Slide Valve Steam Engine Project

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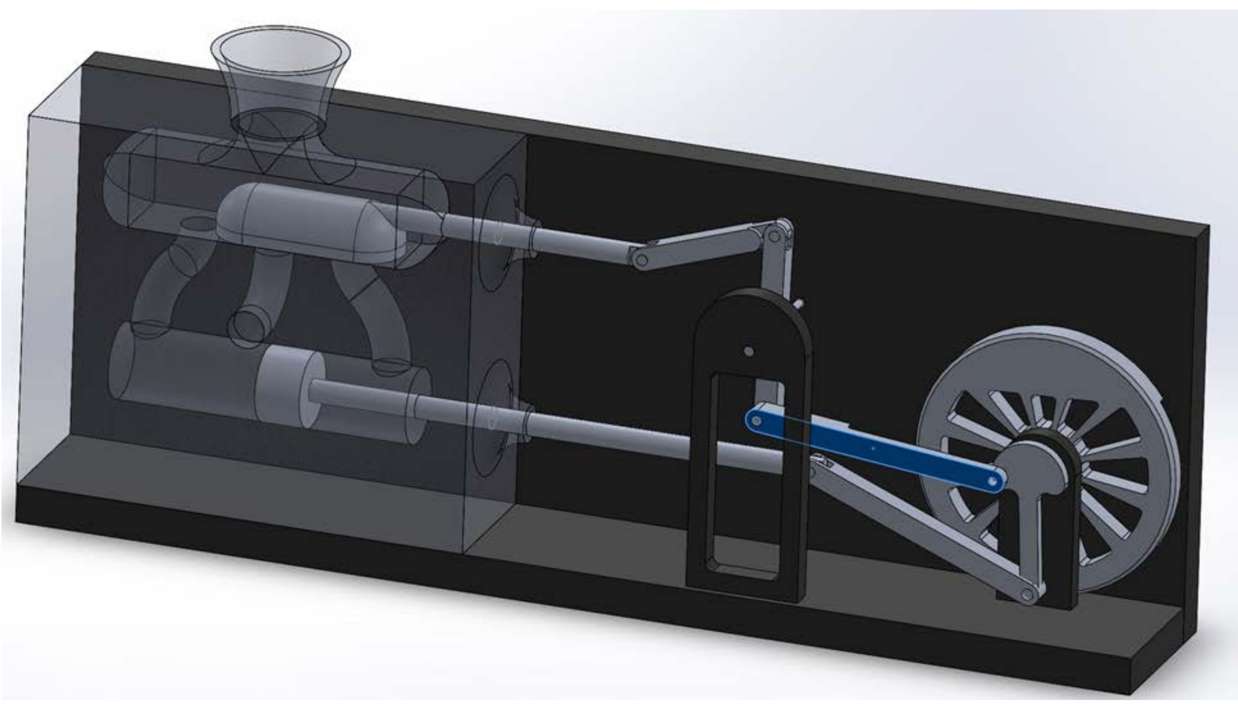
Introduction

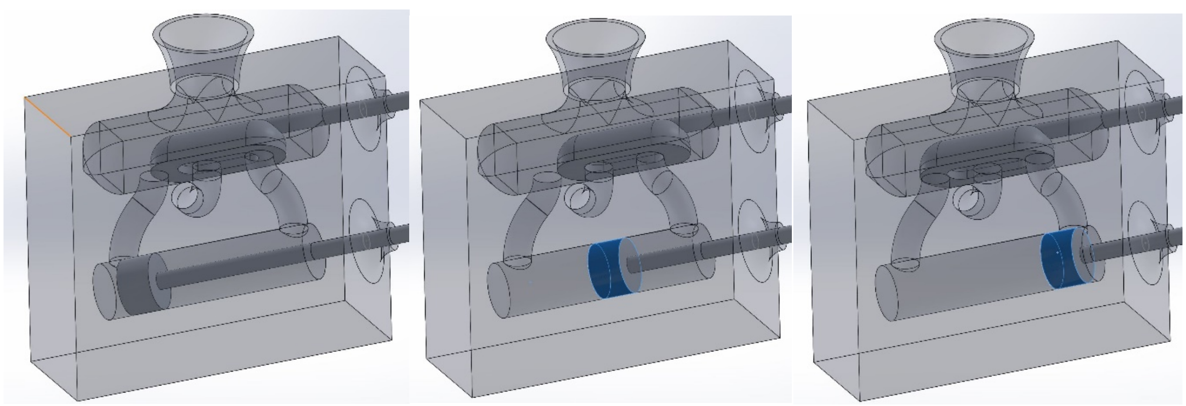
The mechanism I chose to model is a slide valve reciprocating steam engine, the basis of the steam powered locomotive prominent in the 18th Century. There are many different variants of the steam engine, but this version in the application of the locomotive contains one large bore cylinder with a D-shaped slide valve which is capable of sealing or connecting the cylinder ports to the exhaust port. Unlike a traditional two-stroke or four-stroke engine, there is pressure on both the top and bottom of the piston. Pressurized steam flows into the plenum alternately through the two cylinder ports which drives the piston up and down. This piston is connected by a linkage to the flywheel which in the case of a locomotive is the wheels itself, moving the locomotive forward on the train tracks.



To upon startup, water must first be boiled in the boiler then the direct pressure drives starts the engine. This report will discuss the three different ways used to model the engine startup before reaching a target steady state RPM and the motion analysis results based on ideal conditions. The model created is a simplified version of the actual steam engine used with less moving parts utilized so in addition, the material choices, software model limitations, and possible optimizations will be covered.

Mechanical Design

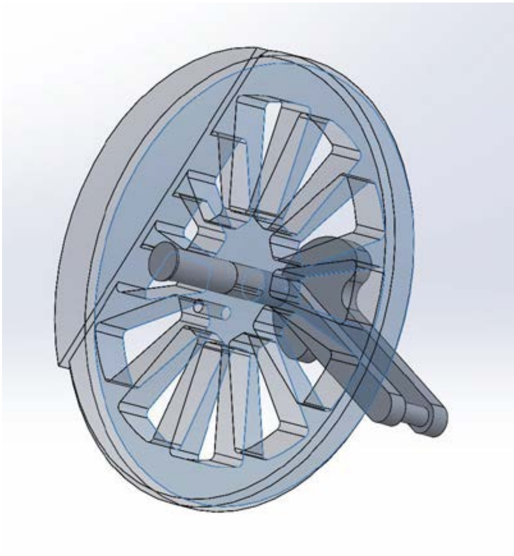
To begin, the principle driving force of the engine is the pressurized steam applied to the top and bottom of the piston. Steam is delivered from the boiled to the plenum and enters the cylinder via two intake ports at opposite ends of the cylinder bore. The delivery of steam to the plenum is beyond the scope of this model.



The cycle ‘begins’ when the piston (and crank-flywheel subassembly) is at top-dead-center with both intake ports sealed. The slide valve leads the cylinder and opens slightly earlier, allowing pressurized steam to apply force to the top of the piston. While steam is applied on tip of the piston, it moves down and compresses the steam below the piston head. As the valve moves down, the shape of the valve allows that steam below the piston head to escape by connecting it to the exhaust port. When piston reaches bottom-dead-center, the cycle reverses.

Meanwhile the flywheel continues to rotate thanks to the inertial from most of its mass being distributed at a large radius. In this case, the flywheel freely rotates but in other applications such as a locomotive, the flywheel is the wheel which moves the railcars and transmits rotational motion to linear motion. The piston-flywheel linkage is attached on one end to the piston by a pin and on the other end, it is concentric to the crankshaft journal. The stroke of the crankshaft correlates directly to the torque output of the engine, in our case we have a 2:3 bore to stroke ratio and a stroke of 150 mm.

As the valve needs to lead the piston, it sits 105º out of phase on a separate crankshaft journal. The valvetrain subassembly was the most complex portion of this project. It was possible to run a single linkage from the crankshaft from the second journal of the flywheel just as the it is done with the piston. However, the angle compared to the X-axis in which the valve moves is too great and would practically impossible to have smooth operation due to forces on the walls of the block. The use of a rocker reduces the force on the walls of the engine block by the slide valve piece by creating a smaller motion angle range of the rocker-valve linkage.



The most difficult part of the project mechanically was adjusting the timing of the valve. Adjusting the stroke of the piston is simple then the only thing required to be adjusted in the subassembly is the length on the linkage so the stroke is centered in the cylinder bore. However, the addition of the rocker and extra linkage made this more complex. The timing was configured over dozens of iterations, changing the rocker length, pivot point position, linkage length until it worked.

Mechanical Model Analysis

The material decision of 6061 aluminum alloy for all non-stationary objects was selected based on its lower weight as well as high strength, availability and manufacturability. It has a significantly greater tensile and yield strength than 1060 alloy with only a 0.1 N/mm^2 decrease in shear modulus. All this while being the same density as decreasing reciprocating mass is important. These parts will likely be milled and can be forged.

The decision for the engine block and all other stationary objects to be gray cast iron is for its vibration dampening characteristics and regular use in the car industry for more robust applications. This component can be made inexpensively in regard to the casting and material cost itself.

The model is mostly accurate without getting into small details outside the scope of SolidWorks motion analysis. One thing that would be necessary in a real-world application would be the need to balance the crank and flywheel subassembly so there would theoretically be no reaction force on the crank/flywheel support more than the weight itself. Further reducing the weight of the linkages (and rotating/reciprocating mass) could be done by milling ribs and surface cavities on the rods, also the model created does not have piston rings or end caps that would be present on practical engines with rods, pistons and pins.

Component:	**Material:**	**Mass:**
Engine Block*	Gray Cast Iron	372.87 kg
Slide Valve	6061 Aluminum Alloy	3.24 kg
Piston/Rod	6061 Aluminum Alloy	2.56 kg
Flywheel	6061 Aluminum Alloy	5.33 kg
Crankshaft	6061 Aluminum Alloy	0.72 kg
Piston-Flywheel Linkage	6061 Aluminum Alloy	0.78 kg
Valve Rocker	6061 Aluminum Alloy	0.59 kg
Slide-Rocker Linkage	6061 Aluminum Alloy	0.42 kg
Rocker-Flywheel Linkage	6061 Aluminum Alloy	0.56 kg
Pin - 30mm	6061 Aluminum Alloy	0.0080 kg
Pin - 60mm	6061 Aluminum Alloy	0.0014 kg
Rocker Pin	6061 Aluminum Alloy	0.054 kg
Flywheel Support*	Gray Cast Iron	3.86 kg
Rocker Support*	Gray Cast Iron	6.77 kg
Base*	Gray Cast Iron	377.80 kg

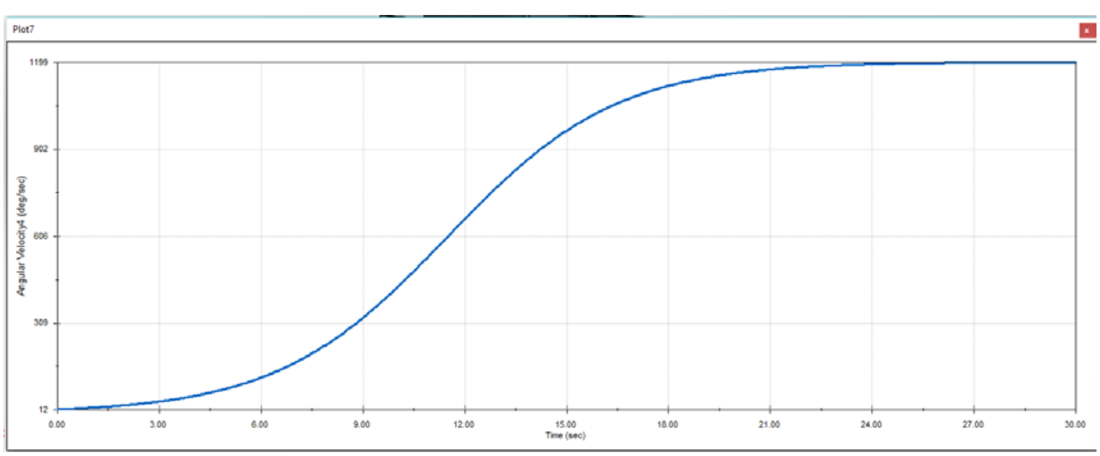
***Denotes stationary part**

Motion Analysis

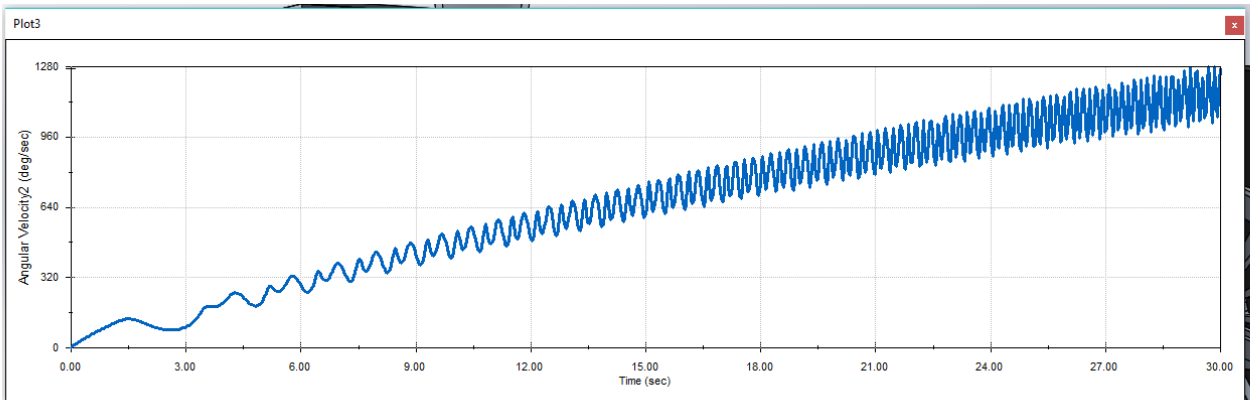
In a gasoline engine, the car is started by a solenoid actuated electric starter, then is no longer needed when combustion commences. In the steam engine however, the steam starts the engine and drives the revolutions throughout its operation. To perform motion analysis on the reciprocating piston and slide valve, the optimal way to do was would be to apply a force as a function of distance to each side of the piston in a harmonic equation. Attempting this however was not fruitful in collecting useful data and information on steam engine torque curve shapes was hard to come by. To cover most possibilities, three different methods to get the steam engine up the target angular velocity of 180-220 RPM within an arbitrary 30 second time range were analyzed.

One solid body contact group was added to each simulation at the concentric mates of the block and slide valve/piston. The second contact group was added between the crank and support/base wall. These solid body contact interfaces of greasy aluminum/aluminum were added to the motion studies as to not neglect friction, but not induce a computer crash. The first way being with a rotary motor applied at the flywheel using the restricted logarithmic growth equation where the degrees per second was expressed as:

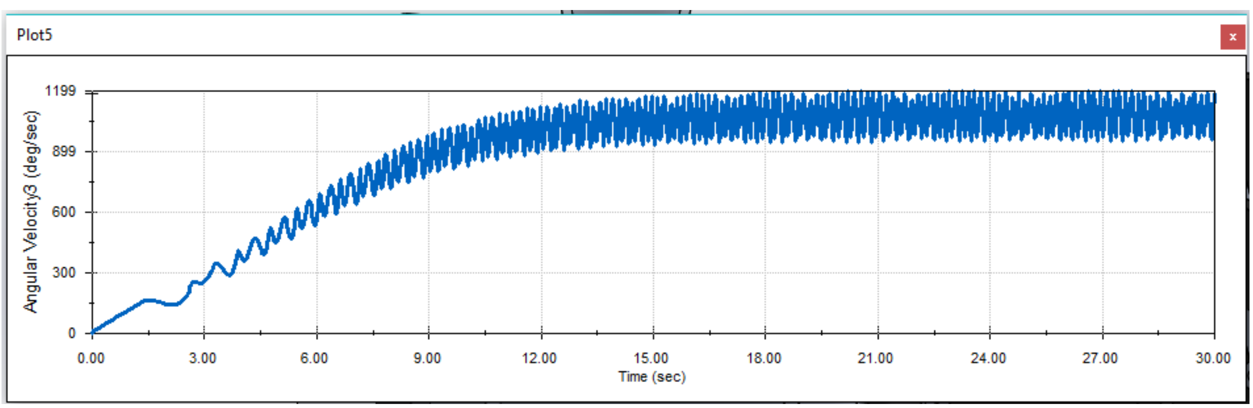
12((100*EXP(Time/2.5))/(99+EXP(Time/2.5))) degrees/sec*.



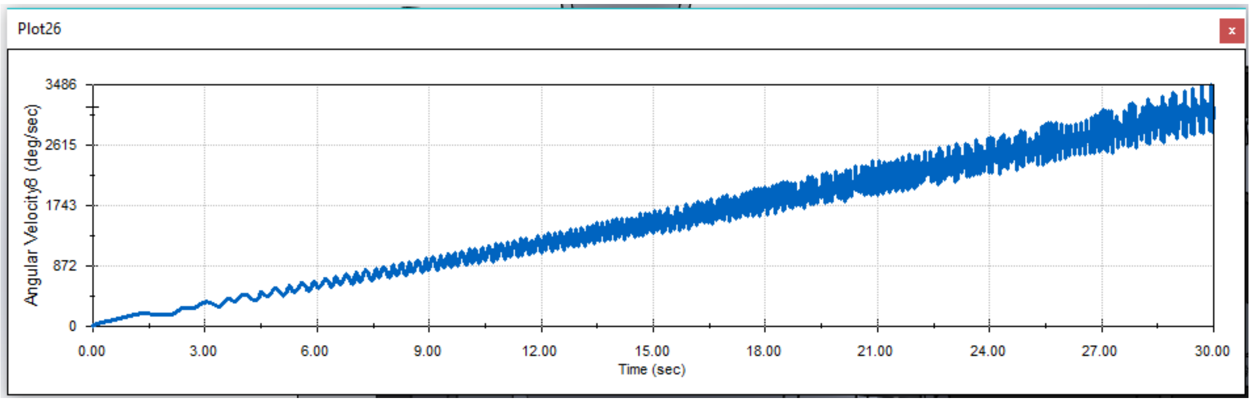
The second way to apply mechanical motion to the system was using a constant torque of 250 N*mm which resulted in an angular velocity curve that was nowhere near as smooth as the rotary motor and was limited not by an equation but would eventually be by friction.



The third method to apply mechanical motion to the system was using a torque expression as a function of velocity to produce a parabolic torque curve very similar to that of a car where torque increases then begins to decrease after a exceeding the optimal RPM range. The expression for torque used was: *-0.0008*({Angular Velocity})^2+0.8*{Angular Velocity3}+250* where 250 represents a startup torque of 250 N*mm.

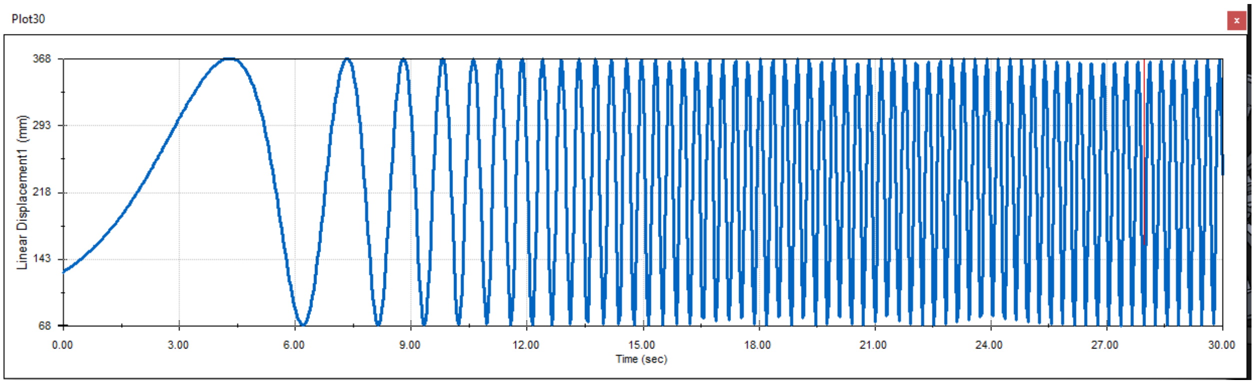


Lastly, a constant torque of 200 N*mm was applied to the flywheel face without friction as a control. In this motion study, the engine reached almost 600 RPM. In all motion studies the crank and piston-crank linkage were started at a 105° angle for proper torque delivery on startup if force was applied to the face of the piston

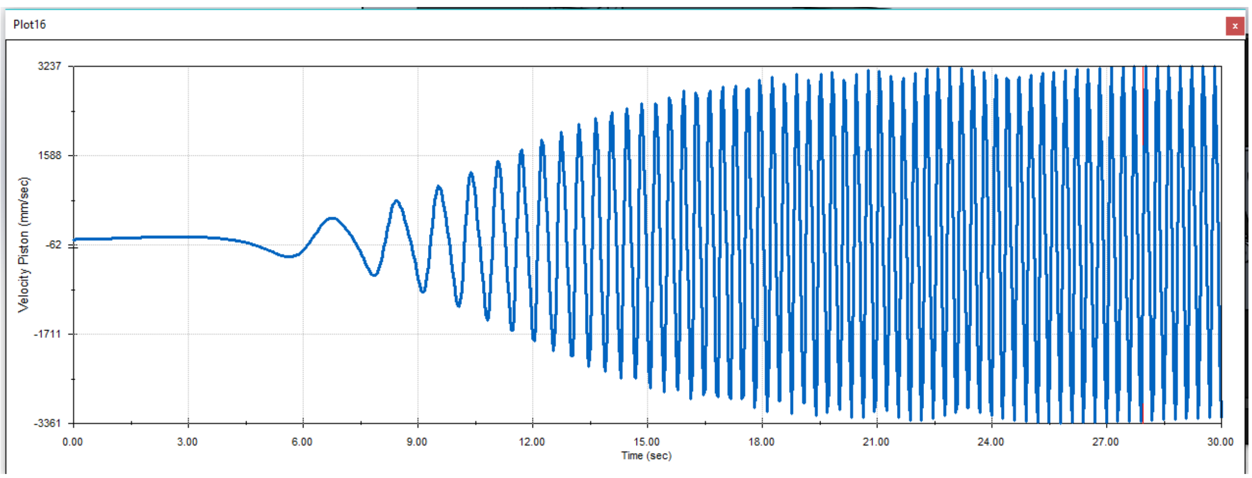


In the results where torque was applied, harmonics can be observed with increasing amplitudes at higher angular velocities and in each graph the angular velocity amplitudes are approximately one-tenth of the operating rpm. It is believed that to be due to the crankshaft/flywheel subassembly in addition to the linkages being unbalanced.

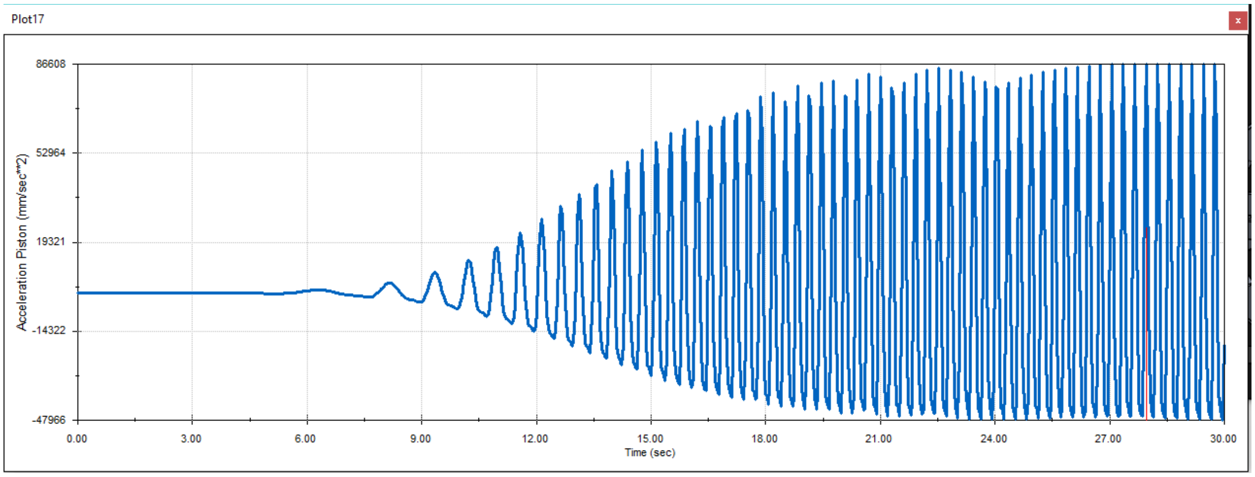
Of the four motion studies, the most stable was the first which incorporated the logarithmic rotary motor and therefore the rest of the plots are based on that as the end product of a locomotive is the transfer of energy from angular velocity to a constant linear motion. Plotting the piston's linear velocity yields a sinusoidal wave with an amplitude of 150 mm as expected since this is the stroke of the crankshaft. The wavelength however gets smaller as the linear velocity increases until the angular velocity reaches steady state of 1200 degrees/sec.



Plotting the x-component of the linear velocity of the piston, the amplitude and wavelength increases until the angular velocity reaches steady state of 1200 degrees/sec.



Plotting the x-component of the linear acceleration of the piston shows the piston accelerates more on the upstroke than downstroke, furthering the hypotheses that the crankshaft/flywheel subassembly is significantly unbalanced.



It would be preferable to plot the reaction forces on the central crankshaft journal but SolidWorks warned the assembly was overdefined and would not yield accurate results.

Conclusion

Creating a model engine in a computer model or physical form, the most important thing I would say is to ensure the assembly is balanced. This includes reducing reciprocating mass, balancing the crankshaft and ensuring the engine has an appropriate rod to stroke ratio not just at the piston and crank, but the rest of the linkages. Not doing so will cause excessive force on the constraining walls. Under real works conditions, extreme vibrations can destroy a reciprocating compulsion engine, I am unaware however with the weight of a locomotive the effects of these vibrations or how close they are to unacceptable levels.

The project itself helped me understand the relationship between torque, force, angular acceleration and angular velocity much better since I was working so much with the governing equations within SolidWorks. I begin writing feedback looks with error and gain which was helpful in learning concepts of controls but ultimately did not yield repeartable results in this application. When conducting motion analysis, I now understand how essential it is to correctly define mates because over defining an assembly can cause issues outside of strictly motion analysis.

Under perfect conditions of the rotary motor to generate a velocity from an expression we can see the relationship between the variation in amplitude as well as the displacement, velocity, and acceleration before and after steady state conditions are met.

References

W. R. King, Steam engineering: a textbook. New York: John Willey, 1913.

B. Daimond, Science of Railway Locomotion, 2011. [Online]. Available: http://www.brooklynrail.net/science_of_railway_locomotion.html.

Appendix