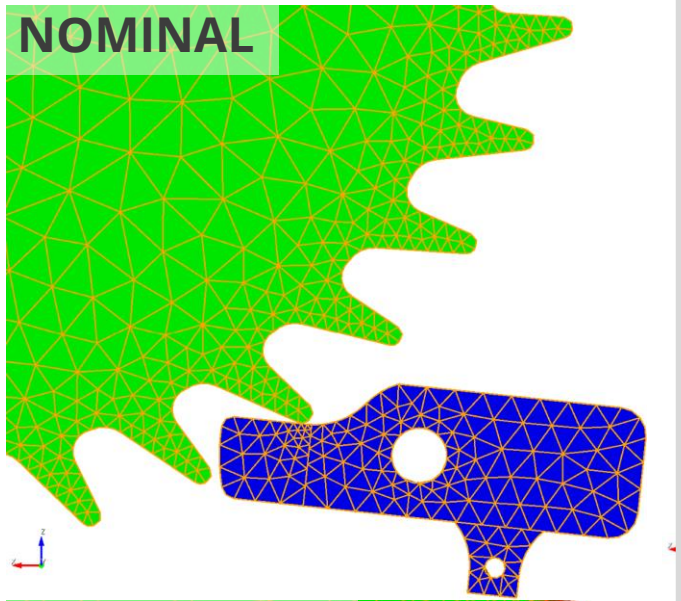


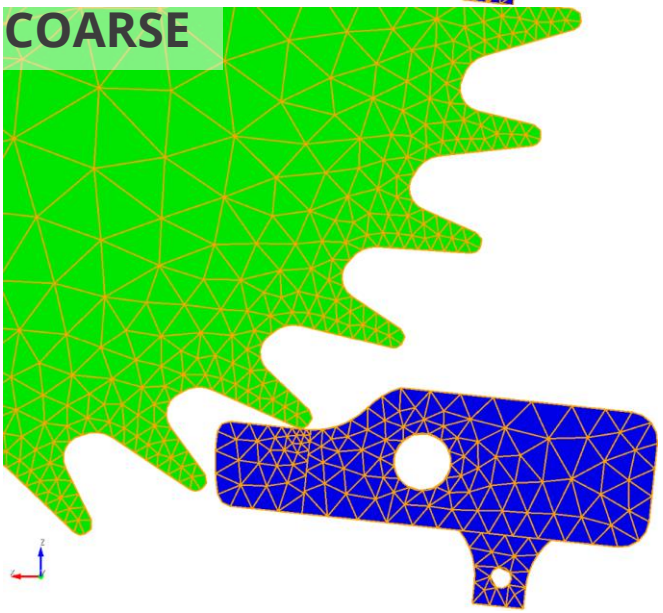
Sinusoidal vibration settings: 10 kHz, 0.005 inch amplitude, 13 lbf pseudo force, 100% mesh density, 128 cores

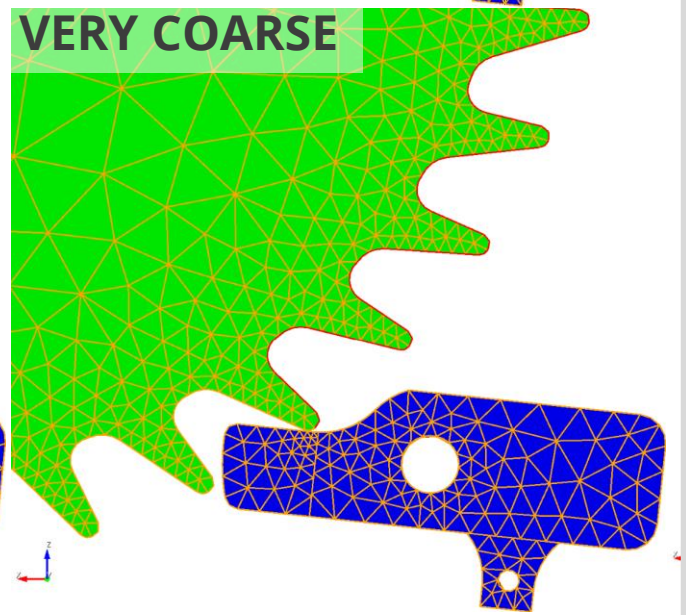
Mesh Density – Pawl-Gear

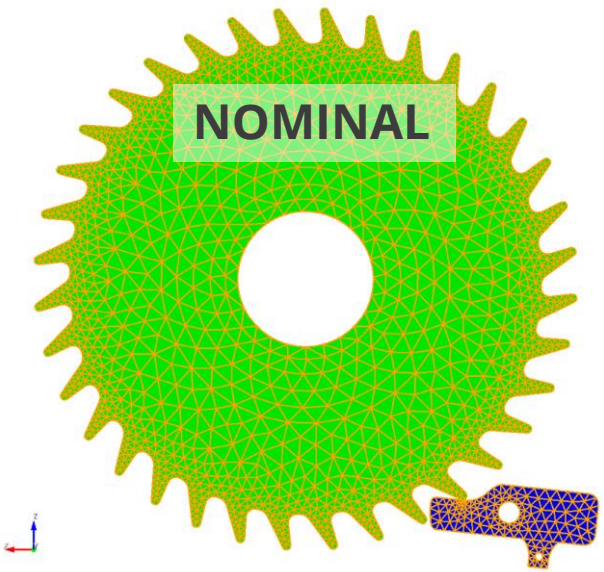
Element Count	FINE: 89705	NOMINAL: 36467
	COARSE: 32736	VERY COARSE: 32106

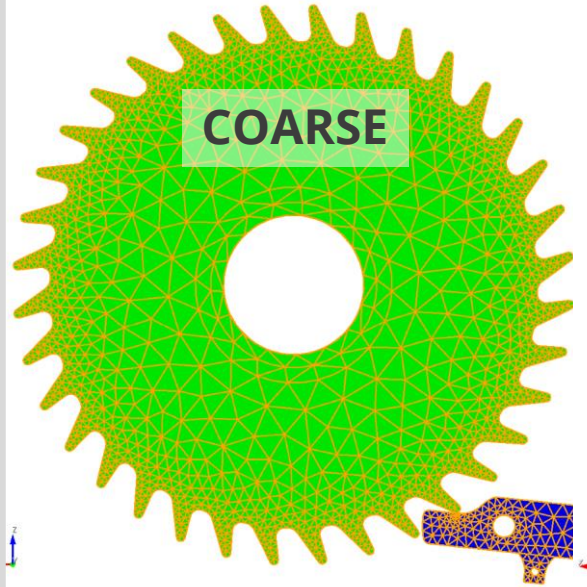


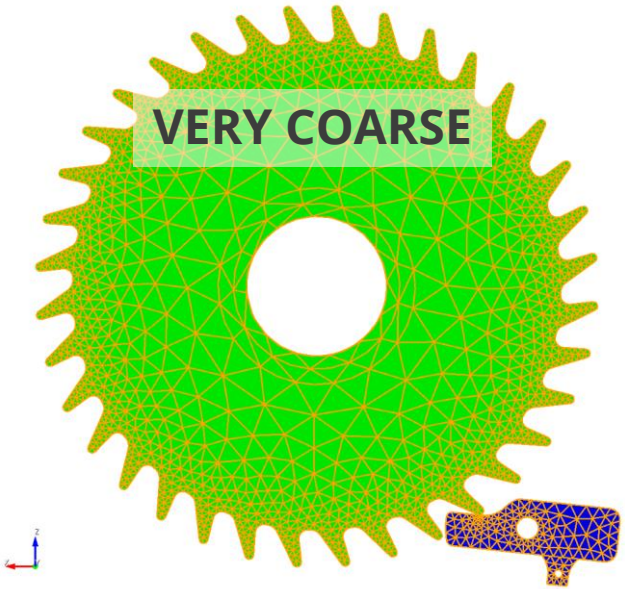




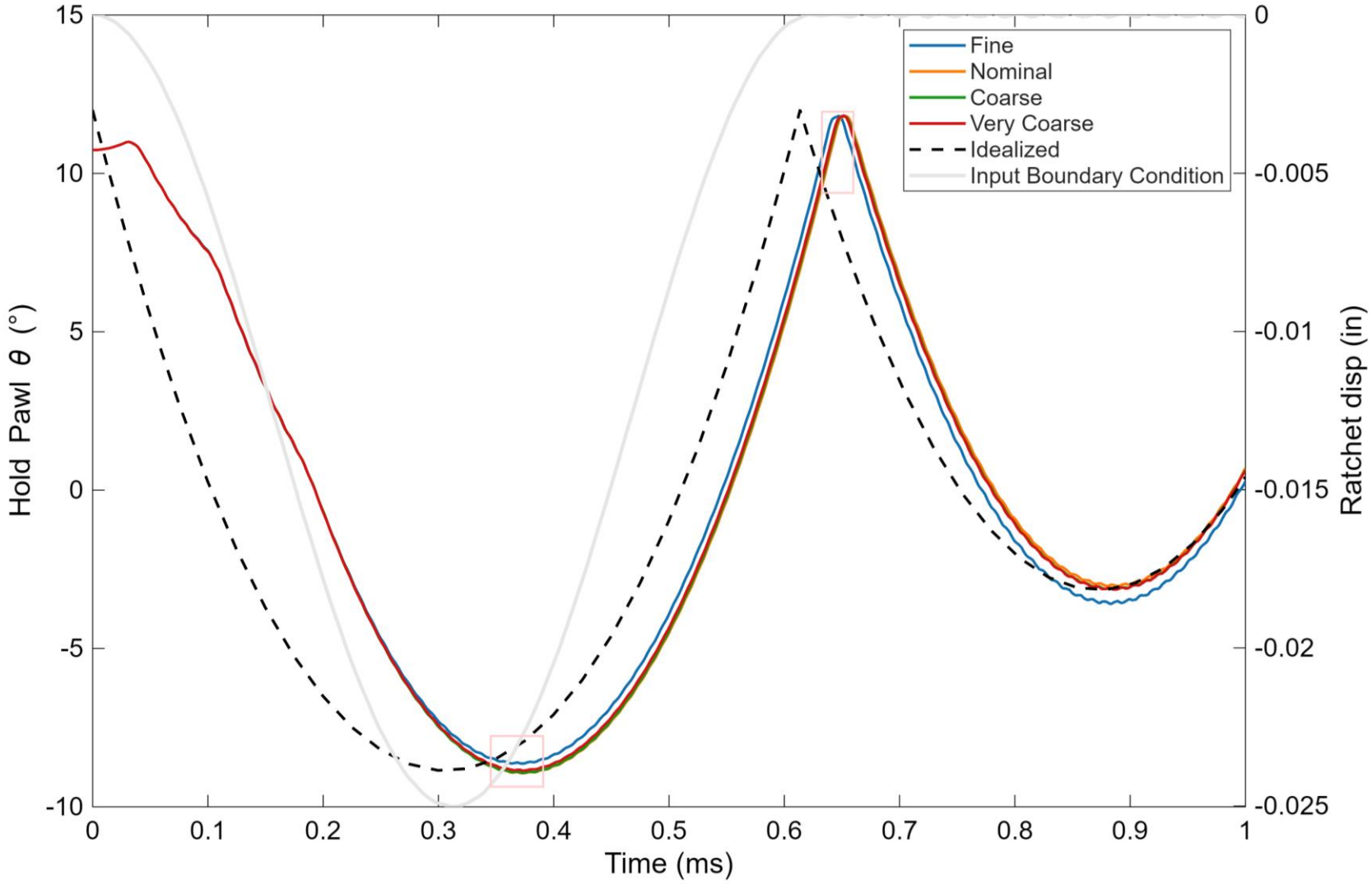


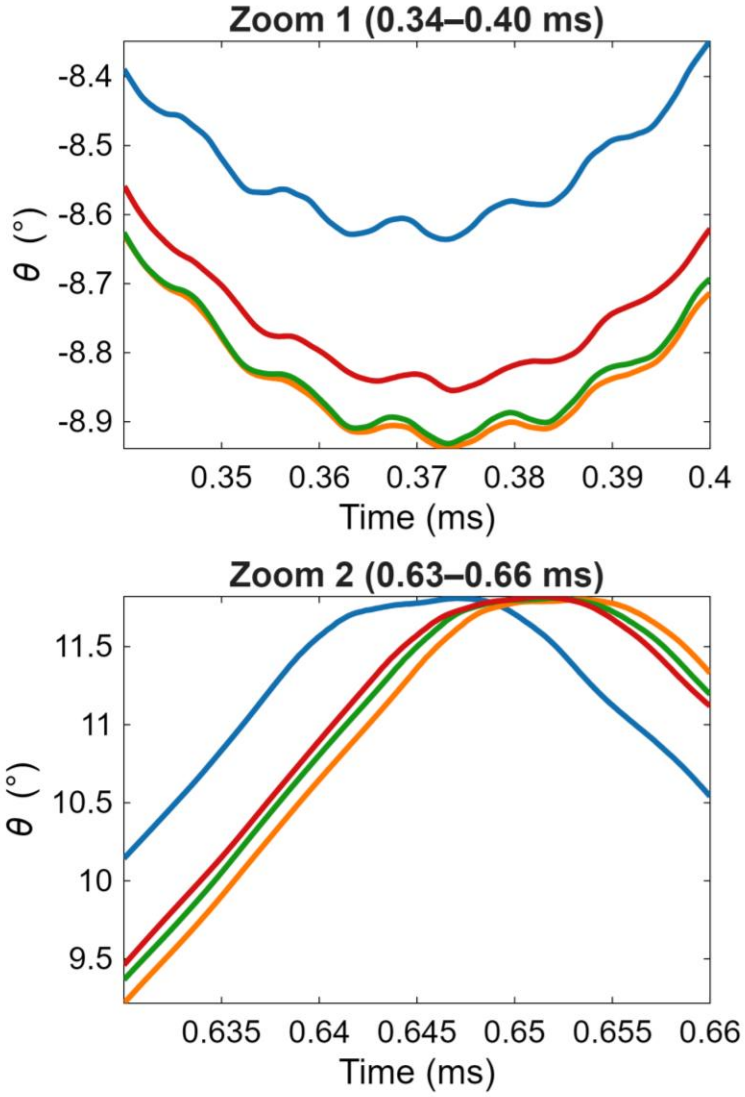






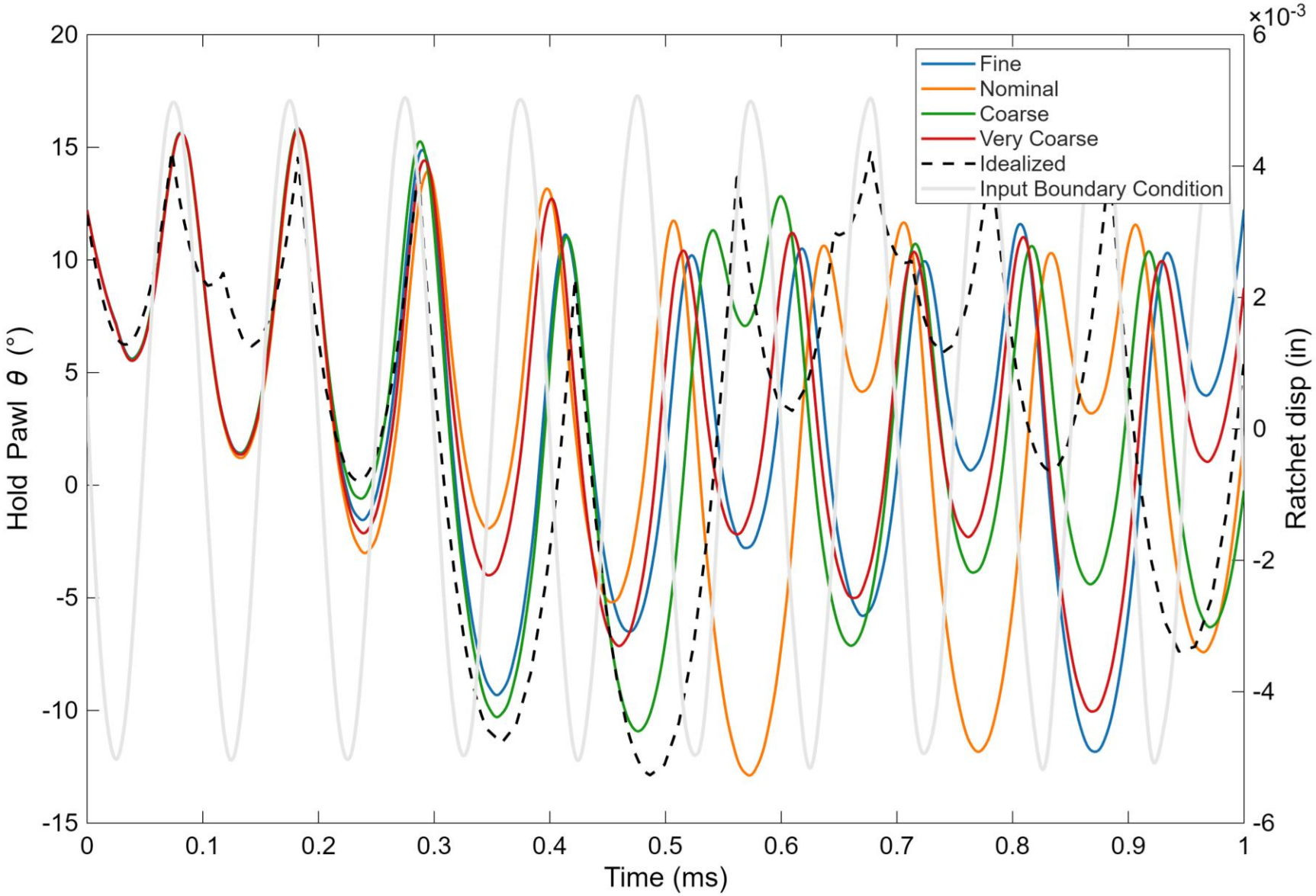
Mesh Density – Pawl-Gear (Haversine Shock)





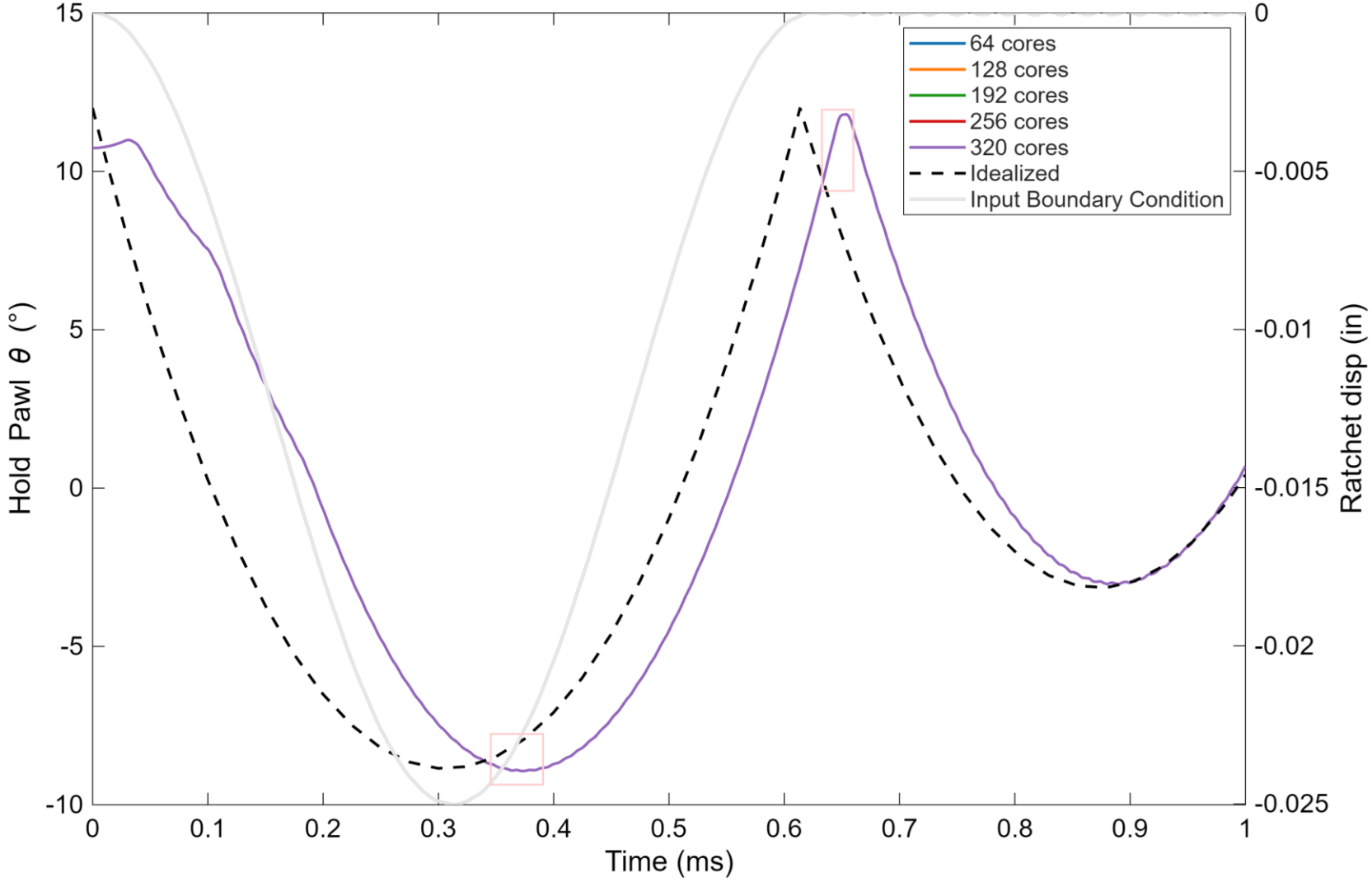
Haversine shock settings: 0.025 inch amplitude, 0.5 lbf pseudo force, 10 mbi, 128 cores

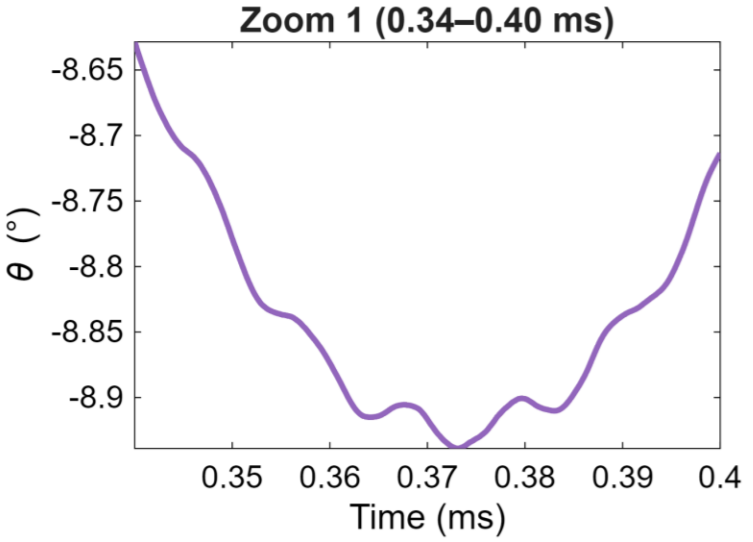
Mesh Density – Pawl-Gear (Sinusoidal Vibration)

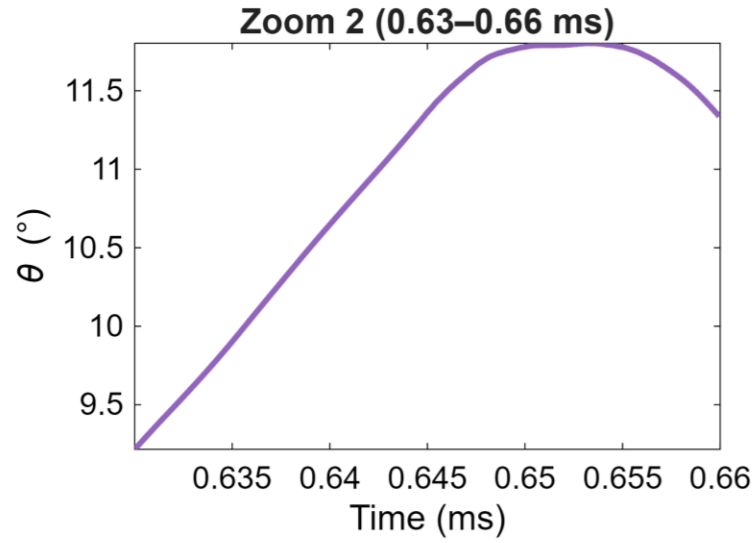


Sinusoidal vibration settings: 10 kHz, 0.005 inch amplitude, 13 lbf pseudo force, 10 mbi, 128 cores

Processor Count – Pawl-Gear (Haversine Shock)

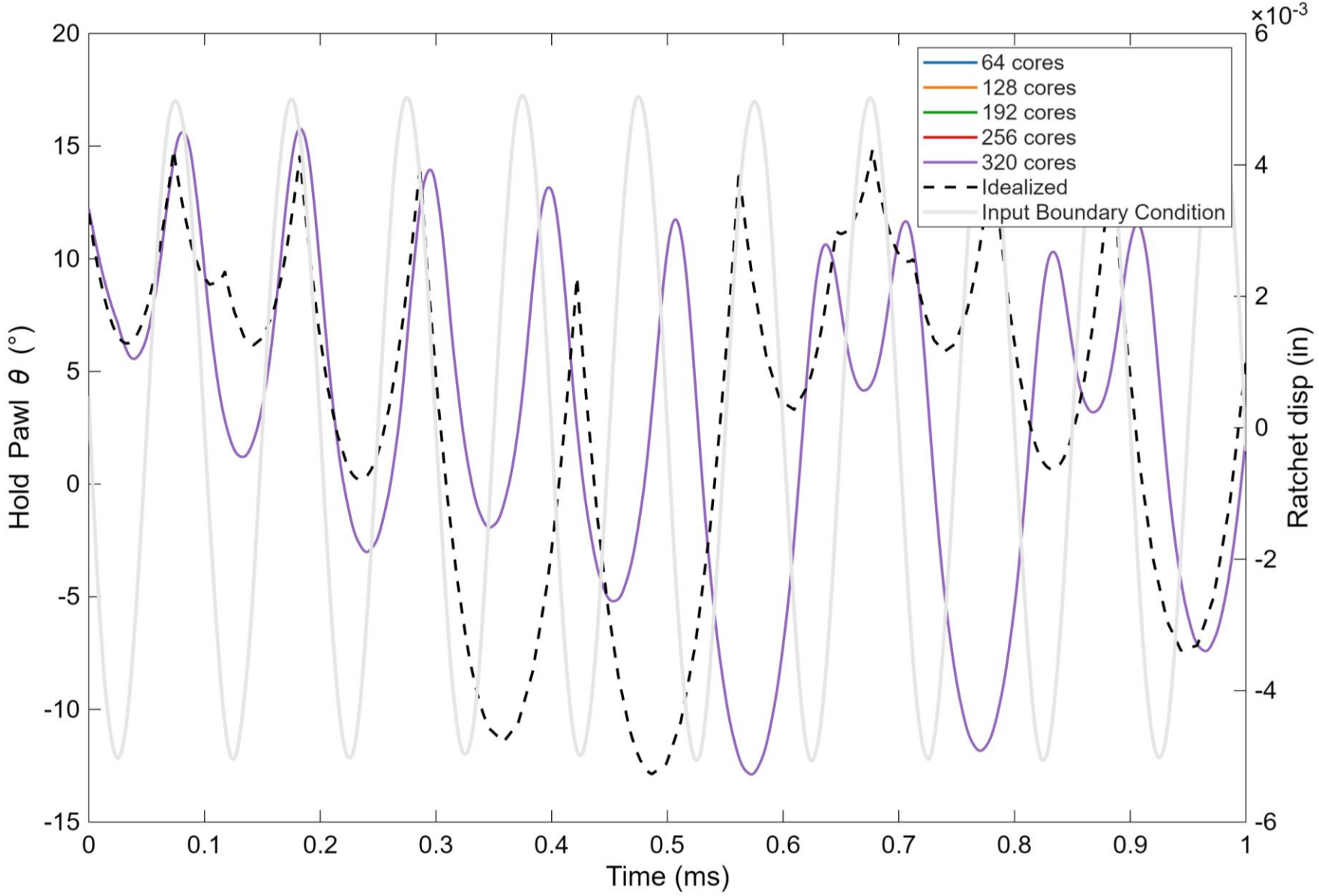






Haversine shock settings: 0.025 inch amplitude, 0.5 lbf pseudo force, 100% mesh density, 10 mbi

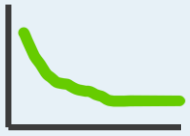
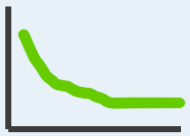
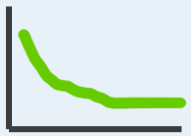
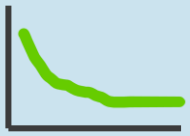
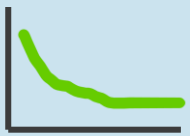
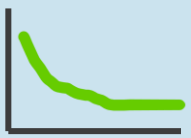
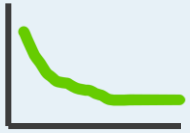
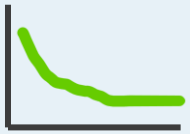
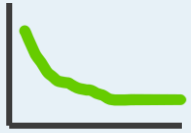
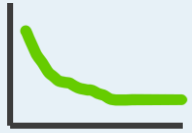
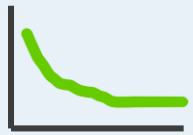
Processor Count – Pawl-Gear (Sinusoidal Vibration)

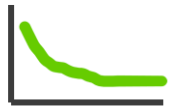


Sinusoidal vibration settings: 10 kHz, 0.005 inch amplitude, 13 lbf pseudo force, 100% mesh density, 10 mbi

Summary of Results

- Sensitivity of submodels to each parameter in each environment

	Pin-Spring-Pawl		Pin-Pawl		Pawl-Gear	
	Shock	Vibration	Shock	Vibration	Shock	Vibration
Momentum Balance Iteration						
Mesh Density						
Processor Count						

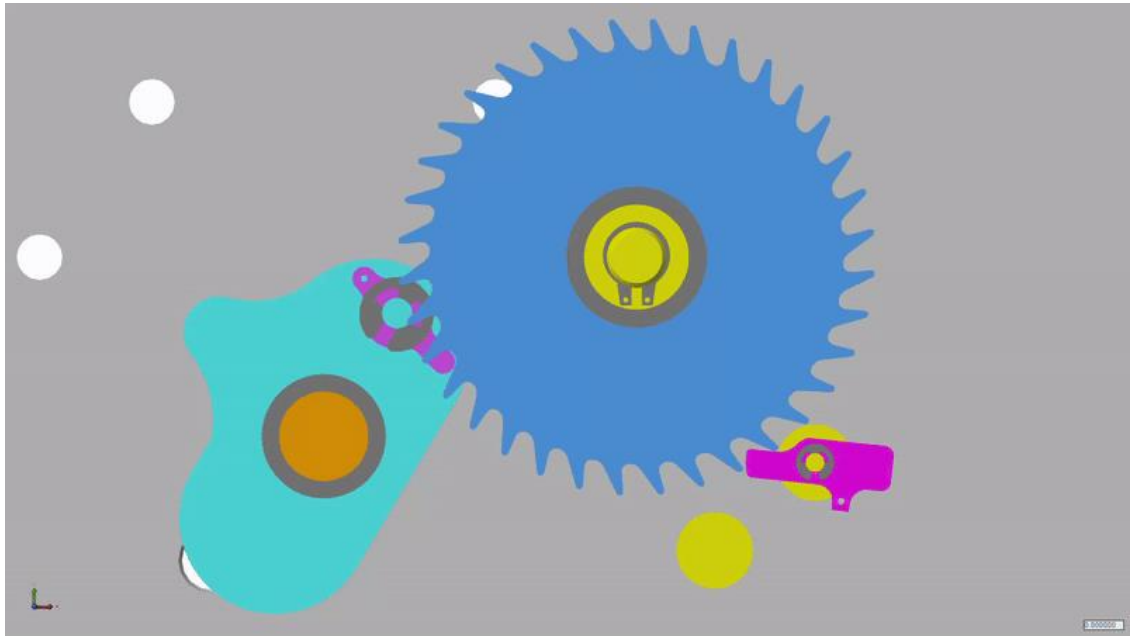
 = Insensitive  = Sensitive

Conclusion

- The parametric study of the non-physical parameters including momentum balance iteration, mesh density and number of processors used has revealed sensitive nature of the FE model even at the sub-model level of a ratcheting mechanism
- Results further affirms the complexity of a ratcheting mechanism and the consideration required to model such mechanism with accuracy

Future Work

- Combine the sub-models and study how the effects of the non-physical parameters change
- Study other non-physical parameters
- Perform similar study on the drive pawl model and extend study to assembly level.



Acknowledgements

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