High Fidelity Modeling and Simulation of Tracked Elements for Off-Road Applications Using MSC/ADAMS

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Abstract

This paper is based on an independent study which had two goals. First was to apply the principles of dynamics to understand the behavior of complex mechanical systems. Within this theoretical framework, the Automatic Dynamic Analysis of Mechanical Systems (ADAMS) software package was used to make virtual models and accurate simulations of these systems. This part of the project was mainly a learning experience; the only objective was to gain experience with the virtual prototyping process and become proficient with the ADAMS package.

Phase two of the study involved applying the modeling and simulation methods learned on a tracked vehicle model (in this case, a hydraulic excavator model). An in-depth description of the modeling process is provided. The goal was to investigate how the response of the model changed when the running conditions and methods of propulsion were varied. Results of the model’s responses are provided, with analysis of the driving torque, forward velocity, bushing shearing stress and the idler tensioning system. The results are followed by a conclusion of the independent study which focuses on the challenges and possibilities that tracked vehicle simulations present.
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1 Introduction

One of the most important steps in any engineering design process is to test the proposed design under various operating conditions. Traditionally, this entails creating one or more prototypes and conducting laboratory/field testing to determine if the design achieves its performance goals. An engineer will want to do as much testing as is necessary to produce the design that minimizes the number of changes that have to be made later on in the design process since these can be extremely expensive. However, creating many prototypes with many testing conditions becomes cost and time prohibitive. This is why engineers have embraced modeling and simulation in the form of virtual prototyping. It costs next to nothing to create and test a virtual design compared to building and testing a physical prototype. The continuous increase in computer processing speed affords the opportunity to model and simulate increasingly complex mechanical systems.

The objective of this independent study was to learn how to use the virtual prototyping package ADAMS, and implement this knowledge to investigate the behavior of a complex model consisting of tracked elements. A model of a hydraulic excavator was provided by Holger Haut [1], which was used as a basis for the investigation of tracked elements in ADAMS. The model was driven over a flat surface and a small obstacle using different propulsion methods. The propulsion methods include applying a constant rotational motion and applying a constant torque at the rear sprocket. Both driving force methods were varied between high and low values. Responses to the various simulation conditions were observed and important characteristics are calculated and discussed, such as: displacement of the suspension system, shear stress in the connecting pins and the forward velocity of the model.

2 MSC/ADAMS Software Package

At the time this paper was written, there were no available courses or training programs at the University of Wisconsin-Madison that taught a modeling and simulation environment such as ADAMS or DADS (Dynamic Analysis and Design System). As such, the first step to investigate anything in a virtual prototyping package is to learn how to use the software. Through a combination of the User Manual, Online tutorials, the MSC online forum and fellow students working in the Simulation Based Engineering Lab (SBEL, [2]), the author was able to build a strong foundation of knowledge about using the ADAMS package. This paper does not go into details about the individual learning exercises, but rather will give a general overview of the ADAMS/View, Solver and Postprocessor programs.

The ADAMS/View program is a 3-D interactive environment in which the parametric properties of the mechanical system are defined. Constraints between individual parts can be applied so that the resulting motion of the system mocks that of its physical counterpart. Forces, torques and motions can be applied so the system moves in a particular fashion. It is also possible to implement control systems. The user only needs to define the parametric data; the equations of motion are automatically applied when the
finalized model is sent to the Solver program. Program scripts can be utilized by the Solver program to customize and guide the simulation process. The Solver automatically integrates the equations of motion for a certain time step which is determined by the Solver and outputs data in the form of result sets for each time step. After the simulation is complete the result sets can be accessed by the Postprocessor and the data can be viewed in the form of plots. If geometric shells are supplied for the parts, three-dimensional animations of the simulations can be created and visually inspected as well. It should be noted that this is an extremely general description of the ADAMS program. Each step in the simulation process is customizable; measures, sensors and user-defined subroutines are only a few of the ways the program can be manipulated to increase the accuracy and fidelity of the model and simulation.

3 Tracked Model Methodology

3.1 Hydraulic Excavator Description

The tracked vehicle used as a basis for this investigation was a hydraulic excavator. This type of tracked vehicle was selected because its simple suspension system and low operating speeds make its behavior easy to predict. The model consists of five road wheels, one main idler, three support rollers, a drive sprocket, and 45 tracks. A side view of the model, rendered in ADAMS/View, is shown below (Figure 1). Since this type of excavator carries heavy loads at low speeds, the suspension system only consists of a tensioning spring on the idler. The road wheels, support rollers and drive sprocket are all rigidly attached to the undercarriage. Each track is connected to the next track by a single pin rubber bushing. There is also a center block that represents a rigid connection to the chassis of the excavator.

![Figure 1: Hydraulic Excavator model used in the investigation](image)
3.2 Hydraulic Excavator Model

The physical model of a hydraulic excavator was implemented in ADAMS using the following methods. Each individual track was connected to its two adjacent tracks by means of a revolute joint about the center of each pin hole, with coulomb friction in the joint to represent the behavior of a single pin rubber bushing. Contact forces between each track solid and each contact body were created. Each track comes into contact with: five road wheels, the idler, three support rollers, each of the three solids comprising the drive sprocket, and the ground block. Therefore, there are 45*13 or 585 contact forces in the model. Calculating contact forces is very computationally demanding; they create very large reaction forces, which results in motion that requires very small time steps to resolve accurately. Simulating the large number of contact forces present throughout the simulation is the major bottleneck in tracked vehicle simulations. In the past, tracked vehicle simulations have used a super-element to describe the behavior of the track system because of the lack of computer power to represent each track and its representative contact forces individually. This method has many drawbacks, including the necessity of testing a physical model to get the data to convert the track system into a super-element, and the inability to change the design of the track system once a super-element has been created. Creating a super-element in place of the track system defeats the purpose of virtual-prototyping and must be avoided if computer modeling and simulation is to be used effectively in the design process of tracked vehicles.

All of the track parts use the same geometry and therefore have the same parameters. The material of the track is steel; using the density associated with steel and the volume of an individual track, the calculated mass is approximately 498 kg. The overall width, depth and length are: 625mm, 275mm and 900mm, respectively. Width, depth and length correspond to the X, Y and Z directions, respectively, in Figure 2. It is apparent from Figure 2 shown below that the modeled bushing pin is loaded under double shear. The diameter of each hole is 100mm.
Each of the five road wheels are modeled identically. Each road wheel spins around its own center axis by a revolute joint that is at a fixed location from the center block. Friction is present in these joints. There are also contact forces between each road wheel and all the individual track elements. The three support rollers are modeled in a similar fashion as the road wheels.

The rear sprocket is composed of three parts: two identical gears and a drive sprocket. Both gears are rigidly attached to the drive sprocket, and each has contact forces with all 45 tracks. The drive sprocket revolves around its central axis by a revolute joint which is a fixed distance from the center block. It is also where the driving torques and motions are applied.

The front idler is modeled almost identically as the road wheels; it is constrained with a revolute joint and has contact forces with the tracks. There is one main difference which is due to the presence of the suspension system. The revolute joint is not a fixed distance from the center block; it is constrained with a translational joint that represents the suspension system. This allows the idler to have an extra degree of freedom so it can move forwards and backwards. A single component horizontal force applied to the center of the idler keeps the tracks in tension.

Obstacle blocks were imported into ADAMS in the form of parasolids created in Unigraphics NX4. They were then fixed to the ground and contact forces were created between the obstacle and the tracks.
4 Virtual Testing Conditions

In order to gain a better understanding of the mechanics involved in tracked vehicle simulations, the model was tested under multiple conditions. The three independent variables chosen to create different operating conditions were: the ground block, the propulsion method, and the rate of the propulsion method. Eight trial simulations were run to be able to judge the effect of the interactions between the independent variables.

Using both a flat running surface and one with an obstacle gave insight into how the reaction forces affect performance on non-flat surfaces. This kind of testing is ideal for any vehicle that is designed to operate in off-road conditions. The obstacle used in place of the flat ground block in this investigation is shown with dimensions in Figure 3. The width of the obstacle is much larger than that of the tracks to prevent any rolling of the model. The incline is 12 degrees, with a 250mm vertical height and a half meter flat top surface. Flat, level running surfaces are present before and after the obstacle, so the model will approach the obstacle at a steady-state condition and will be allowed to return to its steady-state condition after.

![Figure 3: Obstacle used in place of the flat running surface. Units are [mm] and degrees](image)

The model’s propulsion method was always applied to the rear sprocket and varied between fast and slow rotational motions, as well as large and small driving torques. Initially, the model was driven by a constant rotational motion at the rear sprocket. This was done to emulate “ideal” driving conditions at a constant forward velocity. In practice, a constant rotational motion does not equate to a constant velocity; rather, the velocity oscillates around a steady-state value. This is due to the presence of the idler tensioning system and its reaction to forces of different magnitudes being applied on the front idler. Simulations were carried out at sprocket rotational velocities of 56.17 and 112.34 degrees per second (9.36 and 18.72 RPM, respectively). By measuring the reaction torque on the sprocket, approximate values for driving torques were determined. Details and an analysis of the torque measurements are provided in the results section.

Multiple measurements were made in order to quantify the desired values and will be discussed in detail in the following section.
5 Results

As was stated previously, eight simulations were conducted with the tracked vehicle model under different operating conditions. Measures were created to gauge different aspects of the vehicle’s performance throughout the simulations. Some measures were specific to certain independent variables, such as the angle of incline of the track as it encounters an obstacle. Others were applicable to all the simulations; the force present in the revolute joint between tracks is an example of this. Each measure will be described and analyzed in detail in the subsequent sections.

5.1 Description of Measures

Before a torque could be applied as the driving method of the model, an appropriate numerical value for the torque needed to be determined. For the simulation to succeed, the torque needed to be large enough to overcome static friction; however, an excessively large value would make the model move faster than it does in reality. The goal was to apply a torque that would result in a comparable forward velocity as was attained when a constant angular velocity was applied to the drive sprocket. This desired value was obtained by measuring the torque on the rear sprocket’s revolute joint. It should be noted that the torque varied substantially; between 0 and 7.43*10^5 N-m. This large variance was investigated and appropriate values for the constant driving torque were determined. The average values were obtained for the slow and fast motions and used for the trials involving the model being driven by a torque.

The forward velocity of the model was of particular interest for two reasons. First, when a constant driving torque is applied, the time it takes the model to reach its steady state velocity marks the point in which the friction forces in the model have completely transitioned from their static to dynamic values. The rate of this transition on flat ground is examined. Second, it can show how substantially an impact with obstacles affects the speed of the vehicle. The center marker of the central block was used to measure the model’s velocity in the forward (negative x) direction.

When designing tracked vehicles that operate at high velocities, such as the M1A1 Abrams tank, the shearing stresses experienced by the bushings between the tracks becomes very important. The strength and reliability of these components can become a limiting factor as far as maximum vehicle speed is concerned. In order to determine the shear stress in the bushings, a measurement of the magnitude of the force in the revolute joint between track one and track two was made. This revolute joint was chosen because it explores all the probable points of maximum stress. It begins the simulation underneath one of the road wheels, is pulled around the sprocket, and moves some distance across the top portion of the track. A picture of the joint described is shown in Figure 4 below. The cross sectional area of the joint was then utilized to compute the shear stress from the forces present in the joint. Values of the shear stress measurements do not have any significance apart from their relative magnitude. In order to compute shear stresses that correlate to actual values, an entire hydraulic excavator model or a
super-element representing the dynamic behavior of the rest of the vehicle would need to be incorporated into the simulations. However, the relative magnitudes of the shear stresses can help pinpoint the location of the maximum stress experienced by the bushings.

The idler tensioning system is modeled as a 500,000N constant horizontal force that acts on the center of the idler. Since the drive sprocket, road wheels and support rollers are all mounted rigidly, the force pushes the idler forward to create tension between the tracks. However, as the model begins to move, so does the position of the idler. Large deflections in the idler tensioning system could have disastrous consequences on a tracked vehicle. This is the motivation for measuring the distance between the center marker of the idler and the center marker of the drive sprocket. The force modeling the tensioning system is activated before the model begins to move, so a steady state value for the measurement can be determined. Any variation from this steady state value is of interest, and is discussed in depth in the following results section.

For the simulations involving the obstacle, an angle of incline was also measured. This was accomplished by simply measuring the horizontal distance between the central block’s and the rear sprocket joint’s center markers, as well as the vertical displacement between the same two objects. The inverse tangent yielded the angle of incline of the model, and was mainly used as a determining factor to pinpoint the simulation times when the model had traversed a certain portion of the obstacle.
5.2 Analysis of Measures

This section provides a detailed analysis of each of the measures described above. The relative magnitudes of the measures rather than the values themselves is what will be examined since this investigation only models the tracked propulsion system of a hydraulic excavator, and not the entire vehicle. Nevertheless, an improved understanding of the behavior of a simulated tracked vehicle can be achieved, which will help guide future work and research in this field.

5.2.1 Driving Torque

As was stated in the previous section, an approximate value for the torque needed to be determined before a constant torque could be applied as the propulsion method. Not only did the torque need to be large enough to overcome static friction, but it would also need to be sufficient to drive the model over an obstacle. Thus, it was decided that torque measurements from the obstacle simulations where a constant sprocket angular velocity driving method would be used. The average and RMS values of both the high and low speed rotational motion simulations were computed in ADAMS/PostProcessor, for \(1 \leq t \leq 10\), where \(t\) is the simulation time in seconds. The values are calculated starting at one second because the step functions describing the force on the idler tensioning system and the rotational motion on the rear sprocket reach their final values after one second of simulation time.

The average and root mean squared (RMS) values of the measured torque during the slow rotational motion simulation were 7.83E7 and 9.56E7 N-mm, respectively. The average and RMS values of the torque during the fast rotational motion simulation were 8.67E7 and 1.07E8 N-mm, respectively. Using these average and RMS values, it was decided to use torques of 8E7 and 1E8 N-mm on the driving sprocket when the rotational motion was deactivated and the driving torque was used as a propulsion method. Both torque plots are shown below and the substantial variance in values as stated in section 5.1 can be easily noticed.
These large spikes in the torque measurements were investigated graphically through the animations provided by ADAMS/PostProcessor. Every simulation had 100 output steps per second of simulation; hence, the correct frame could be easily obtained by multiplying the time a spike occurred by 100, rounding to the nearest integer and adding one (frame 1 corresponds to time=0). The most noticeable variance in the torque measurements can be seen in Figure 5 at time=8.5679 seconds. Frame 857, which is shown below in Figure 7, holds the answer to the torque spike. The front end of the track had cleared the obstacle, but not yet touched the flat ground while the center tracks were still on the top landing of the obstacle. This would have generated a large acceleration and the rotational speed of the rear sprocket would have increased greatly if the constant
rotational motion constraint was not present. The largest torque spikes in Figure 6 were all due to odd load distributions on the road wheels while the model was traversing the obstacle.

Figure 7: Snapshot of the animation at which a torque spike occurred

All the torque spikes in Figures 5 and 6 are due to the transient event of the model clearing the obstacle. Therefore, during steady state operation the torque’s variance should be able to be calculated to a high degree of accuracy. This was true when the model was driven with slow and fast rotational motions on flat ground, and torque plots from these simulations are included in the Appendix (Section 8.2, Figures 13 and 14).

5.2.2 Forward Velocity

The method of applying a constant torque consisted of placing a single component torque on the drive sprocket and using a step function to ramp up the torque to its nominal value between 0.6 and 1.0 seconds. Consequently, the simulations involving small and large driving torques have similar initial velocities when the full torque is applied, as well as similar coefficients of friction. The time it takes for a model to reach its steady state velocity is the time it takes to transition from static to dynamic values of friction. Plots of the forward velocity for large and small driving torques on flat ground are shown below.
The forward velocity of the model in Figure 8 does not reach steady state after ten seconds; a 15 second simulation needed to be completed to find a steady state speed of 3264 mm/sec. This value can be compared to the final speed of the model in Figure 9, which is approximately 4371 mm/sec. One conclusion that can be drawn from these results is that the model driven by a torque of 8E7 N-mm does not completely transition to dynamic friction values. If this were the case, the steady-state velocity should be 80% of that experienced by the model driven by the larger torque, since a majority of the mechanical losses in the system are assumed to be due to friction.

In order to better understand the track model’s response to an obstacle, the following table was created. It lists the time at which the model encounters a certain point in the simulation and the forward velocity associated with that point. Times were determined with the angle of incline measure or by visual inspection of the animations.
Table 1: Simulation time of certain events, and associated forward velocities

<table>
<thead>
<tr>
<th>Driving Torque [N-mm]</th>
<th>Model engages obstacle</th>
<th>Model at peak of obstacle</th>
<th>Model disengages obstacle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Simulation time [s]</td>
<td>Velocity [mm/sec]</td>
<td>Simulation time[s]</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8*10^7</td>
<td>3.67</td>
<td>1431</td>
<td>6.03</td>
</tr>
<tr>
<td>1*10^8</td>
<td>2.86</td>
<td>1842</td>
<td>4.47</td>
</tr>
</tbody>
</table>

This table shows that the smaller driving torque is just large enough for the model to maintain its forward velocity when encountering the obstacle and can be proved with the forward velocity plot for this simulation (Appendix 8.2). Once static friction is overcome, the model has no problems traversing a small obstacle.

5.2.3 Modeled Bushing Shear Stress

Even though actual bushings were not used in this simulation, revolute joints with corresponding frictional forces were used to approximate their behavior. Thus, the forces being transferred through the revolute joints are the same forces that would be applied to the bushings. The shear stresses resulting from these forces were analyzed for the flat ground simulation when a small and large constant torque was applied. Using a bushing diameter of approximately 100 mm, and knowing the loading case is double shear (See Figure 2), the area used for shear stress calculations is 0.01571 m^2. The following table lists the simulation times at which the revolute joint is at a location where large shear stresses are likely. Times were obtained graphically from the animations and are not exact.

Table 2: Simulation time of certain positions of measured joint

<table>
<thead>
<tr>
<th>Driving Torque [N-mm]</th>
<th>Encounters road wheel [s]</th>
<th>Engages drive sprocket [s]</th>
<th>Disengages drive sprocket [s]</th>
<th>Encounters first support roller [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8*10^7</td>
<td>2.01</td>
<td>3.39</td>
<td>4.85</td>
<td>5.49</td>
</tr>
<tr>
<td>1*10^8</td>
<td>1.62</td>
<td>2.63</td>
<td>3.77</td>
<td>4.24</td>
</tr>
</tbody>
</table>

Shearing stresses at the listed times can be calculated by dividing the force in the joint at the given time by the cross sectional area calculated above.

Table 3: Calculated Shear Stress at Various Joint Positions

<table>
<thead>
<tr>
<th>Driving Torque [N-mm]</th>
<th>Encounters road wheel [MPa]</th>
<th>Engages drive sprocket [MPa]</th>
<th>Disengages drive sprocket [MPa]</th>
<th>Encounters first support roller [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8*10^7</td>
<td>17.70</td>
<td>18.40</td>
<td>15.02</td>
<td>15.79</td>
</tr>
<tr>
<td>1*10^8</td>
<td>19.42</td>
<td>19.03</td>
<td>13.88</td>
<td>17.76</td>
</tr>
</tbody>
</table>
Although the values calculated are individually of no significance, their relative magnitudes are. The largest sustained shear stresses occurred when the joint was traversing the road wheel section of the track.

There were a few significant spikes in the measured joint force, which occur at seemingly random locations of the revolute joint. Measures of the forces in other joints were made to corroborate this claim, and it became evident that all the spikes occur after a joint disengages the drive sprocket, and begins to traverse the support rollers. A plot of the joint force is provided below for the case of the 1E8 N-mm driving torque simulation on flat ground. Notice the spikes begin to occur at 5.5 seconds, just after the joint has encountered the first support roller. Problems stemming from integrating the contact forces could be one cause of the discontinuities in the joint forces, but other possibilities can not be ruled out at this time. Suggestions for in-depth research of this problem are given in the conclusion section.

\[ \text{Figure 10: Force experienced by a revolute joint during a simulation} \]

5.2.4 Idler Tensioning System

The motivation for monitoring the displacement of the idler tensioning system is due to the variation in the model’s velocity during simulations. Plots of the velocity and idler displacement from the same simulation are overlaid in Figure 11 and a clear pattern emerges. The frequencies of both waveforms are the same and their average values are inversely proportional to one another. As the average forward speed of the model increases, the average displacement of the idler tensioning system decreases. In other words, at higher speeds there is less tension in the tracks. This would cause problems for tracked vehicles which operate at high speeds; however, this is not the case for a
A closer view of a few periods of the waveforms of Figure 11 shows an interesting event. While an increase in overall speed decreased the average displacement of the idler tensioning system, an instantaneous increase in speed results in an instantaneous increase in the displacement of the idler. This observation is shown in Figure 12. One hypothesis to explain this behavior is that there is a response time similar to that of a mass-spring system. The mass of the idler is fairly large compared to the mass of the individual tracks, and this could cause the idler’s response time to be about ½ of a wavelength as seen in Figure 12. The frequency of these oscillations in Figure 12 is approximately 7 Hz. As the forward velocity increases, so does the frequency of these oscillations. Appendix 9.3 contains a plot (Figure 16) to illustrate this claim.
Figure 12: Direct correlation between the waveforms produced by the forward velocity and the idler displacement

6 Conclusions

A semester-long learning process delving into the world of computer modeling and simulation was completed. Experience was gained with the MSC/ADAMS multibody dynamics analysis software, which led to an investigation of tracked model simulations. Results of this investigation gave insight into many characteristics of a tracked vehicle including: driving torques, model velocity, bushing shear stresses and suspension behavior. A design of experiments was created to test these elements under varying operating conditions. Any of these characteristics could be the subject of in-depth research, such as describing the means by which force inputs are transmitted from the ground to the chassis of a tracked vehicle [3].

7 Acknowledgements

I would like to thank Dr. Dan Negrut for his guidance and enthusiasm towards the independent study on which this paper is based. I would also like to thank Holger Haut for graciously providing support for the track model which this investigation is based on. Finally, I would like to thank Makarand Datar for his assistance while learning MSC/ADAMS, as well as Nick Shafer and Toby Xu for their help in editing this paper.
8 References


9 Appendices

9.1 Torque plots

Figure 13: Plot of the torque exerted on the driving sprocket during a simulation on flat ground where a slow rotational motion was the driving method.

Figure 14: Plot of the torque exerted on the driving sprocket during a simulation on flat ground where a fast rotational motion was the driving method.
9.2 Velocity Plots

Figure 15: Plot of the forward velocity v. time, showing the velocity being maintained while the model overcomes the obstacle

9.3 Idler Tensioning System Plots

Figure 16: Waveforms of the displacement of the idler tensioning system and the forward velocity of the model at high speeds