Static and Dynamic Simulation of Rubber Tracks using Multi-Body Dynamics to Study Pressure Contact on Treads

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Abstract: The study of large and complex structures that involve multiple bodies functioning kinematically together have been rather challenging until recent technological advancements were found. When one was faced with such a task, the most viable way to proceed was to make use of simplicity and assumptions to conduct analysis at the component of interest. In what is called "system-level" simulation, all components of a given structure or system are taken into consideration in the analysis, thus reducing the yield of error in the theorization of your model. Such analysis method must be used when studying the phenomena of tread wear in a rubber track as it rolls over an undercarriage type-motion system. The strains and contact pressure distributions present on the tread contact surfaces are affected by too many variables, and to great extent; the kinematics of the undercarriage components. Thus, the purpose of this study is to understand the intricate physical mechanisms that lead to tread wear in such dynamic system in order to obtain an analytical tool which will allow us to generate an optimal tread design that will perform better in same conditions.

Keywords: Rubber, static, implicit dynamics, MBD, Contact

1 Introduction

For many years engineers have been making use of computer models to predict mechanical behavior of components by use of finite element codes (FEM). Many engineering industries have greatly benefited from this, and seen the potential of using virtual testing to help and fasten the research and development of their products. As a consequence, great efforts have been placed in developing new and improved mathematical methods to increase the efficiency of these computer models. Moreover, thanks to the advancements in finite element analysis (FEA) programming and computer technology, especially in the last couple of decades, engineers are now more capable than ever in predicting complex mechanical behaviors. This progress in research using virtual testing (VT) has allowed, among others, for faster design-to-manufacturing processing, reduction of prototyping and experimental testing, cost savings and overall better product performance.

There are now many different mathematical methods and codes to accommodate all types of industries in creating virtual testing for their products. In the case for automotive and manufacturing industries, such as in the case of this paper, the partial derivative method (PDM) is greatly used in their FEA codes. The use of these analytical methods will allow the engineers to develop essentially any kind of VT upon which to conduct studies in their products. It is just as important for them to determine exactly what analytical method to use as how to properly use it. Whenever the engineers or researchers create computer models they have to take many crucial decisions on how to properly describe the actual component and test setup, so the virtual representation will be similar to the actual one. Among these, they will have to determine if the analysis can be simplified to study the product alone or if indeed, there are other factors or other structural components that will affect the test.

When a virtual study can be simplified and is conducted only on the component of interest, the analysis can be referred as a "*component-level*" simulation type. This is normally the selected method when the test is dedicated only on the part to learn more about its intrinsic physical characteristics, or that a great deal of assumptions and simplifications can be used without affecting the overall accuracy of the VT with respect of the actual one. In some other cases, no such simplifications can be done and/or, for instance; the nature of the test is to study the performance of an assemblage of components. For those case studies, the engineer will make use of a "*systematic-level*" approach.

The main purpose of this paper is to provide the engineers and researches at Camoplast Solideal Inc. with an analytical tool upon which they can design the next generation tracks, using system-level techniques. In the case of the studies of the wear or thermal blowouts found on rubber treads, it is of great importance the tied-in mechanisms of the track with the undercarriage that holds it in place. Hence, it is believed that a great deal of the performance of the treads in the track will depend on how kinematically the undercarriage is linked to the track. For this reason, multi-body dynamic (MBD) research simulation was conducted and is the main topic of this paper.

1.1. Background information on Multi-Body Dynamics

Multi-Body dynamics, as the name implies, is the study of multiple bodies or objects, flexible or rigid (more to that later on), interacting with each other in a dynamic state. The motion of the bodies is described by their kinematic characteristics, which is basically the rotation and translational displacements. The description of this motion will be dependent whether the body is rigid or flexible. It is said that a body is "rigid" if any two points in the material body exhibit no relative displacement; or that such displacement is insignificant enough to not affect the system's dynamic behavior. There are cases; however, where body deformation plays a significant role in its dynamic motion; in these cases the body is considered "flexible". Consequently, Rigid Body Motion is considerably less complex than that of the Flexible type fundamentally because of the lack of this occurrence. Thus, different numerical approaches must take place when using these different types of bodies.

The theoretical background of multi-body dynamics can be easily summarized by the example of the sliding block. The motion of the block can be described using two different frames of references: 1) a relative coordinate system, in which the motion of the block and its position in space can be described with respect of the previously linked body, or 2) a global Cartesian coordinate, in which the absolute position of the block is independently located by a set of Cartesian generalized coordinates. The following figure shows the schematic of this example.

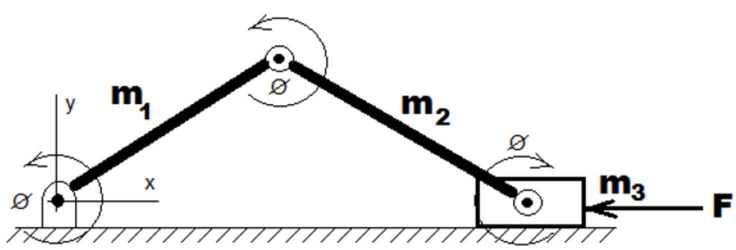


Figure 1 – Schematic of motion of a sliding block

Methods to solve multi-body systems can be categorized into DAE, differential algebraic equations and ODE, ordinary differential equations. Dynamic equations are usually second-order differential equations. In the case of the sliding block, there are three bodies in this simple dynamic system (two rods and one block), and they are connected by linkages that allow rotational and translational degrees of motion. The position and orientation of the block can be resumed (in absolute coordinate system) as follows:

$$q = [x^1, y^1, \emptyset^1, x^2, y^2, \emptyset^2, x^3, y^3, \emptyset^3]$$

The differential algebraic equation of motion that would predict the dynamic behavior of this system could be summarized in the following Lagrange equation:

$$\frac{d}{dt}\left\{\frac{\partial T}{\partial\{\dot{q}\}}\right\}^T-\left\{\frac{\partial T}{\partial\{\dot{q}\}}\right\}^T=\{Q\}$$

Where, T is the kinetic energy of the system and {Q} is the vector of generalized forces in the system. By making use of Lagrange multipliers and Jacobian denominations, we can arrive to the following statement:

$$\left\{\Phi_{(q,t)}\right\}=0$$

Later on, it resolves to the following equations of motion:

$$[M]\{\ddot{q}\}+[\Phi]^T\{\lambda\}=\{Q\}$$

$$\begin{bmatrix} M & \Phi_q^T \\ \Phi_q & 0 \end{bmatrix}\begin{Bmatrix}\ddot{q} \\ \lambda\end{Bmatrix}=\begin{Bmatrix} Q \\ \bar{\gamma}\end{Bmatrix}$$

Note, for further clarification on the formulas one can find many documented articles and papers on the subject of the computation of the equations of motion of rigid bodies. It's clear that the motion of one body is intrinsically dependent on the relative position of the conjoint bodies from the system. A system of multiple bodies will generate a matrix of considerable size, composed of intrinsically dependent equations.

1.2. Case Study: CAMOTRACK tread study

In the rubber track business, there is always a need to improve the performance of the tracks, as well as the durability of them. Rubber, as it is widely known, has many great advantages and it is the material of choice for tires, transportation belts and rubber tracks among many other products. In the case of caterpillar tracks, rubber is basically the only material available capable of providing all that is demanded to do in a track. However, like all manufacturing products it has its shortcomings. Wear and thermal blowout can be typically found in caterpillar rubberized tracks. Wear is normally produced by a combination of factors such as friction, contact pressure and sliding distance. A blowout is generally produced by a combination of cyclic loading (in terms of contact pressure) and temperature. There is also a dependency in the other factors such as topology of the component, and the inherent properties of the rubber compound; such as aging, tan-delta, modulus, etc.

As mentioned earlier, the purpose of this paper was to show the benefits of using VT technology in the research and development of the Camoplast Solideal Inc. tracks. To be more specific, the goal was to create a tool that can be utilized to simulate the tracks in a more dynamic, field-type test, and one that it will help us understand better about the wear and blowout failures found typically on rubberized caterpillar tracks. For this purpose, we chose to use the case study of the tracks that are mounted on an agriculture type tractor (hereafter code name CamoTrack).

There are different rubber tracks that can be mounted to this type of vehicle, but for the study on this paper an 18-ich wide track with wide- tread type design was used. The following figure (Figure 2) shows the vehicle and the rubber track used for both the field and laboratory testing.

Figure 2 – CamoTrack vehicle used for Field testing

Note that for a friction-driven tracks, such as the one in this case study, the main components that will affect the performance of the rubber track will be the vehicle weight, its velocity, ground condition, undercarriage system and the components of the track itself. There are other factors of course, but for this case study these will be the parameters used for the analysis and evaluation of the proceeding tests.

2 Static Analysis – Load distribution

Upon a simple examination of our case study, one can substantiate that the creation of a computer model that will encompass such level of components and details will be an enormous task. Some simplifications and assumptions will need to take place in the MBD model. The most important one that was decided to be done was the elimination of the tractor and all its components from the systems of bodies acting on the track. But in order to do so, we need to replace it by lumped-sum point at its center of gravity.

Therefore, the first procedure in the study of the rubber track for the CamoTrack is to conduct a static analysis to determine the current loading distribution of the vehicle. This will be essential to determine the center of gravity of the vehicle. Once this critical parameter is obtained, the vehicle itself can be replaced by a lumped-sum point in space.

The static analysis will provide the background information to feed the dynamic model.

2.1. Experimental Testing – Weight distribution of CAMOTRACK

An experimental test was conducted in the lab to determine the vehicle weight distribution. The goal, as explained earlier, was to obtain the center of gravity of vehicle. This unique point in space describes the center of mass of the entire vehicle in which all the distribution of the mass is balanced. This point is located in a 3-D space; that is it possesses an x-y-z coordinate position. There are methods that describe how to calculate the center of gravity in all coordinates, but for this early stage in testing, only the x-y coordinates are of interests. The reason behind this assumption is that the MBD model and its center of gravity will not experience much displacement or motion into the z-direction. So, there is no crucial need to determine this parameter.

The way that experimental test was conducted was to elevate the entire vehicle in, and place it in 4 positions. Basically, this set up will allow us to determine the x-y coordinates of the center of gravity. The vehicle would be lifted from the ground to platforms located directly under the front idlers and rear drivers, that way we are guaranteed that the entire weight of the vehicle is distributed exactly to these 4 locations. Figure 3 shows the schematic on how the x-y coordinates of the center of gravity will be placed with regards of these 4-point locations of the vehicle.

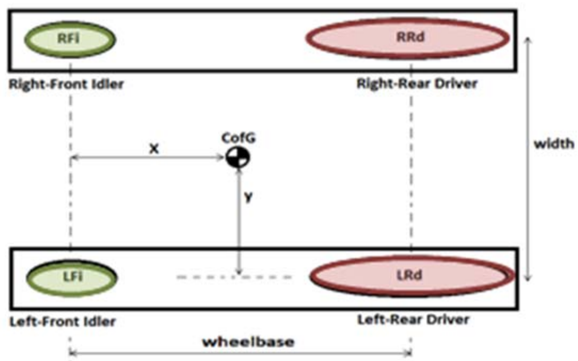
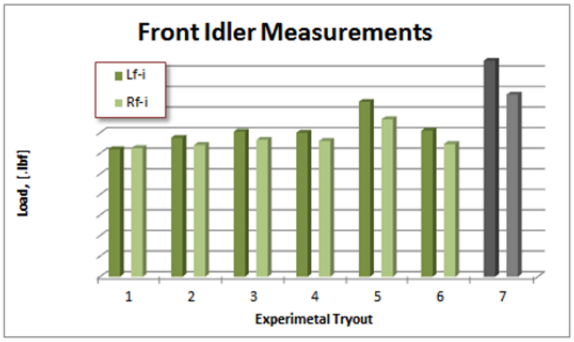


Figure 3 – Schematic to obtain the center of gravity for a vehicle, based in 4-point measurements

For correct measuring, it is important to raise the vehicle enough so that the track, which envelops the entire undercarriage systems (consisting of the multiple idlers, midrollers and drivers components), does not touch the ground, except right under the track sections underneath the idlers and drivers. For this experimental testing we only had access to 2 weight system to calculate the reaction forces, so we could only do two measures at a time.

So, in order to obtain all 4 distributed points, first the weighting scales are placed under the left and right rear drivers (LRD and RRD, as seen on the Figure 3), and a similar fixture with the same elevation are placed on the front left and right idlers (LFI and RFI). Once the data is recorded, then the weighting scales and fixtures are cross-changed so measures can be taken at the front idlers.

The following figures show the weight distributions found for the case when the vehicle is fully loaded. A total of 4 different measurements were carried out. From our test results, we noted there is a considerably divergence in the output of the reaction data (note, that the values have been taken for confidential reasons). This is believed to be due to the difficulty to reproduce for each measurement, the exactly same contact region when elevating the vehicle and holding it in place with the four fixture setups.



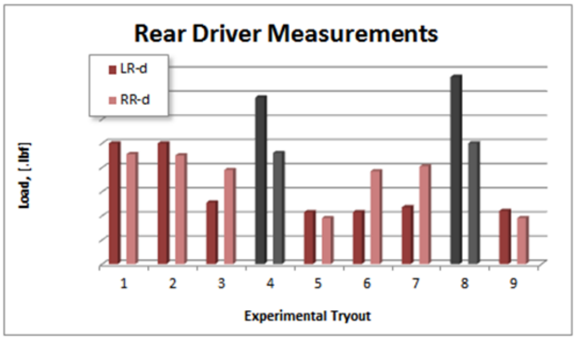


Figure 4 – Idler and Drive weight measurements

As it can be seen from the images, there is a tendency in the weight distribution to lean to the left side of the vehicle (this is more apparent in the driver measurements). Hence, this tells us that the y-coordinate position of the center of gravity is not at the middle of the x-y plane as one would hope (refer to Figure 3). From all these results, it can be assumed that the center of gravity of the vehicle is placed on a [1212.8, 1252.0-mm.] location from the LFI (left-front idler), with a standard deviation of 20-mm and 5-mm, for the x and y coordinate positions respectively.

2.2. FEA Model Description and Material Characterization

As part of our simplification decision matrix, the entire vehicle was omitted from the modelisation of the virtual test. This was done, as we explained earlier on, to reduce complexity in the FE analysis and allow for a more realistic simulation time. Thus, only the track and its undercarriage were to be modeled. In addition, symmetry will be used for the analysis. As we found out in the experimental testing, the center of gravity does not quite lie under the middle of the x-y plane, but for simplification purposes it will be assumed it does. Hence, only one side of the track and undercarriage system of the vehicle will be modeled. The location of the center of gravity of the vehicle, and where the loading (weight) will take effect into the FEA model, will be placed accordingly with the results generated from the experimental test; at a distance of approximately of 1212.8-mm from the front idler.

The modeling and simulation of the tracks and UC were done very similar as in previous simulations done at Camoplast Solideal Inc. All the components of the undercarriage were obtained from the CAD library and assembled correctly in ABAQUS. Similarly, the track was modeled separately and incorporated in the final assembly. The following figure shows the entire CAD model, as it was designed in ProEngineer.

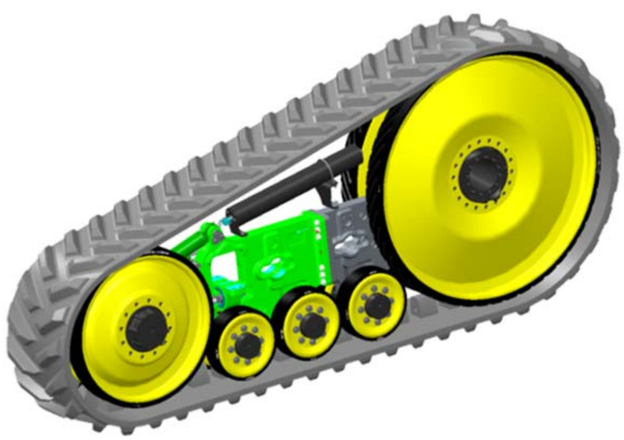


Figure 5 – ProEngineer Assembly CAD model

The final FEA model is of course a much more simplified model; that is, the entire assembly of the CamoTrack was stripped off unnecessary components and detail information; such as bolts, washers, shafts, beams, letterings, etc. Note, that the track was wrapped around the assembly of the UC (as CAD specifications), and not in the "relax" state (ie, not round)

The computer model for the CamoTrack could be categorized into three major categories: Flexural, Rigid and Display Body. Table 1 shows all the components used for the buildup of the FEA model. The computational model (those used for computation) is composed of a total of 15 components. Note that the idlers, midrollers and drivers are described in the table as a set (each component has two parts). Additionally, certain components were used only as display bodies. These components do not provide any analytical weight in the computation of a solution for the virtual test; they are just assembled there for visualization purpose.

Table 1- Component description table for the creation of the FEA model

COMPONENTS	Part type	FEA component Type	Element Type	# of Elements	Material
Main					
Carcass	Flexural	Continuum (hybrid)	C3D8RH	124761	Compound A, D
treads	Flexural	Continuum (hybrid)	C3D8RH	20028	Compound B
Lugs	Flexural	Continuum (hybrid)	C3D8RH	2124	Compound C
Layer - 0deg	Flexural	Truss	T3D2	25080	Steel, Compound D
Layer - bias	Flexural	Membrane	SFM3D4R	3672	Steel, Compound D
Layer - 90deg	Flexural	Membrane	SFM3D4R	4228	Steel, Compound D
Idler set	Rigid	Continuum	C3D8R	2784	Steel
Midrollers (3 sets)	Rigid	Continuum	C3D8R	4896	Steel
Driver set	Rigid	Continuum	C3D8R	2560	Steel
Ground	Rigid	Analytical Rigid	-	-	-
Abridged					
UC Main Frame	Display Body	beam	3D connector	8	-
Shafts	Display Body	hinge	3D connector	6	-
Idler hub & case	Display Body	-	-	-	-
Driver hub & case	Display Body	-	-	-	-
Midroller hub & case	Display Body	-	-	-	-
Front bogie	Display Body	beam	3D connector	1	-
Rear Bogie	Display Body	-	-	-	-

The undercarriage (UC) component which encompasses extremely rigid members was modeled using beam-type connectors (labeled as abridged members in Table 1). The hinge-type connectors were used to model the kinematics of the shafts found on the midrollers. Therefore, the midrollers were linked to the UC by means of hinge and beam connectors. Similarly, both the idler set and driver set were modeled. Note, that the rubber layer in these sets was assumed to cause negligible difference for this simulation case, so they all were modeled as discrete rigid regions.

The following figure, Figure 6, shows the model assembly ready for simulation. Note, the components in white are the display bodies.

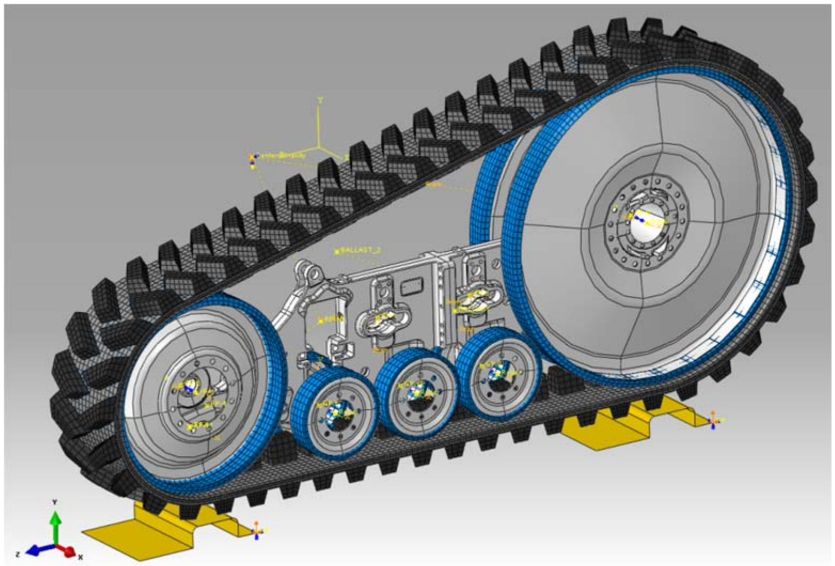


Figure 6 – FEA Assembly of CAMOTRACK static model

The track for the CamoTrack was modeled closely to its manufacturing specifications. The inclusion of the reinforcement information into the analysis is extremely important because it affects both the tensile and flexural capabilities of the rubber track. As described on Table 1, the 0-degree layer was modeled using TRUSS type element, whereas MEMBRANE type element was used to modeled both the bias and 90-degree layers.

A total number of 5 different materials were used in the finite element analysis model. Aside of steel used for the cables as part of the reinforcement, 4 different rubber compounds are used for the manufacturing of the main track components, and are labeled as rubber compound type A, B, C or D.

These rubber compounds generally have different intrinsic properties that are designed to obtain the best mechanical properties for the purpose of its component. Rubber compound A, for instance, is the compound used for the carcass component, and therefore has rubber properties (i.e., flexibility, rigidity, rolling resistance, etc.) best fitted for its use. Rubber compound B and C are the materials used for the construction of the treads and drive lugs, respectively. Rubber compound D is used as the adhesion compound for the reinforcement of the rebar layers.

All these rubber compounds were characterized on a laboratory. Each compound was pre-strained to magnitude found on field testing, and stress-strain curves were generated to define their properties in terms tension, planar and compression (or bi-axial). Then, they were evaluated with Abaqus software to find the best hyperelastic model to fit each rubber compound.

All rubber parts were modeled using hybrid elements. The carcass and lugs were modeled using linear hexahedral reduced elements, C3D8RH. The carcass was portioned into several areas to define the sections where the reinforcement layers were located. These regions were given the material properties of the compound A for carcass and compound D for adhesion (which is used to embed the rebar layer into the carcass). The treads were modeled using C3DR8H as well, with their geometric properties simplified to allow the use of hexahedral elements.

Once all components are assembled as vehicle specifications, and they have been correctly given element properties and material definitions, the next step in the FEA model is to define the interaction properties. Table 2 below shows all the definitions used for the constraint library of the computer model.

Table 2 – Definition table of the constraints for the FEA model

Connector Pairs	Master	Slave	Constraint Type	contact Property
Ground to Treads	Ground	Treads	two-way Contact	steel to rubber
Idler Set to inner track	inner track	Idler	two-way Contact	rubber to rubber
Driver Set to inner track	inner track	Driver	two-way Contact	rubber to rubber
midroller Front set to inner track	inner track	midroller	two-way Contact	rubber to rubber
midroller Midle set to inner track	inner track	midroller	two-way Contact	rubber to rubber
midroller Back set to inner track	inner track	midroller	two-way Contact	rubber to rubber
Lugs to Carcass	Lugs	Carcass	TIE	-
Tread to Carcass	Treads	Carcass	TIE	-
Reinforcement 0-deg	layer 0-deg	Carcass	Embedded	-
Reinforcement bias	layer bias	Carcass	Embedded	-
Reinforcement 90-deg	layer 90-deg	Carcass	Embedded	-

The following figure, Figure 7, shows all the CONTACT interactions (a total of 7 definitions) needed for all the components to correctly interact with each other. From the image, we can see all the "slave" type connection (represented in pink color) coming into contact with the "master" type connections (represented in red color).

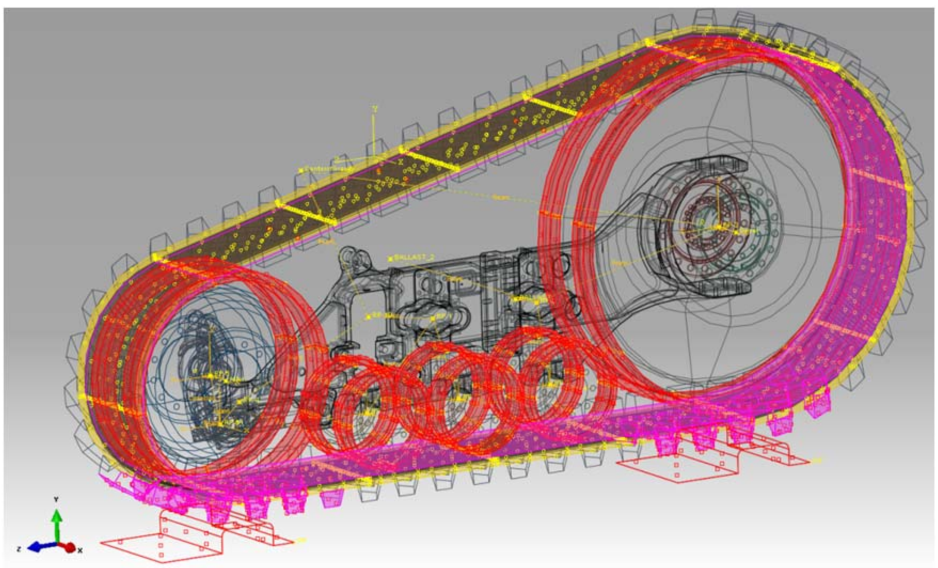


Figure 7 – Contact definitions for the FEA model

Finally, the FEA model was given appropriate steps to model the conditions found on the laboratory and field testing. Note, that initially the entire assembly of components is separated. The simulation will be carried out into 3 steps:

1. Tension STEP, the entire undercarriage is fixed in space and induced displacement is given to the front idler set to produce tension on the track and fix all UC components with the track itself.
2. Contact STEP, the coupled track and UC is now lowered to Ground to force contact between the treads and the ground.
3. Weight STEP, since only one side of the vehicle is modeled (assuming symmetry), only half of the weight will be loaded into the center of gravity of the model.

The problem size is estimated to be approximately of 630,000 degrees of freedom. Simulations were conducted on a dual quad-core computer (3GHz processor, 16GB of RAM, 300GB HDD), using Windows XP 64-bit. The software version was Abaqus 6.12. The solver made use of the parallelization capabilities and 7 cores were used in the solution process. The estimated CPU usage time per simulation was approximately 4.5-hours of wall-clock time

2.3. Validation and Results

Numerous runs were simulated in which the center of gravity is push forward and backward, using the tolerance provided by the deviation on the test results. This was done to obtain a replication of the reaction data. Both the experimental tests and our FEA model provide reaction results that were very sensitive with the location of the center of gravity.

As it was mentioned earlier, the experimental test provided large variations in the force reaction data recorded at the idler and driver regions. These deviations are believed to be due largely by the location upon which the test fixtures (both the fix frame and weight scale fixtures). The variance in contact patch (i.e. region of treads underneath the idlers/driver that hold the weight of the vehicle), along with the correct centered position of the idler/driver with respect to the fixture will produce different results in the reaction data. Hence, both the calculated position of the center of gravity and the reaction data of the idler and driver need to be taken with a degree of uncertainty.

The ground component was modeled to reproduce the same contact regions used on the experimental testing. Note that the tread pattern at the contact region might not be exactly as it was done during the experiments, thus the results will probably varied with the actual laboratory test.

The following figure, Figure 8, shows the results for the FEA calculation of the reactions at the idler and driver components with the same vehicle weight used during the laboratory testing.

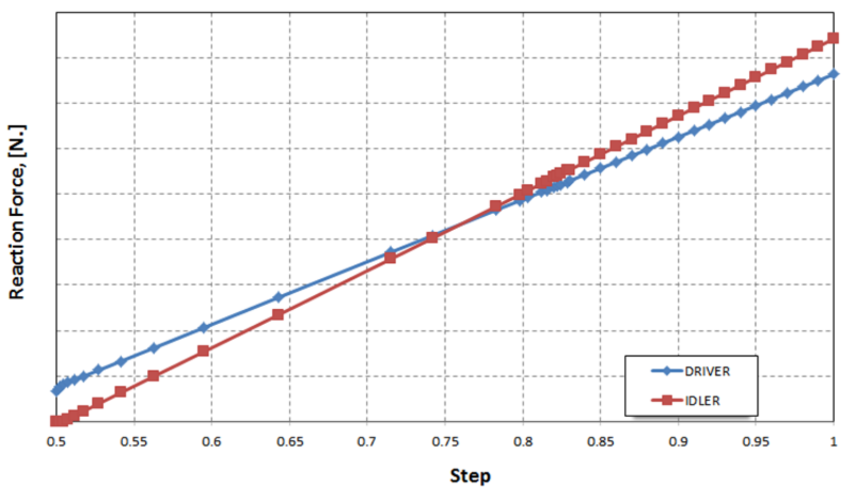


Figure 8 – Idler and Driver reaction data for the FEA model

It is important to mention that the FEA simulation crashed at about 82% of the solution; sometime after a switch in loading distribution occurred between the idlers and the drivers. The reason why this occurs is, of course, because the position of the center of gravity of the vehicle is closer to the idler. At the beginning of the static loading, the contact patch is higher at the Driver section, but as the load increases, the location of the center of gravity becomes more pivotal in the loading distribution than the contact areas of the track with the fixtures. The data after that was linearly extrapolated to meet 100% of the load. Several tries were conducted in order to obtain a full solution, but in all cases it was unsuccessful. The reason of this is because the large rubber deformations at the treads, and complex contact interactions at the ground supports. A likely solution to this issue would be to run the model using explicit code.

These values are within a 5% of margin of deviation from the experimental test results. Furthermore, considering the large deviation in our lab data, we can conclude that the FEA model is representative of the static conditions found on the experimental test.

3 Dynamic Analysis – track and Undercarriage behavior

Once the FEA static model has been validated with laboratory data and it has been determined the right weight distribution and position to place the center of gravity of the vehicle in the FEA model, the next step is to proceed with the creation of a dynamic model. As the goal of this paper was to provide with a system-level analytical model to conduct VT for rubber tracks, it is important to conduct some testing in a dynamic setting.

Therefore, a series of tests were conducted on the field to collect sufficient dynamic data to use for validation of the FEA model. All previous parameters from the static conditions (i.e., track and UC geometry, vehicle weight and tension) were replicated into the dynamic models. Added to these, will be the steady-state vehicle velocity. It important to mention that in this series of tests there are many parameters that will not be fed into the dynamic model. These factors were taken out of the FEA model because they either represent parameters that cannot be measure (i.e., driver effect, ambient and tarmac temperature, etc.), or are believed to not affect the correct dynamic simulation of the rubber track.

The dynamic analysis is discussed next.

3.1. Field Testing – Dynamic analysis

For the studies of the CamoTrack in a more dynamic simulation, it was decided to take advantage of the field testing activities that it was taking place in Yuma, Arizona. The field test was comprised of conducting a series of runs over tarmac, running the vehicle at two different speeds and loading conditions. Other data and parameters were of interest and recorded, but for the purposes of the studies on this paper only the vehicle weight, speed and roller force reaction data will be discussed subsequently.

The following figures, Figure 9 and Figure 10, the test vehicle at rest at the side of the tarmac and the reaction data for the midrollers, respectfully.

Figure 9 – Field Test vehicle at the tarmac, Yuma

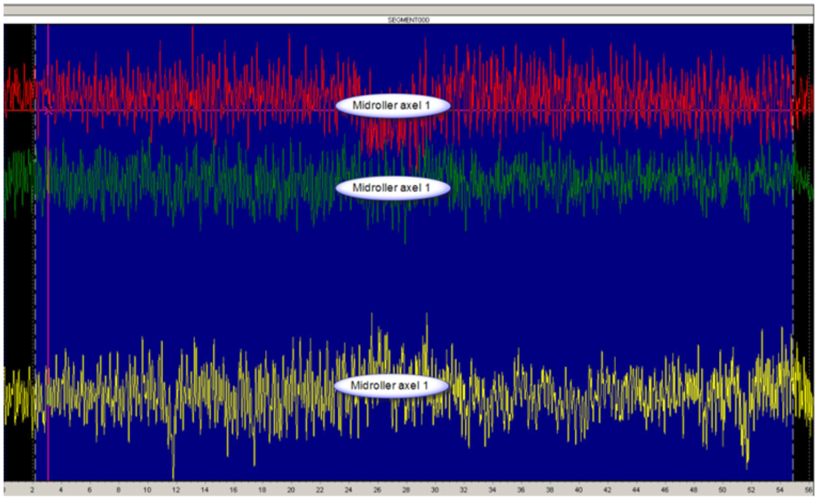


Figure 10 – Steady-State reaction data for midrollers axles

The results from the field test yielded interesting results. First, we found clearly different loading reaction at the midrollers axles. The highest reactions were found at the front roller (red data at Figure 10), whereas the middle roller experienced a mean value a little lower than that of the front roller. Interestingly, the back roller sensed considerably less forces than the first two; approximately half of the front roller.

This data represents the steady-state conditions, at tarmac, with a consisting rolling with the vehicle speed (i.e. not slipping), and the values have been erased for confidentiality reasons as well.

3.2. FEA model Description and Simulation

The FEA modelisation for the dynamic conditions will be similar to that of the static simulation. In fact, the alteration required to fit a dynamic setting will be just an obvious continuation of that model. The simplification of the vehicle omission will be still be used in this model, along with the findings about the center of gravity obtained from the static analysis. Additionally, the ground part of the static model was modified to be a straight, flat region (like the actual tarmac) and long enough for the entire system to move across. Among some of the major changes done to the static model would be:

- Modification to accommodate for mass and rotary inertias to the UC bodies (more display bodies)
- Switch to Implicit Dynamics, and addition of the “Rolling STEP”.

Other small changes in the model would be necessary but in large, these two are the most important ones. Since the bodies for the undercarriage (UC) were all modeled onto the ABAQUS software, and later used only for display bodies, the incorporation to the assembly was done fairly quickly. However, once all the components of the assembly model become fully flexible (both track and UC), the computational time will be tremendous. For this reason, it was decided to reproduce Rigid Body constraints into all the components of the UC. This change to discretize the UC bodies into rigid will not remove the mass and rotary information from the code (i.e. ABAQUS would still automatically calculate them in the solution), and still would considerably reduce the computation because each body’s degrees of freedom would be reduced to just 6.

Some alterations were required to be done, however. A discrete body can only have one representative point (reference node), and in some cases, a UC body would be linked to other bodies from different points. So, in these cases, different sets of the same body were created and then constraint to rigid condition, and linked to each other via MPC type constraint. Henceforth, a body in the undercarriage would still be allowed to be rigid and have different points of connection. The following figure, Figure 11, shows the construction of the entire assembly of UC bodies.

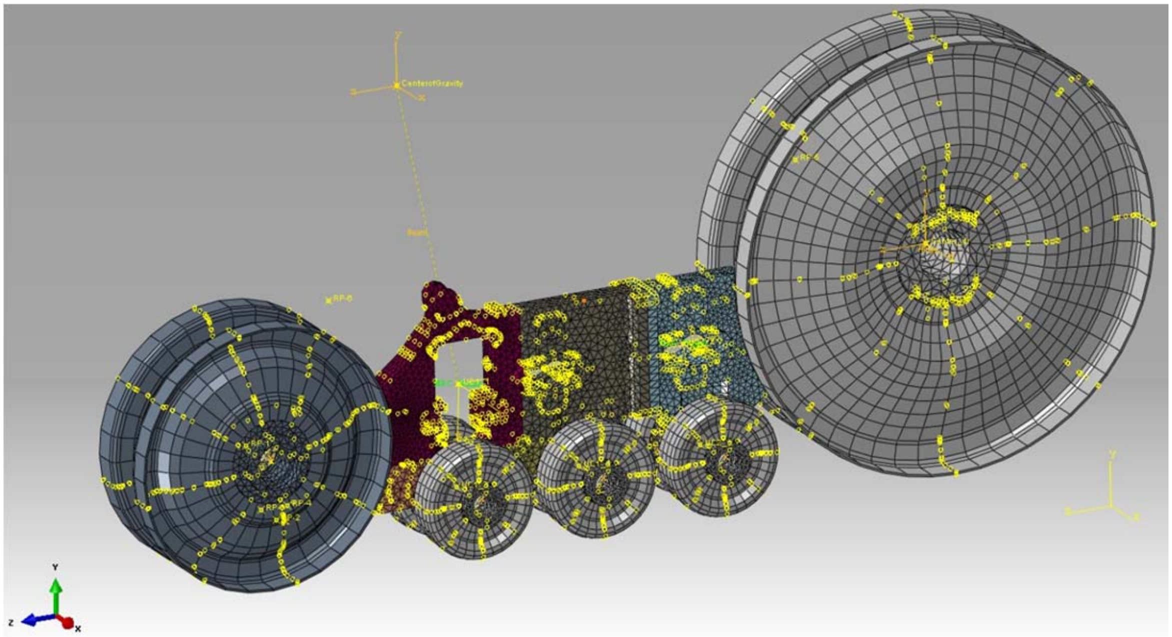


Figure 11 – Assembly of undercarriage components

Once all the major components of the undercarriage are modeled and meshed, the model will take into account their mass and rotary inertias to faithfully reproduce the dynamic motion of the entire assembly. An alternative method to this one would have been to utilize the static model as it was developed and incorporate all the individual mass values and rotary inertias. On the other hand, these values need to be calculated and placed individually at each component's center of gravity, and that would also mean to re-structure all the beam connectivity of the undercarriage (UC) assembly. Therefore, it was decided to use the method mentioned on this paper.

Additionally, it was decided to refine the mesh overall for each component in the assembly. This was done to help convergence, as well to obtain more accurate solution. When all of this was done, with the addition of the elements generated by the incorporation of all the components of the UC, the problem size raised to close 600,000 elements. The total number of variables (degrees of freedom) is now $1.8\text{-}e^6$.

The other major alteration to the static model is the use of implicit dynamics as the solver type. Consequently, an additional step in the solution definition is introduced to model the motion and rotation of the track. Just like in the field test, this step describes the horizontal motion of the vehicle by inducing a velocity parameter to the center of gravity and a rotation to the driver component. In reality, motion of the entire vehicle, is only produced by the torque generated by the motor of the vehicle transferred to the driver components of the undercarriage. The Drivers, as they turn, force the track to roll, by means of friction to the inner side of the carcass. But for simplification and to aid the computation of our model, both the speed of the vehicle and this rotational motion is supplied to the ROLLING step. At this step, a smooth amplitude was added to account for a large acceleration in the first second, and maintain the same speed for the subsequent 5-seconds.

3.3. Validation and Results

Since the problem size has increased considerably from the static model and the solution takes place with the same software/hardware setup, the computational time has increased to about 3 days of wall-clock time.

The following images will show the results of the simulation shortly after the ROLLING step has started. Figure 12 shows the maximum principal strain with a limit of strain set to 10%.

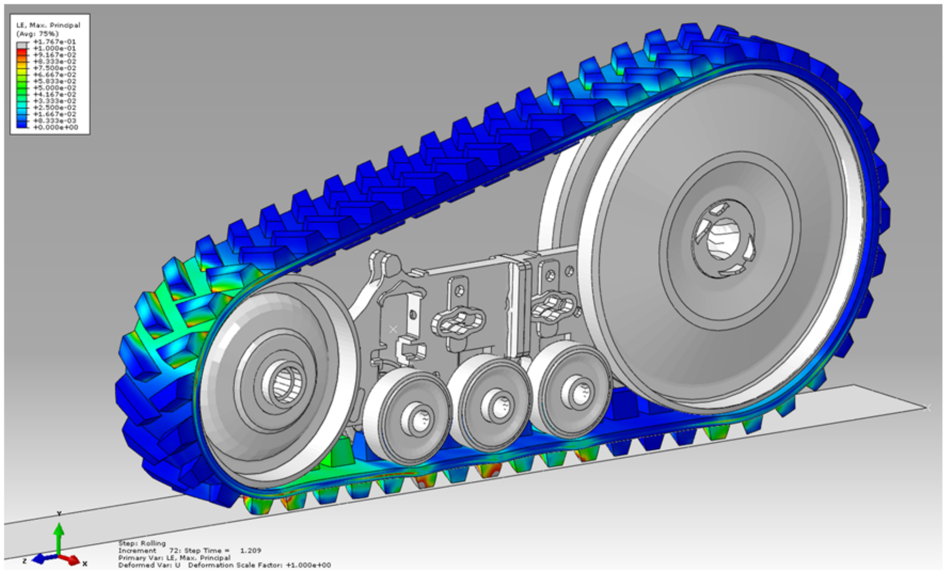


Figure 12 – LE, Max. Principal contour plot for rolling step

The simulation seems to be working properly and accordingly to how it was seen during the field testing. Subsequently, we can now obtain the reaction data from the midrollers to validate if indeed they experience the same loading behavior as seen from the field. The following figure, Figure 13, shows the vertical reaction data for the Front, Middle and Back roller.

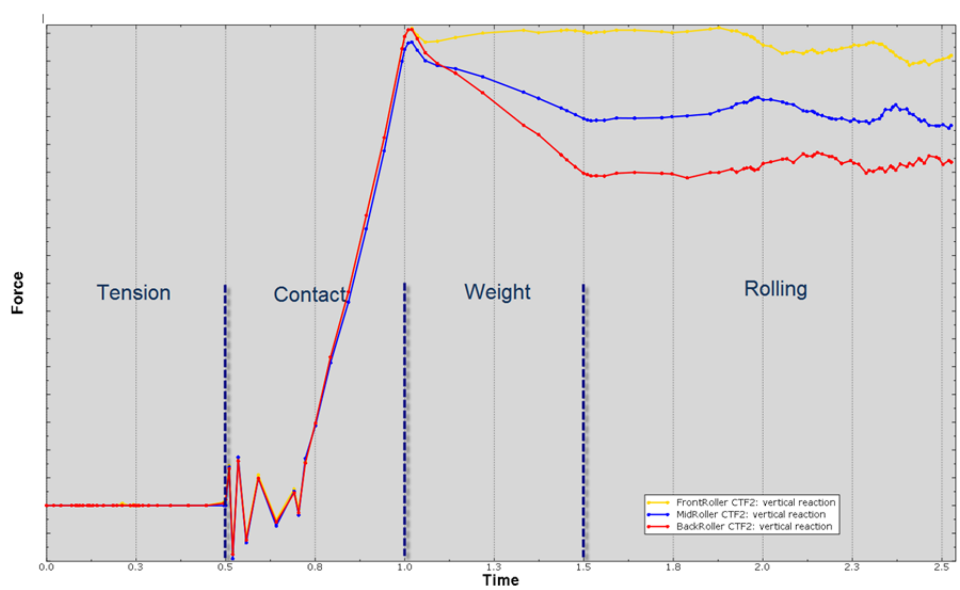


Figure 13 – CTF2; connector force in the vertical direction for all rollers

The data represented in the figure is incomplete; it only shows 1 full second of the Rolling step (5 seconds). However, as we can see from the figure, the Middle roller is not providing a similar response as we have seen on the field tests. Similarly, the overall values with respect to the field test are a bit disparate. The difference with respect to the field test data is about 8%, 12% and 28% for the Front, Middle abd Back rollers, respectively. It can be seen too that as the rolling progresses there seems to be an instability effect acting on the system since the reaction data starts to fluctuate.

This diversion in the data needs to be studied and understood. If we look at the horizontal reaction data for both the Idler and Driver, we can check if the tension is correctly applied. On Figure 14 (on the left) we can see that the reaction history of these components is approximately the value of “A” (note, actual data has been eliminated for confidentiality reasons), which is very close to the value experienced on the field tests. However, if we take a close look at the energies generated by the system (Figure 14, on the right), we can see that there is a big increase of kinetic energy in our system. This suggests that our dynamic model is introducing non-steady state effects in the computation. This tells us that the smooth amplitude applied to the Rolling step might be too abrupt, and it will require a modification in future simulations.

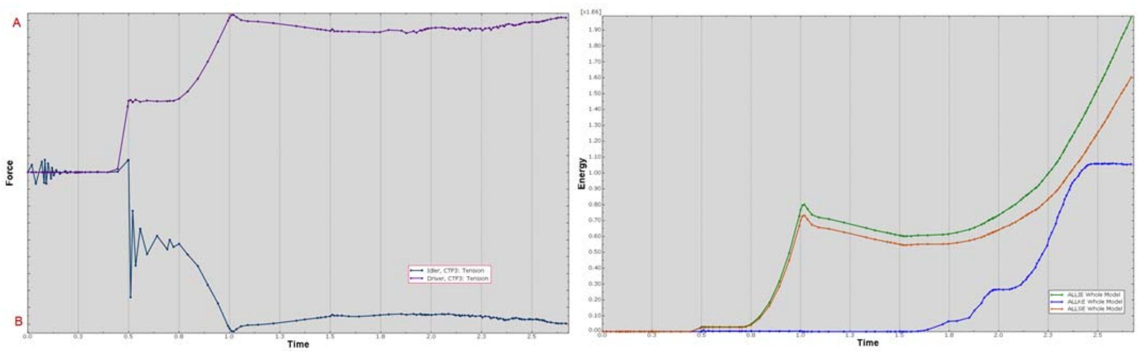


Figure 14 – CTF2; connector force in the vertical direction for all rollers

Other factors could cause the divergence of the overall results of the reactions at the midrollers, as well as the disturbances as the track rolls over the ground. This matter will take place in future simulations.

4 Conclusion and Future work

The use of multi-body dynamics (MBD) has proven to be an attractive and noteworthy method to study rubber tracks dynamically. MBD can helps us understand more about our structure and its mechanical behavior in a dynamic setting, but also in a static simulation. In fact, the simulation of the rolling of the track can be considered a quasi-static simulation event. The results provided by the static simulation of the MBD proved to be very accurate, and match the data from the laboratory testing. On the other hand, the simulation of the dynamic rolling of the track proves to be much harder to reproduce and validate, due to the large number of parameters affecting the outcome of the solution. Validation with field data will require further work, and this will take over the course of the ongoing simulations done at Camoplast Solideal Inc.

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