

Finite element model of tire UM FETire

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33. UM module for simulation of flexible tire dynamics

33.1. General information about UM FETire

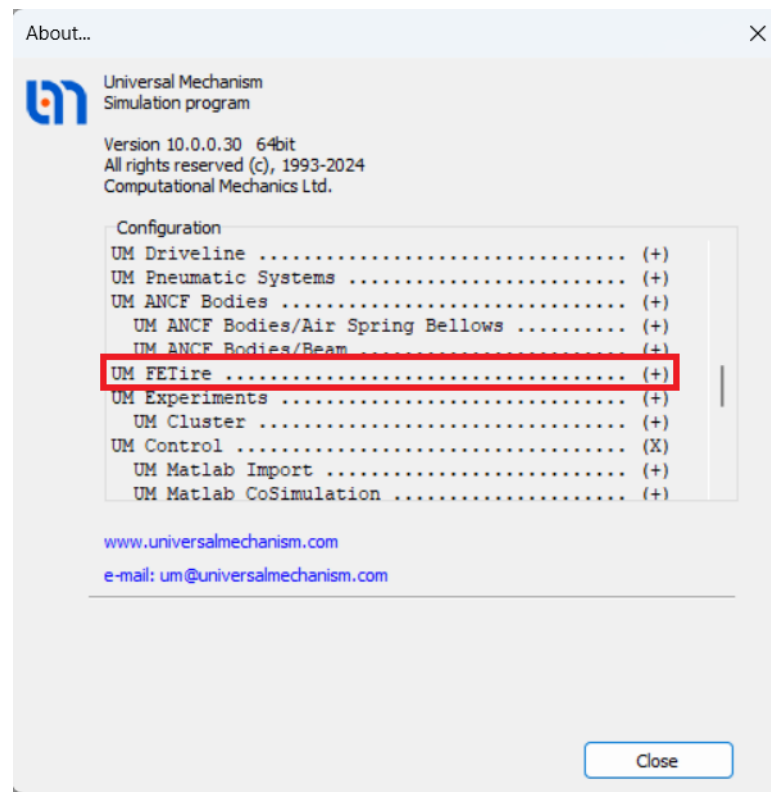


Figure 33.1. List of UM modules in 'About' window

The **UM FETire** module contains tools for development and using finite element models of flexible tires. The flexible tire can be used in simulation of dynamics of road vehicles, monorail trains, and airplane landing gear.

To check whether the current version of UM includes this module, run the **UM Input** program, select the menu command **Help | About...** and check that in the window that appears there is a (+) sign after the name of the UM FETire module.

The development of a flexible tire model requires the user to have a detailed description of the parameters of the tire's multilayer material.

The current version of UM offers the use of a flexible tire in the following main areas:

- Calculation of the curves of the massless tire model used in standard simulation of the dynamics of a wheeled vehicle (tabular model and TMEasy);
- Use for modeling vehicle dynamics in non-standard situations: hitting a curb, hitting a stone, driving through a damaged part of the road, etc.;
- The movement of an SUV on the ground with sinkage.
- Modeling of tire wear.

33.2. Fundamentals of mathematical modeling of a flexible tire in the UM FETire module

33.2.1. Single plate finite element model

In the UM, a flexible tire is modeled by a shell using a trapezoidal finite element. The mathematical model of the element was developed by the authors of the module and is described in detail in the papers [1] [2]. The papers are in the public domain and can be obtained via links [paper1](#), [paper2](#). The finite element model takes into account the multi-layer nature of the tire material [3].

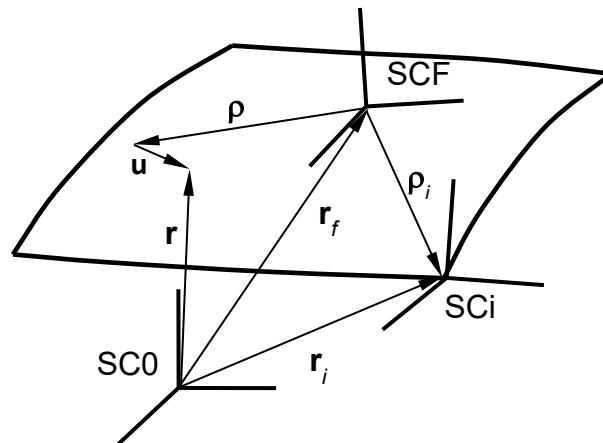


Figure 33.2. Finite element of a plate in absolute coordinates

The shell is modeled by flat finite elements (FE), in our case, trapezoidal. Let us consider a FE model, Figure 33.2. Four nodes are placed at the vertices of the trapezoid. Each node is associated with a coordinate system (SC_i in the figure) that has six degrees of freedom relative to the inertial coordinate system (SC_0): three coordinates of the origin (vector r_i) and three orientation angles of SC_i relative to SC_0 (angle vector π_i). A sequence of rotations 3,1,2 is used to set the orientation. These are the so-called absolute coordinates of the nodes. Thus, a free FE has $6 \times 4 = 24$ degrees of freedom.

Along with absolute coordinates, Craig-Bampton coordinates are introduced [4]. For this purpose, arbitrary spatial motion of the FE is decomposed into motion as a rigid body together with the floating coordinate system (SCF, Figure 33.2) and small elastic displacements relative to the SCF, specified by elastic forms of the FE. SCF has 6 degrees of freedom: three coordinates of the origin (vector r_f) and three orientation angles (angle vector π_f). Thus, there are $24 - 6 = 18$ elastic modes, which are the natural modes of free oscillations of the FE and are calculated using mass and stiffness matrices constructed using the classical finite element method. The coordinates corresponding to the elastic modes of the element are designated as Δq . Examples of bending modes of free vibrations of a trapezoidal finite element are shown in Figure 33.3.

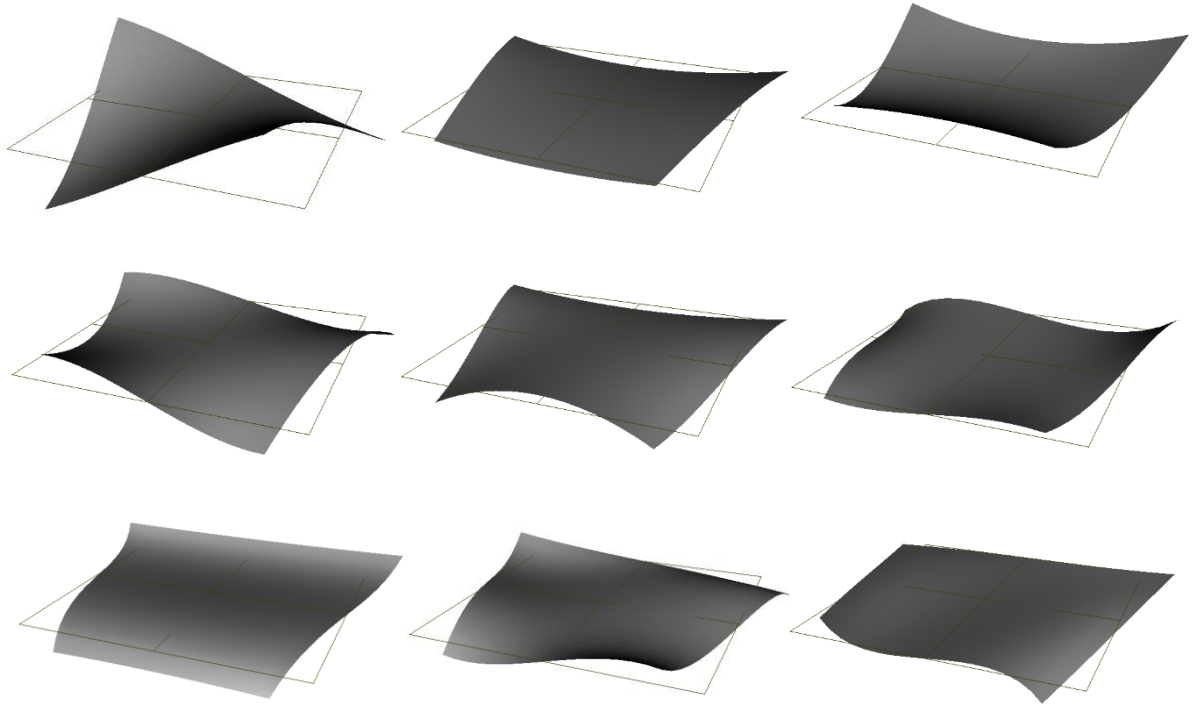


Figure 33.3. An example of bending modes of FE corresponding to natural modes of free vibrations

The equations of motion for each of the finite elements are constructed in accordance with the Craig-Bampton method [4] in the Craig-Bampton coordinates.

$$\begin{aligned}
 m a_f - m \tilde{\rho}_c \varepsilon_f + M_{r q} H \Delta \dot{q} &= -\tilde{\omega}_f \tilde{\omega}_f \rho_c \\
 m \tilde{\rho}_c a_f + J \varepsilon_f + M_{\pi q} H \Delta \dot{q} &= -\tilde{\omega}_f J \omega_f \\
 H^T M_{r q}^T a_f + H^T M_{\pi q}^T \varepsilon_f + \Delta \ddot{q} &= -\Omega^2 \Delta q - (\delta \Omega + \beta \Omega^2) \Delta \dot{q} - H^T V(\omega_f) \\
 H^T &= (H_{r i}^T \quad H_{\pi i}^T)
 \end{aligned}
 \tag{33.1}$$

Here $a_f, \varepsilon_f, \omega_f$ are the acceleration of the origin, angular acceleration and angular velocity of the SCF; Δq are the elastic modal coordinates of the plate relative to the SCF; m, J, ρ_c are the mass, the matrix of the inertia tensor and the radius vector of the center of mass of the plate relative to the SCF; H is the matrix of the coefficients of the elastic modes of the FE; Ω, Ω^2 are the diagonal matrices of the natural frequencies of the FE and their squares.

Equation (33.1) uses the matrix form of writing equations and relations of kinematics. You can get acquainted with the features of this style in the book [5]. The book can be downloaded from the [link](#).

The term $\Omega^2 \Delta q$ in the third equation of system corresponds to the elastic forces arising during deformation of the element, and the term $(\delta \Omega + \beta \Omega^2) \Delta \dot{q}$ introduces internal dissipation into the element. Two types of internal dissipation are taken into account: the dissipation proportional to the stiffness matrix is set by the parameter β , and the parameter δ specifies the damping ratio for each of the elastic modes of the element. The assignment of values to this parameter is discussed later. Here we only note that $\delta \sim 1, \beta \ll 1 \text{sec}$. Finally, the vector $V(\omega_f)$ includes the inertial forces depending on the angular velocity of the wheel.

After this, the transition from the Craig-Bampton variables to absolute nodal coordinates is carried out. The number of Craig-Bampton coordinates and absolute nodal coordinates is the same. They are connected by a system of nonlinear algebraic equations, which should be solved with respect to the Craig-Bampton variables for given absolute nodal coordinates at each step of integrating the equations of motion:

$$\begin{aligned} r_i &= r_f + A_{0f}(\pi_f)(\rho_i + u_i), \quad u_i = H_{ri}\Delta q \\ A_{0i}(\pi_i) &= A_{0f}(\pi_f)A_{fi}(H_{\pi i}\Delta q) \\ i &= 1..4 \end{aligned} \quad (33.2)$$

Here r_i, A_{0i} are the radius vector of the origin and the matrix of direction cosines of SC_i relative to SC₀.

The first and second derivatives of these equations with respect to time give equations for the relationship between the absolute velocities v_i, ω_i and accelerations a_i, ε_i of nodes and the velocities and accelerations of the Craig-Bampton variables.

$$\begin{aligned} v_i &= v_f - (\tilde{\rho}_i + \tilde{u}_i)\omega_f + A_{0f}H_{ri}\Delta\dot{q} \\ \omega_i &= \omega_f + A_{0f}B_{fi}H_{\pi i}\Delta\dot{q} \\ a_i &= a_f - (\tilde{\rho}_i + \tilde{u}_i)\varepsilon_f + A_{0f}H_{ri}\Delta\ddot{q} + \tilde{\omega}_f\tilde{\omega}_fA_{0f}(\rho_i + u_i) + 2\tilde{\omega}_fA_{0f}H_{ri}\Delta\dot{q} \\ \varepsilon_i &= \varepsilon_f + A_{0f}B_{fi}H_{\pi i}\Delta\ddot{q} + \tilde{\omega}_fA_{0f}\omega_{fi} + A_{0f}\dot{B}_{fi}H_{\pi i}\Delta\dot{q} \\ i &= 1..n \end{aligned}$$

Eliminating the accelerations $a_f, \varepsilon_f, \Delta\ddot{q}$ from equations (33.1) using these relations, we obtain the equations of motion of an individual FE in absolute nodal coordinates

$$M_k\ddot{q}_k + k_k(q_k, \dot{q}_k) = Q_k, \quad (33.3)$$

where k is the element index, q_k is the vector of absolute coordinates of nodes of size 24, k_k, Q_k are the columns of inertial forces and generalized forces, M_k is the mass matrix of the finite element of size 24×24 relative to the inertial coordinate system.

33.2.2. Use of relative node coordinates

The equations of motion of a separate finite element (33.3) are written in absolute coordinates of the nodes, i.e. in coordinates that determine the position of the nodes' SC relative to the inertial SC₀. When modeling the dynamics of a tire, i.e. in the process of integrating the equations of motion, it is more effective to use the node's coordinates relative to an absolutely rigid wheel rim.

Let $\Delta r_i^r, \Delta\pi_i^r$ be the vectors of the node displacement and the rotation angles of the node SC during tire deformation relative to the wheel rim SC, i.e. these values are equal to zero in the undeformed state of the tire. Then the position SC_i of the node relative to the inertial SC₀ is determined by the equations

$$\begin{aligned} r_i &= r_w + A_{0w}(r_{i0}^r + \Delta r_i^r), \\ A_{0i} &= A_{0w}A_{wi}(\Delta\pi_i^r), \end{aligned}$$

where r_w, A_{0w} are the radius vector and the matrix of direction cosines of the coordinate system associated with the rigid wheel rim; r_{i0}^r is the radius vector of the node relative to the SC of the wheel rim in the undeformed state of the tire. We also present expressions for the velocities:

$$v_i = v_w + \tilde{\omega}_w A_{0w}(r_{i0}^r + \Delta r_i^r) + A_{0w}\Delta\dot{r}_i^r,$$

$$\omega_i = \omega_w + A_{0w}\omega_{iw}^r.$$

Here v_w, ω_w are the velocity of the wheel center and its angular velocity, ω_{iw}^r is the angular velocity of SCi relative to the wheel rim.

33.2.3. Multilayer finite element

33.2.4. Model of wheel with flexible tire (FE tire)

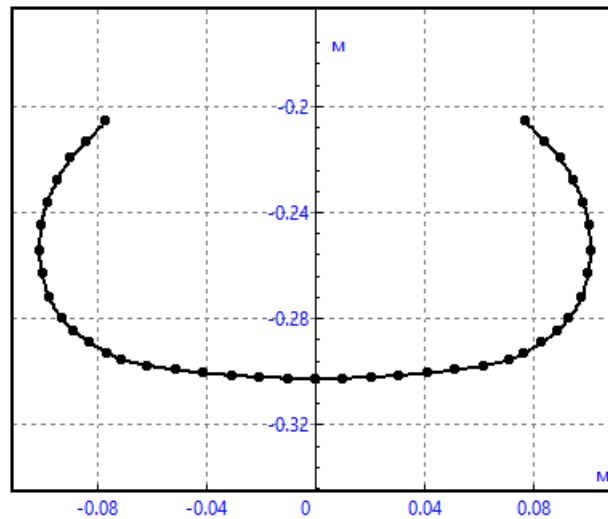


Figure 33.4. Example of a curve of a flat section of a shell modeling a tire

The wheel is modeled by an absolutely rigid wheel rim with six degrees of freedom, with which a finite element model of an elastic shell is connected (FE tire). Since the shell in the undeformed state is a surface of revolution, the FE meshing is performed by rotating a plane section around a horizontal axis of symmetry. n_s segments (generally speaking, of variable length) are introduced on the section curve (the tire profile) and, accordingly, n_s+1 nodes, Figure 33.4. The section rotates n_α times around the axis of symmetry by a constant angle $\Delta\alpha = 2\pi/n_\alpha$ forming a uniform mesh into $n_s \times n_\alpha$ equilateral trapezoids or finite elements, Figure 33.5. The number of nodes is $(n_s + 1)n_\alpha$. The boundary nodes of the shell are rigidly connected to the wheel rim on both sides. The remaining $(n_s - 1)n_\alpha$ nodes have six degrees of freedom each, as described in the previous section. In each of the n_α sections, the nodes are numbered sequentially along the tire profile

$$1 + (n_s + 1)(i - 1) \dots 1 + (n_s + 1)i,$$

where i is the section number $i = 1 \dots n_\alpha$.

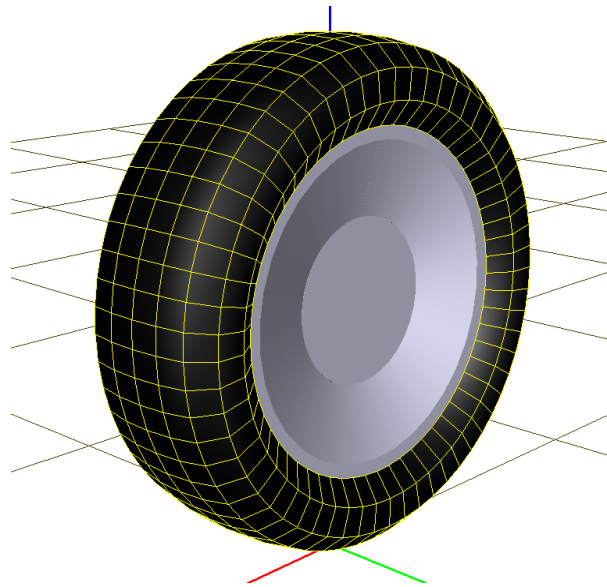


Figure 33.5. A car wheel with an elastic tire in the input program. Trapezoidal finite elements are highlighted

The equations of motion of the elastic shell are formed using the equations for each of the elements (33.3) in a standard manner for FEM.

Complete equations of motion for a wheel with an elastic tire include equations for a the rigid rim and shell:

$$M\ddot{q} + k(q, \dot{q}) = Q, \tag{33.4}$$

where q is a coordinate column of size $6(n_s - 1)n_\alpha + 6$, M is the mass matrix of the wheel model, and k, Q are columns of generalized inertial and active forces.

To improve the profile of the model matrices, the indices of the six coordinates corresponding to the rigid rim are located at the end of the list of coordinates.

33.2.5. Contact interaction of tire with road

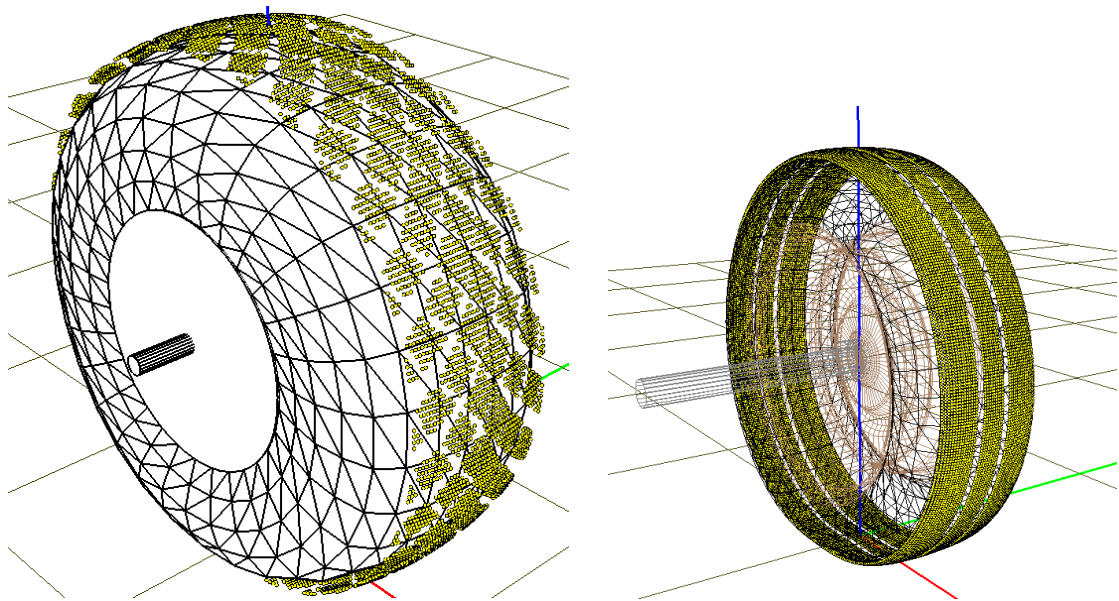


Figure 33.6. Examples of a brush model of wheel contact with the road

A brush model is implemented for contact interaction of the tire with the road surface. The contact surface of the tire is represented by a set of elastic massless bristles. One end of a bristle is connected to the FE and the other one can come into the contact with the supporting surface realizing elastic-dissipative interaction in the radial direction and friction in the tangent one. Examples of a brush model for a wheel of a heavy-duty and passenger car are shown in Figure 33.6. The yellow dots correspond to the ends of the bristles making contact. As it is seen from the picture, the bristles allow taking into account the tire tread pattern.

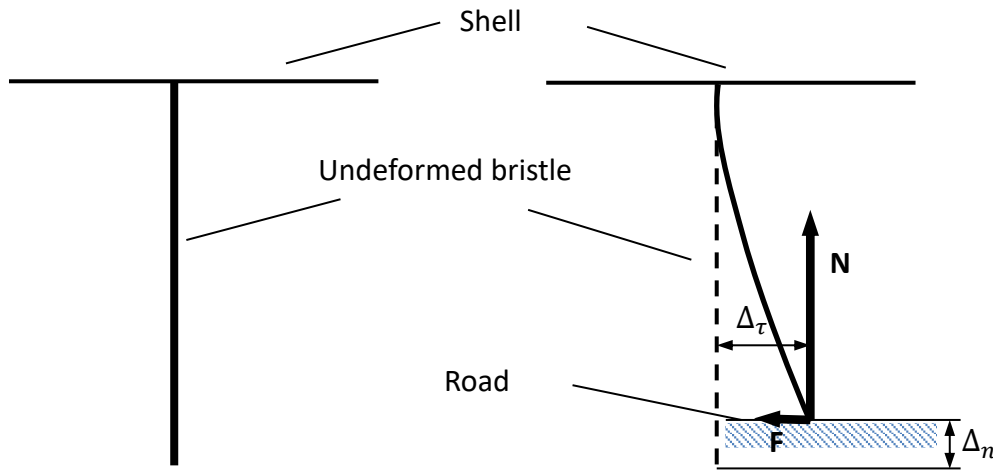


Figure 33.7. Brush contact model

Let us consider a mathematical model of the forces of interaction between a bristle and the road, Figure 33.7. The normal force N depends on the magnitude of bristle compression in the normal direction Δ_n and on its time derivative $\dot{\Delta}_n$:

$$N = k_n \Delta_n + d_n \dot{\Delta}_n, \tag{33.5}$$

where k_n, d_n are the longitudinal stiffness of the bristles and the damping constant.

The friction force F has two modes: sliding and sticking,

$$F = \begin{cases} fN, & \text{sliding} \\ k_\tau \Delta_\tau + d_\tau \dot{\Delta}_\tau \leq f_0 N, & \text{sticking} \end{cases} \tag{33.6}$$

Here f, f_0 are the sliding and static friction coefficients; Δ_τ is the deviation of the end of the bristle from the undeformed state in the tangential direction; k_τ, d_τ are the shear stiffness of the bristle and the damping constant.

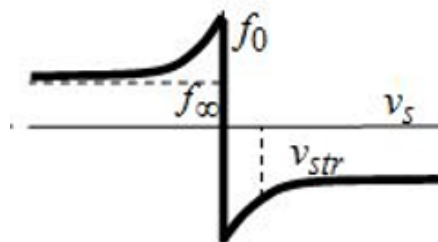


Figure 33.8. Dependence of the friction coefficient on the sliding velocity

The sliding friction coefficient depends on the sliding velocity v_s in accordance with the Stribeck formula (Figure 33.8):

$$f(v_s) = f_\infty + (f_0 - f_\infty)e^{-(v_s/v_{str})^\delta} \tag{33.7}$$

where

f_0, f_∞ are the coefficient of friction values at zero and infinite sliding velocities;

v_{str} is the Stribeck velocity, which determines the value of the velocity interval at which the Stribeck effect of an exponential drop in the friction coefficient takes place;

$\delta \in [0.5, 1]$ is the empirical exponent depending on material.

33.2.6. Tire pressure

The air pressure in the tire is taken into account for each FE in the form of a distributed load perpendicular to the element. A compensating force and moment are applied to the wheel rim so that the total force and moment from the pressure for the wheel as a whole are equal to zero.

33.2.7. Reduced tire models

Since the model described above has a large number of degrees of freedom (usually several thousand), the important task is to reduce this number in order to speed up the simulation.

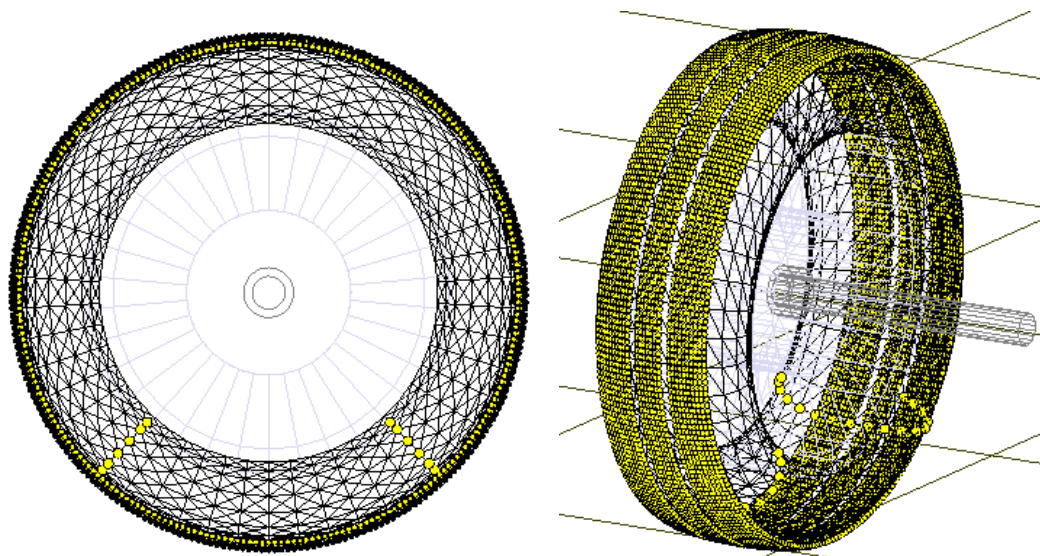


Figure 33.9. Interface nodes for reducing the upper part of tire

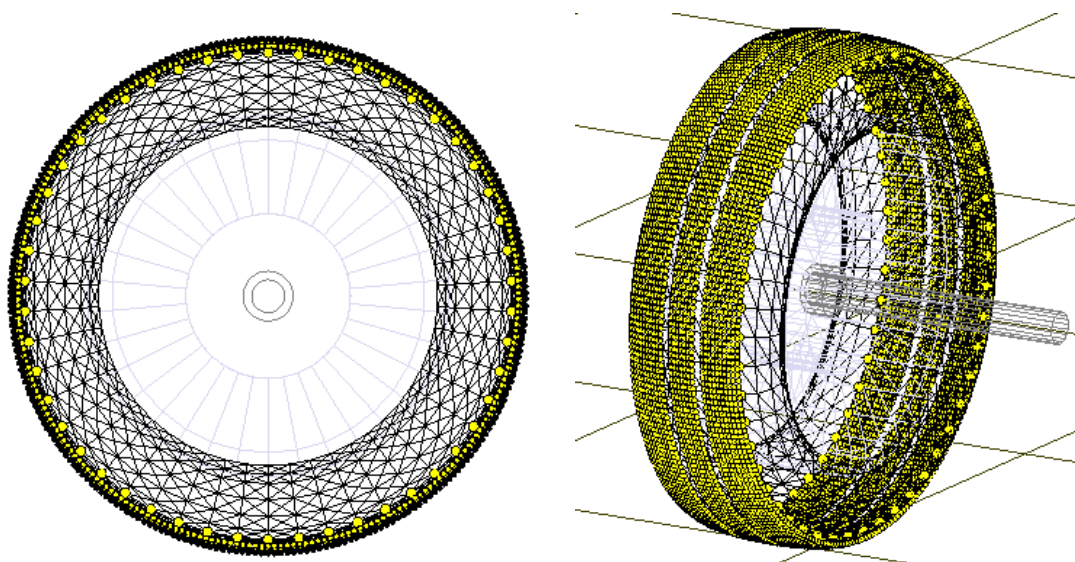


Figure 33.10. Interface nodes for reducing the side parts of tire

UM implements two **reduced** tire models that have a significantly smaller number of degrees of freedom than the full model described above: a model with a reduced **upper part** (Figure 33.9) and **side parts** (Figure 33.10). Both approaches use the same idea: to express the linear and angular displacements, velocities and accelerations of nodes included in the reduced regions through the displacements, velocities and accelerations of **interface nodes**. Thus, the degrees of freedom of the reduced nodes can be excluded from the equations of motion (33.7). In the figures above, interface nodes are marked with bold yellow dots. In the first case, the reduced area is defined by two angles of the left and right sections relative to the vertical (α_{start} , α_{end}) containing interface nodes; the values of the angles α_{start} , α_{end} can be different, for example $\alpha_{\text{start}} = 60^\circ$, $\alpha_{\text{end}} = 45^\circ$. In the case of reduction of the side parts, the interface nodes are located along the border of the side walls.

The Craig-Bampton method applied to the reduced areas of the tire is used for the practical implementation of reducing the number of degrees of freedom. The equations of motion of the reduced areas are expressed through the coordinates of the interface nodes and the wheel rim. This is possible because the interface nodes and the wheel rim form a continuous boundary of the reduced areas. We use only static forms of interface nodes in the Craig-Bampton method.

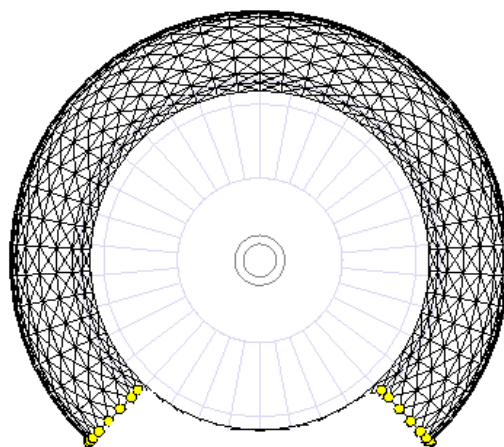


Figure 33.11. Reduced upper part of tire

Let us consider the implementation of the idea in a highly simplified form.

Consider an example in Figure 33.11 that shows the reduced upper area of the tire. Let us fix the wheel rim and linearize the equations of motion of the tire relative to the equilibrium position. Using the stiffness matrix, calculate the static displacements of the reduced nodes with a unit increment of each coordinate of the interface nodes separately. In matrix form, these dependencies can be written as follows:

$$\Delta q^r = G \Delta q^{\text{in}}, \quad (33.8)$$

i.e. the displacements of the reduced nodes Δq^r linearly depend on those of the interface nodes Δq^{in} .

Please note that this relationship was obtained for displacements relative to the wheel rim as a rigid body. Next, we accept similar relationships for relative velocities and accelerations

$$\Delta \dot{q}^r = G \Delta \dot{q}^{\text{in}}, \Delta \ddot{q}^r = G \Delta \ddot{q}^{\text{in}} \quad (33.9)$$

After that the absolute coordinates, velocities and accelerations of the reduced nodes can be expressed via the coordinates, velocities and accelerations of the interface nodes and the wheel rim. This leads to the construction of the equations of motion of the reduced region in terms of coordinates of the interface nodes and to the exclusion of the reduced d.o.f. from the equations of motion (33.4).

- Note 1.** The described approach is approximate, as in any case of using the Craig-Bampton procedure. The user, having the opportunity to use the full tire model along with the reduced ones, can evaluate the difference in the modeling results due to the introduced simplifications.
- Note 2.** Reducing the **upper part** of the tire makes it possible to reduce the number of degrees of freedom several times but it has significant disadvantages: when the tire rotates, the equations are rebuilt after turning by an angle $\Delta\alpha = 2\pi/n_\alpha$ (the angle between adjacent sections), which leads to an additional loss of accuracy and decrease of the integration step size. When hitting a high step, the contact may go beyond the area of the active nodes, which will lead to incorrect modeling results. For this reason, this type of reduction is recommended for use in static problems when the wheel does not rotate, in cases of low-speed driving or in cases where the accuracy of the simulation does not play a special role (for example, when calculating tire wear).
- Note 3.** The tire model with a **reduced sidewalls** has **increased rigidity** in load deflection tests at rated and increased loads. A modified model has been implemented that eliminates this problem, Sect. 33.3.13.4.4 *Evaluation of the parameters of a modified tire model with a reduced sidewall*
- Note 4.** To identify the model type, abbreviations are often used in directory and file names: **f** (full - full model), **ru** (reduction upper), **rs** (reduction side).

33.2.8. Estimation of the influence of damping proportional to the stiffness matrix on eigenvalues

Let us consider the linearized equations of a wheel with a flexible. We fix the wheel rim to eliminate zero frequencies.

At the first stage of the study, we set the pressure equal to zero. In this case, the stiffness matrix K_0 of the model is symmetrical and the linearized equations with external forces turned off have the form

$$M\ddot{q} + D\dot{q} + K_0q = 0,$$

where D is the damping matrix.

Consider the damping proportional to the stiffness matrix

$$D = \beta K_0.$$

The damping coefficient β is included in the equations of a separate FE of the shell (33.1) and has the time unit.

Consider the eigenvalue problem

$$(M\lambda^2 + \beta K_0\lambda + K_0)v = 0. \quad (33.10)$$

Here, λ , e are the eigenvalue and eigenvector to be determined. Along with this problem, consider the problem of calculating the natural frequencies of the model, which is written as follows:

$$(K_0 - M\omega^2)e = 0,$$

where ω , e are the natural frequency and the corresponding natural mode of vibration.

Let the problem of calculating frequencies be solved, $\omega_j, e_j, j = 1, 2, \dots$ is the set of frequencies, ordered in ascending order, that is, ω_1 is the lowest natural frequency of the tire. Since the wheel rim is fixed, there are no zero frequencies corresponding to movement as a rigid body.

$$(K_0 - M\omega_j^2)e_j = 0.$$

Let us present an important property of natural modes, which consists in their orthogonality in the norms of the mass and stiffness matrices:

$$e_k^T M e_j = \begin{cases} m_j > 0, k = j \\ 0, k \neq j \end{cases}, \quad e_k^T K_0 e_j = \begin{cases} k_{0j} > 0, k = j \\ 0, k \neq j \end{cases}$$

$$\omega_j^2 = \frac{k_{0j}}{m_j}$$

We will look for eigenvectors of problem (33.10) in the form of an mode expansion

$$v = \sum c_j e_j$$

with complex coefficients c_j ,

$$\sum c_j (M\lambda^2 + \beta K_0\lambda + K_0)e_j = 0.$$

Multiplying this equation on the left by e_j^T and taking into account the properties of orthogonality of modes, we obtain scalar quadratic equations

$$m_j \lambda_j^2 + \beta k_{0j} \lambda_j + k_{0j} = 0,$$

or

$$\lambda_j^2 + \beta \omega_j^2 \lambda_j + \omega_j^2 = 0.$$

The solution to this equation has the form

$$\lambda_{j1,2} = -\frac{\beta\omega_j^2}{2} \pm \sqrt{\frac{\beta^2\omega_j^4}{4} - \omega_j^2}. \quad (33.11)$$

Let us introduce the critical frequency ω^* at which the radical expression becomes zero:

$$\omega^* = \frac{2}{\beta}. \quad (33.12)$$

For all eigenfrequencies greater than the critical one, the eigenvalues are negative real numbers (33.11), that is, frequencies that are higher than the critical frequency are completely damped and do not appear in the dynamic simulation.

For frequencies below the critical one we have two complex conjugate roots

$$\lambda_{j1,2} = -\frac{\beta\omega_j^2}{2} \pm i\omega_j \sqrt{1 - \frac{\beta^2\omega_j^2}{4}}$$

Here i is the imaginary unit.

Let us introduce an indicator of the degree of frequency damping: the damping ratio

$$\delta_j = -\frac{\text{Re}\lambda_j}{\omega_j} = \frac{\beta\omega_j}{2} = \frac{\omega_j}{\omega^*}. \quad (33.13)$$

Then the complex eigenvalues are given by the formula

$$\lambda_{j1,2} = -\delta\omega_j \pm i\omega_j\sqrt{1 - \delta^2}.$$

In this expression, the imaginary part determines the frequency of damped oscillations

$$\omega'_j = \omega_j\sqrt{1 - \delta^2} < \omega_j. \quad (33.14)$$

Now let us discuss the effect of pressure. Instead of the stiffness matrix, it is now more correct to talk about the matrix of positional forces K , since the pressure forces are non-conservative (non-potential) and the matrix K of linearized forces independent of velocities is not symmetric,

$$K = K_0 + \Delta K(p).$$

Here $\Delta K(p)$ is a non-symmetric addition to the matrix from pressure p . In addition, damping in the tire model is constructed using the K_0 matrix without taking into account the influence of pressure. For these reasons, the previous analysis can only be considered an approximation. However, the studies below show the importance of these estimates for understanding the damping rate of the tire model when using damping proportional to the stiffness matrix.

Based on the research results, the following conclusion can be drawn. Damping proportional to the stiffness matrix has an important positive effect: it eliminates the influence of high frequencies. For example, at $\beta=0.001\text{s}$ frequencies above the critical $\omega^* = 2000\text{rad/s} \approx 318\text{Hz}$ are suppressed. The damping ratio is easily assessed by the frequency using formula (33.13). These properties make it possible to use this type of damping as the main one when modeling the dynamics of a wheel with an elastic tire.

33.3. Development and study of flexible tire model

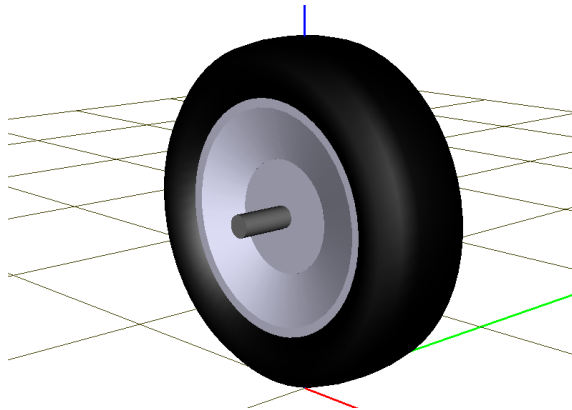


Figure 33.12. Template of a wheel with flexible tire in UM Input program

To create a flexible tire and to study its properties, a standard template is used contained in the *TireTestRig template* model. **In fact, the template is a virtual model of the tire test rig.**

The model is located in the directory [{Data UM}\Samples\FETire\TireTestRig template](#)

To create a new tire model, you must create a copy of the template (i.e. copy this directory), rename it according to the tire name and make the necessary adjustments to the model, first in the input program and then in the simulation program.

A tire template is a model of a virtual tire test rig and is used

- to create a tire by specifying geometric data, parameters of the multilayer material, description of the tread,
- to study some properties of the model (natural frequencies, damping level, deflection in the vertical, longitudinal and transverse directions depending on the corresponding load, etc.),
- to calculate the curves of the massless tire model used in standard modeling of the dynamics of a wheeled vehicle (tabular model and TMEasy),
- to create the reduced tire models;
- to create a finite element tire model which can be included in the model of a wheeled vehicle (car, monorail, motorcycle, etc.).

The main problem when creating models is that the parameters of the tire material, as a rule, are not publicly available and can be obtained from the tire manufacturer.

The template contains a standard wheel subsystem with a flexible tire, a hub and an auxiliary body **Ground**, which specifies the movement of the supporting surface under the wheel. The connection of the wheel rim with the hub is modeled by a bushing element. The joint, which sets the position of the hub relative to CK0, allows performing basic tests with the tire model including setting the values of the toe and camber angles, setting rotations according to the desired law, and so on.

33.3.1. Tire template in UM Input program

Consider the elements included in the tire template in the UM Input program.

33.3.1.1. Wheel with flexible tire as a standard subsystem

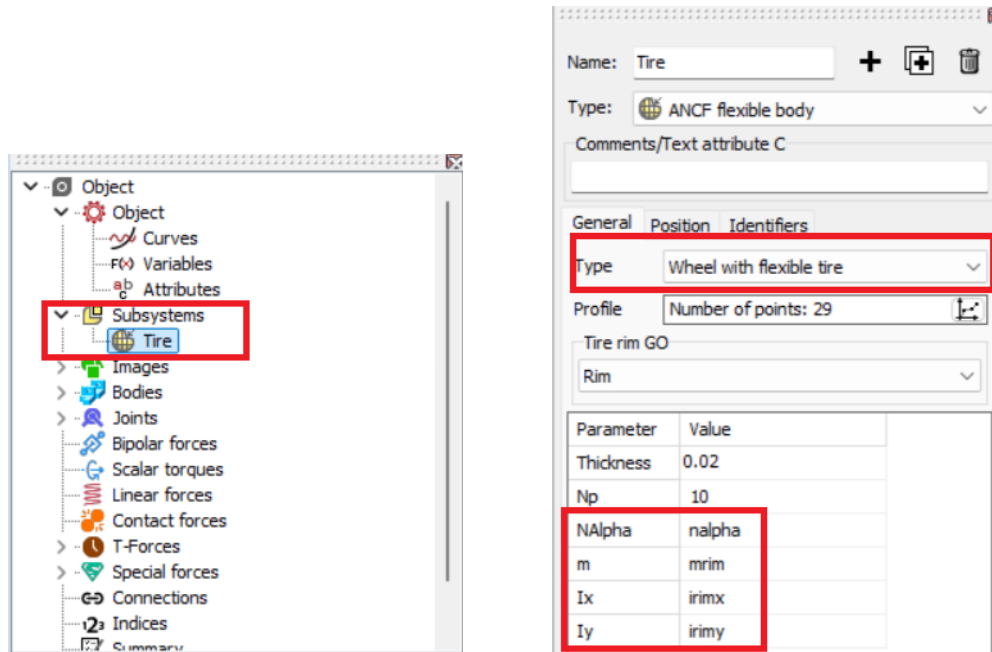



Figure 33.13. Wheel with flexible tire as a standard subsystem

To create a wheel with a finite element tire, you should add the **ANCF flexible body** subsystem and set its type to **Wheel with flexible tire**, Figure 33.13. An **external** subsystem is added to the model, which includes one absolutely rigid body - a wheel rim. The flexible tire is added to the wheel in the UM Simulation program, but in the input program it is represented only by its image. The tire image as a surface of rotation is specified by the section (profile) using the button , Figure 33.14.

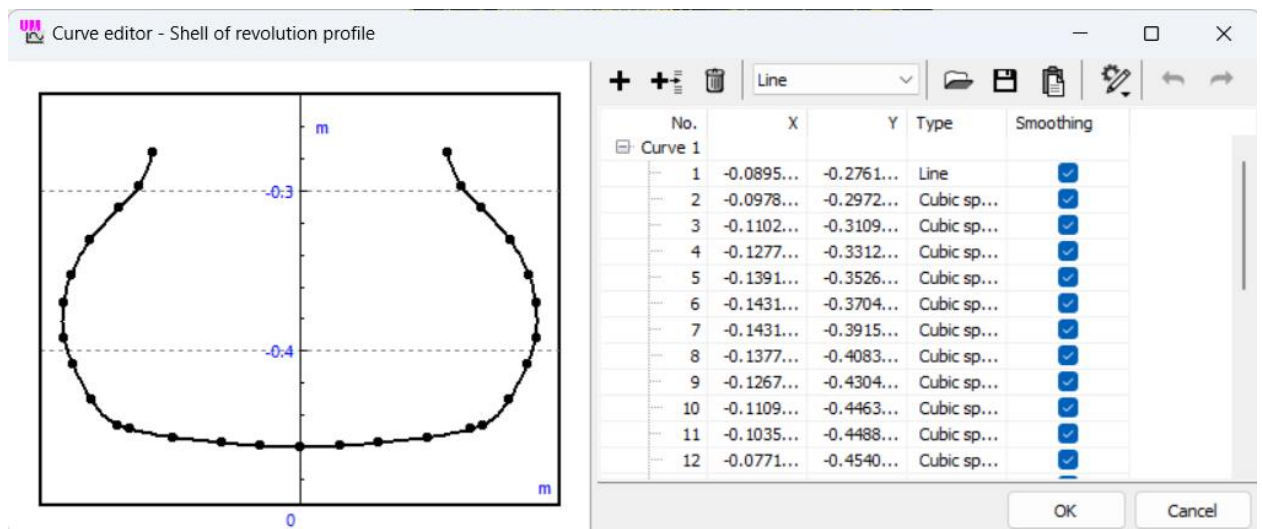


Figure 33.14. Tire cross-section curve (profile)

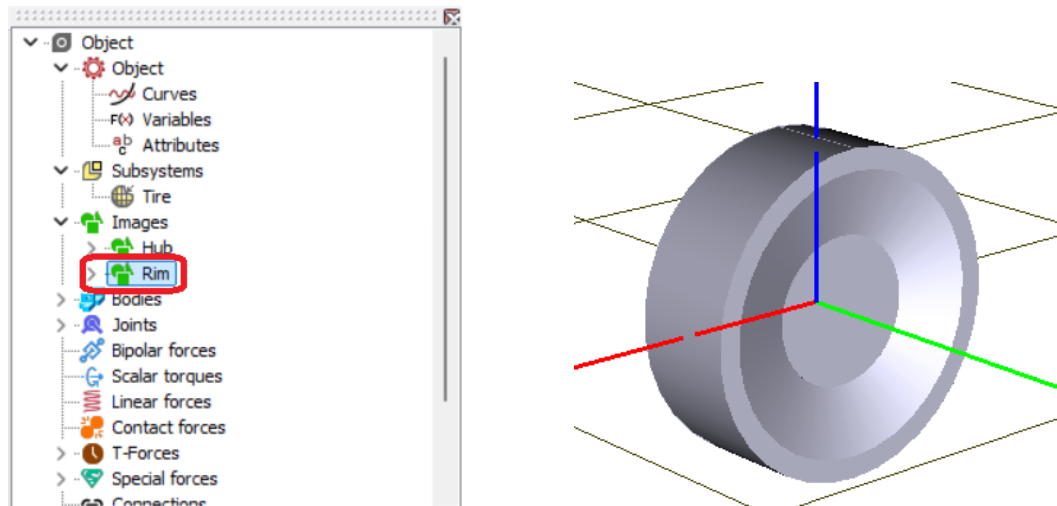


Figure 33.15. Graphic object for wheel rim

A graphic object (GO) of a wheel rim is assigned to the subsystem in the **Tire rim GO** section (Figure 33.15) by selecting a preliminary created GO, Figure 33.13. When creating a specific tire, the user can use his own GO, created in one of the CAD systems. In the template, we propose a simplified parameterized GO that can be used with any tire size, Figure 33.16. Identifiers are used to parameterize the GO

- r_{rim_out} – outer radius of rim,
- r_{rim_in} – inner radius of rim,
- w_{rim_out} – maximal width of rim,
- w_{rim_in} – minimal width of rim.

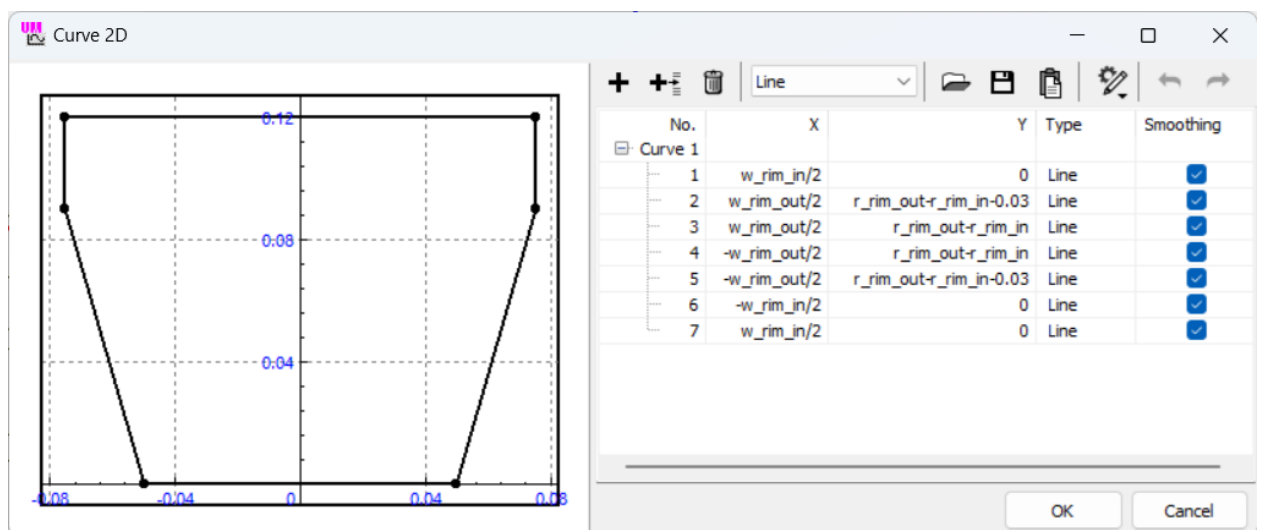


Figure 33.16. Parameterization of simplified image of wheel rim

Flexible tire parameters specified in the input program are listed below (Figure 33.13).

N_p is the number of finite elements in the tire section. Used only in the input program for the tire GO. Ignored in the simulation program.

$NAlpha$ is the main parameter that specifies the number of finite elements around the circumference of the tire. Must be parameterized with an identifier for possible modification.

m, I_x, I_y – inertial parameters of the wheel rim – mass, moments of inertia relative to the longitudinal (I_x) and transverse axis of symmetry (I_y). In the standard template, these parameters are specified by the identifiers $mrim, irimx, irimy$.

33.3.1.2. Bodies and joints

The model additionally includes two bodies: a hub (**Hub**) and an auxiliary body **Ground**, simulating a rotating roller of a test rig that simulates the movement of the wheel on the road.

33.3.1.2.1. Hub

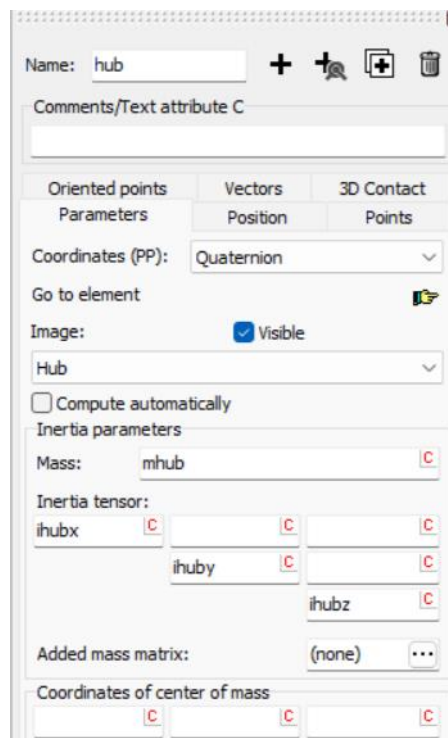


Figure 33.17. Body Hub and its inertia parameters

The inertial parameters of the hub are specified by identifiers, Figure 33.17. The position and movement of the hub is determined by joint **jHub** of the generalized type. In particular, the joint introduces three degrees of freedom: movement in the longitudinal direction and vertically, as well as rotation around the axis of the wheel.

Consider seven elementary transformations (ET) including in the joint description, Figure 33.18.

1) TCz – constant shift along Z axis

The shift specifies the position of the center of the hub and rim above the road surface using the wheel radius identifier $rwheel$ and the additional shift identifier $h0$. Additional displacement allows the user to obtain the initial position of the wheel above the road.

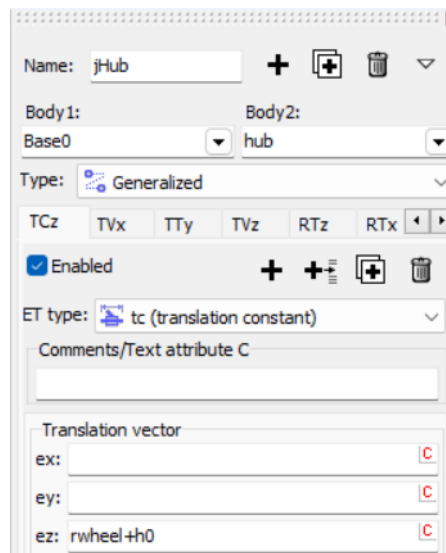


Figure 33.18. List of elementary transformations. The first ET: TCz – constant shift along the vertical direction

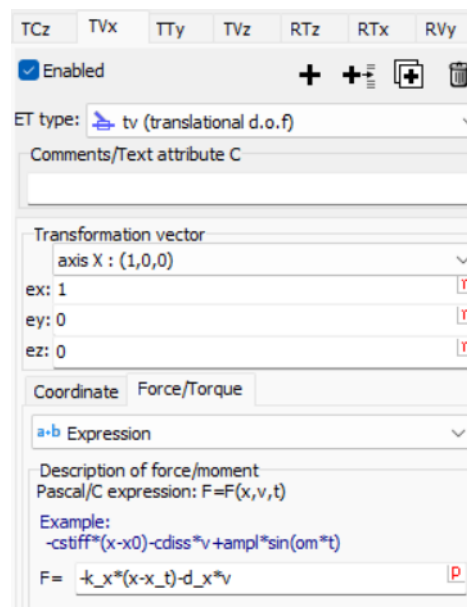


Figure 33.19. the second ET: TVx – degree of freedom in longitudinal direction.

2) TVx – degrees of freedom along X axis

This ET implements the displacement of the hub in the longitudinal direction in accordance with the law specified by the x_t identifier. To implement the displacement, a linear elastic-dissipative joint force is introduced, specified by the expression $-k_x(x-x_t)-d_x*v$, Figure 33.19. In this expression, k_x , d_x are the stiffness and dissipation coefficients, x is the longitudinal coordinate of the hub. When the x_t identifier value is zero, the force blocks the longitudinal displacement of the wheel. When values of identifiers k_x , d_x are zero the wheel can move freely in the longitudinal direction.

Change of the x_t identifier value in *standard wheel tests* is used to determine the tire's longitudinal stiffness.

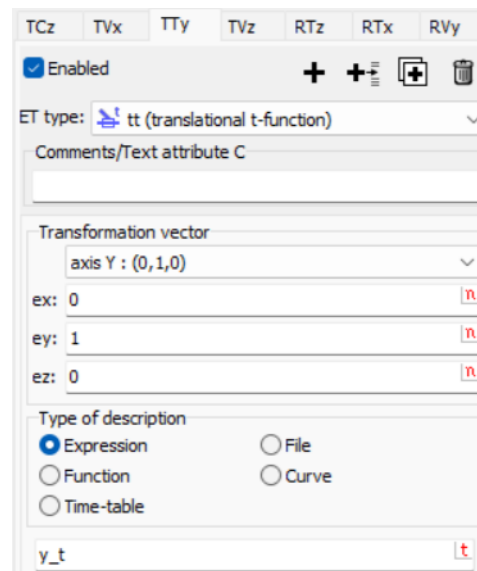


Figure 33.20. The third ET: TTy – shift in lateral direction as a function of time

3) TTy – shift along Y axis as a function of time

This ET implements the hub displacement in the lateral direction/ Th shift is specified by the identifier y_t . The change in the y_t identifier in standard wheel tests is used to determine the tire's lateral stiffness.

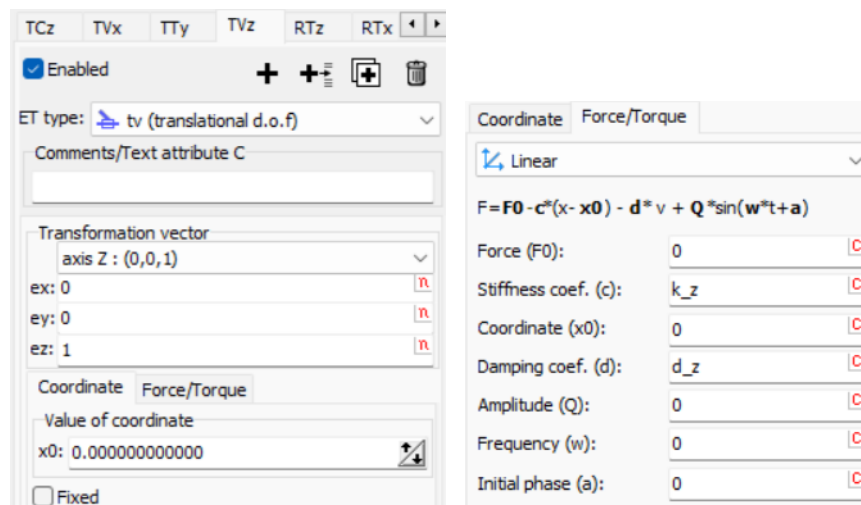


Figure 33.21. The fourth ET: TVz – degree of freedom in vertical direction

4) TVz – degrees of freedom along Z axis

The vertical degree of freedom is used to implement a given tire load parameterized by the *tireload* identifier. The linear joint force introduces an elastic-dissipative force with stiffness and damping coefficients specified by the identifiers k_z , d_z , Figure 33.21. A non-zero value of the stiffness coefficient is used solely to block the vertical degree of freedom, which is necessary when calculating frequencies and when constructing reduced models. Otherwise, this identifier is zero. The dissipation coefficient usually has a non-zero value, except for the test with free fall and wheel rebound when analyzing the degree of damping of the tire model, Sect. 33.3.13.3 *Estimation of internal damping by coefficient of restitution of bouncing tire*.

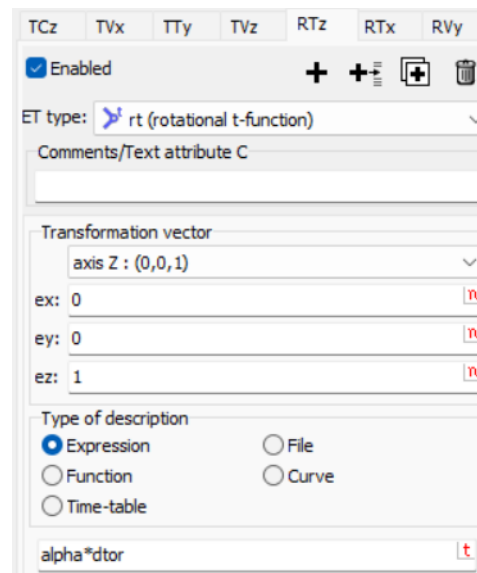


Figure 33.22. The fifth ET: RTz – rotation about the vertical axis as a function of time

5) RTz – rotation about Z axis as a function of time

The angle of rotation about the vertical axis is specified by the alpha identifier in degrees, Figure 33.22. The standard identifier $dtor = \pi/180$ converts degrees to radians. The rotation is used in tests mainly to set the lateral slip of the tire.

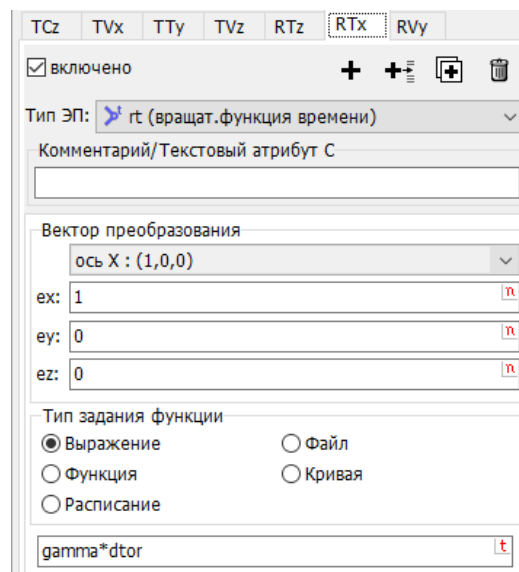


Figure 33.23. The sixth ET: RTx – rotation about the longitudinal axis as a function of time

6) RTx – rotation about X axis as a function of time

The angle of rotation about the longitudinal axis sets the camber angle in degrees with the $gamma$ identifier, Figure 33.23. The standard identifier $dtor = \pi/180$ converts degrees to radians.

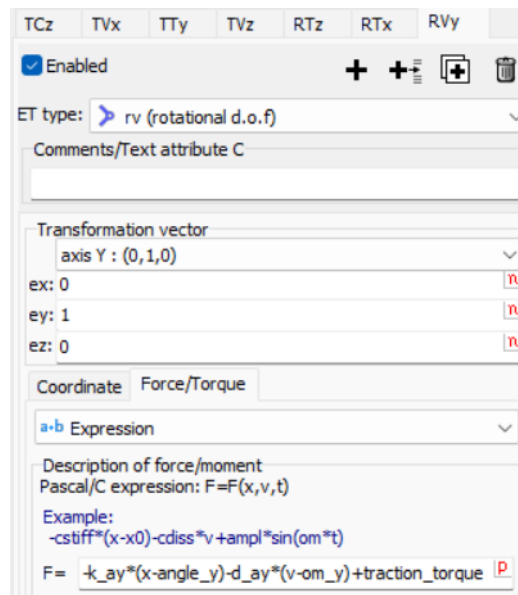


Figure 33.24. The seventh ET: TRy – degree of freedom corresponding to the wheel rotation about its axis

7) RVy – degree of freedom corresponding to the wheel rotation about its axis

Features of using this degree of freedom are determined by the joint torque (Figure 33.24)

$$-k_{ay}*(x-angle_y)-d_{ay}*(v-om_y)+traction_torque$$

This expression introduces the linear elastic-dissipative moment with the stiffness coefficient k_{ay} and damping coefficient d_{ay} as well as the traction torque specified by the identifier $traction_torque$.

The joint torque allows implementing the following modes of wheel rotation:

- **Free rotation**, including application of a traction torque by the identifier $traction_torque$. In this mode, the stiffness identifier k_{ay} should be set to zero, and the identifier d_{ay} (at zero om_y) approximately determines the steady angular velocity $traction_torque/d_{ay}$.
- **Blocking of wheel rotation**, identifiers $angle_y$, om_y , $traction_torque$ are zero, stiffness and dissipation coefficients are non-zero. This mode is used when calculating frequencies and generating reduced tire models.
- **The mode of rotation of the wheel, specified by the dependence of the angular velocity on time**. The angular velocity is set by the om_y identifier, and the rotation angle is specified by the $angle_y$ identifier. Since the angle of rotation is equal to the time integral of the angular velocity, the variable $angle_y_t$ equal to the integral of the identifier om_y is added to the list of model variables, Figure 33.25. In some of the tests described below, this variable is assigned to the identifier $angle_y$ in the UM Simulation program using the identifier control tool.



Figure 33.25. List of variables in the wheel template model

33.3.1.2.2. Auxiliary body Ground

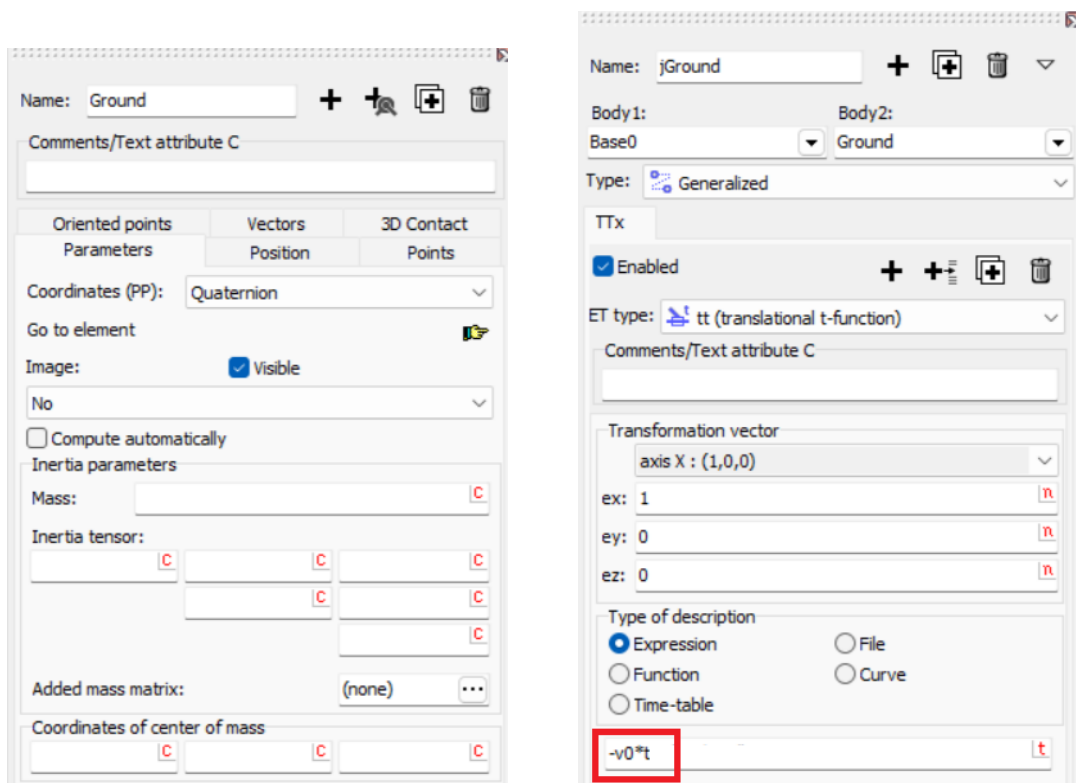


Figure 33.26. Auxiliary body Ground

The auxiliary body **Ground** must be included in the template model (test rig). This body simulates a rotating roller of the test rig. For this body, inertial parameters and a graphic image are not specified, Figure 33.26 left. The joint in Figure 33.26 on the right introduces the motion of the body along the X axis at a non-zero speed, given by the identifier $v0$. When the speed is positive, the motion is directed against the X axis. The contact interaction of the tire in the template model is carried out with the Ground body. When this body moves, the wheel begins to rotate, although the wheel axis does not move in the longitudinal direction, as in the case of the test rig.

33.3.1.3. Coupling wheel with hub

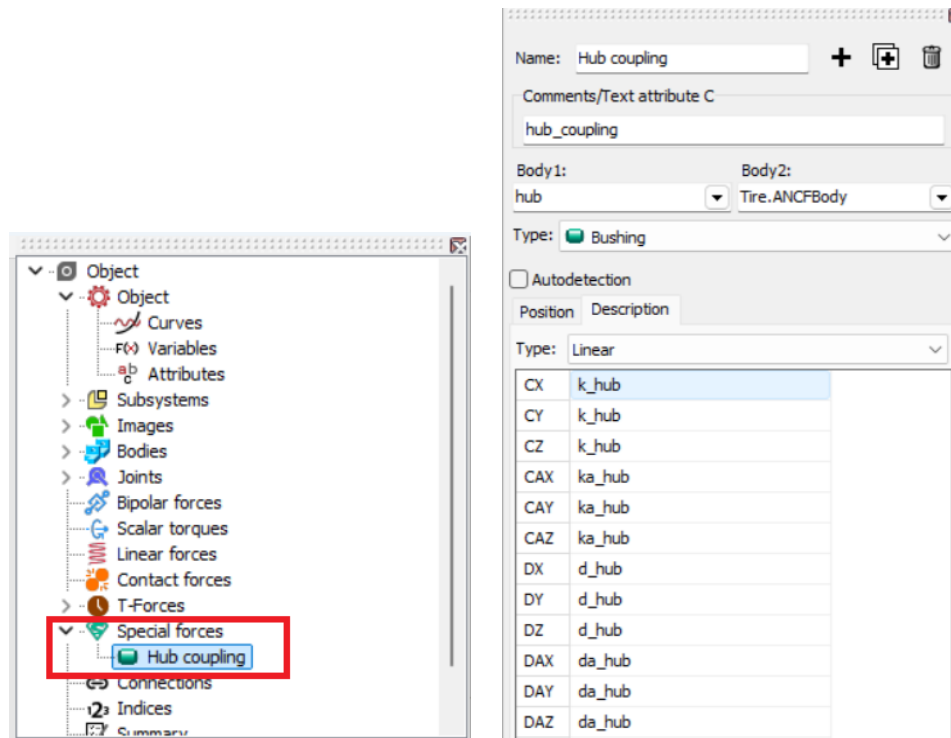


Figure 33.27. Wheel/hub coupling

To attach the wheel to the hub, a special force element **Hub coupling** is used, which is of the bushing type, Figure 33.27. The stiffness and damping coefficients are specified by identifiers.

33.3.1.4. Wheel load

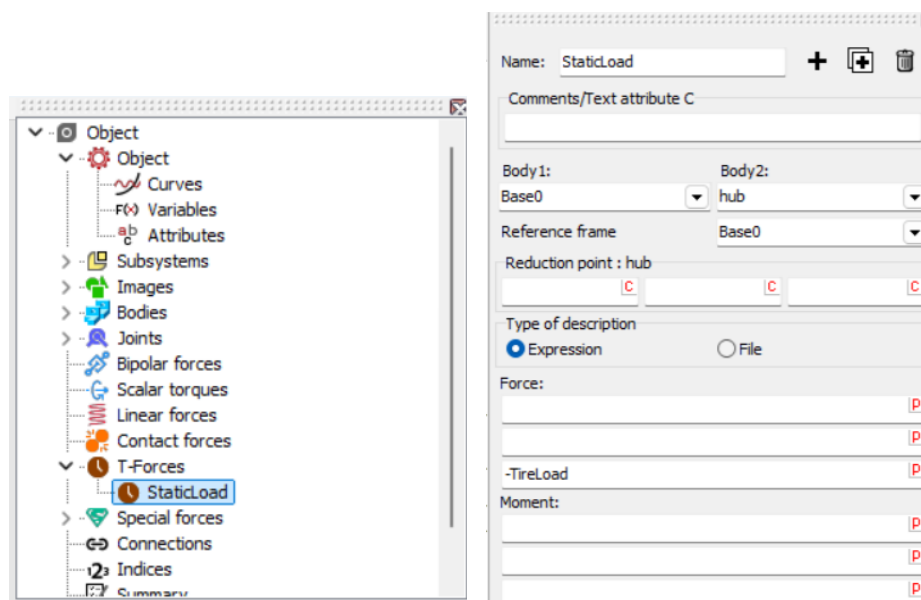


Figure 33.28. Setting wheel load

The vertical load on the wheel is set by the T-force **StaticLoad** and parameterized by the *tire-load* identifier. This vertical force is applied to the hub.

33.3.1.5. List of identifiers

The list contains all the identifiers listed above as well as some auxiliary identifiers that are used in the simulation program, including for parameterizing the tire material. Auxiliary identifiers will be discussed below, see Sect. 33.3.13 *Main tests for studying tire properties*

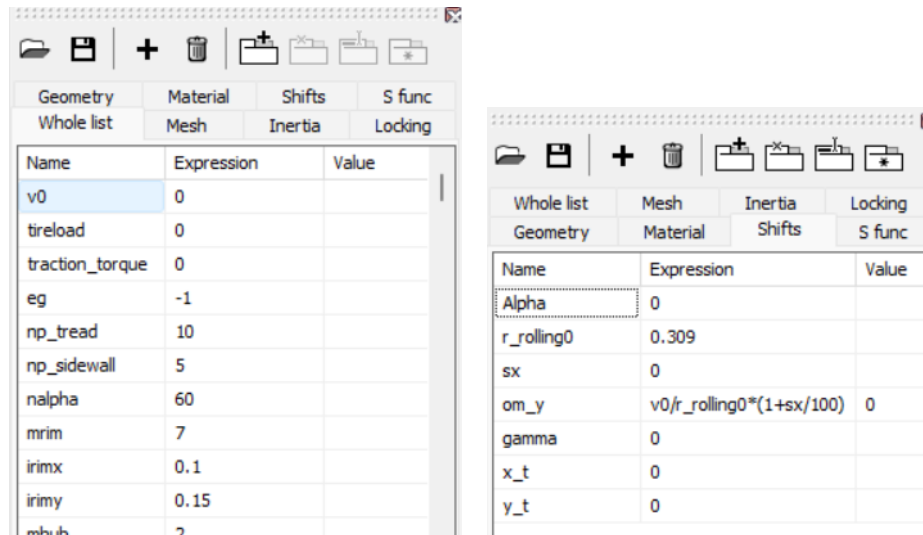


Figure 33.29. List of identifiers

For ease of use, some of identifiers are included in groups by type of use. For example, a group of identifiers is shown in Figure 33.29 on the right, which are responsible for the position and rotation of the hub, lateral slip (*alpha*) and longitudinal slip of the tire (*sx*).

Note. When creating a new tire model, it is recommended to rename the identifiers related to the geometric dimensions and inertial parameters of the tire (for example, *rwheel*, *mrim* and others) and the finite element meshing (*nalpha*). This will simplify the use of saved test configurations (Sect. 33.3.13.1.4 *Use of preliminary created configurations*), since when reading the parameters file, these identifiers will be changed to values for another tire, and this will lead to the need for additional adjustments.

33.3.1.6. List of variables

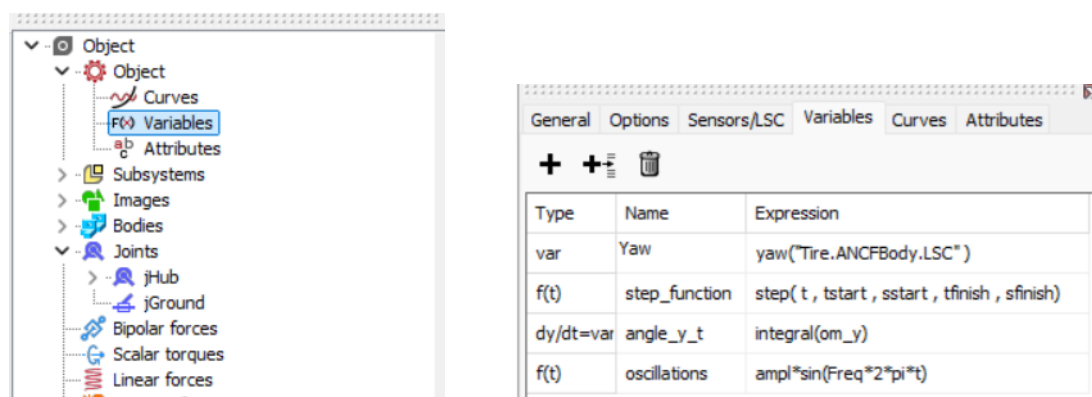


Figure 33.30. List of variables

The list includes the following variables.

tire_yaw – a kinematic function that takes during the simulation the value of the angle of rotation of the tire around the vertical axis, i.e. the angle of lateral slip in radians.

step_function – time function for a smooth transition from one numerical value to another $\text{step}(x, x_0, h_0, x_1, h_1)$, Figure 33.31; the function is parameterized by four identifiers placed on a separate tab **S func**. The function is used in simulation tests to manage identifiers, see Sect. 33.3.13 *Main tests for studying tire properties*.

angle_y_t – time integral of identifier om_y ; it is used to automatically calculate the tire rotation angle when specifying its angular speed, see Sect.33.3.13.7 *Dependence of longitudinal force on slippage*.

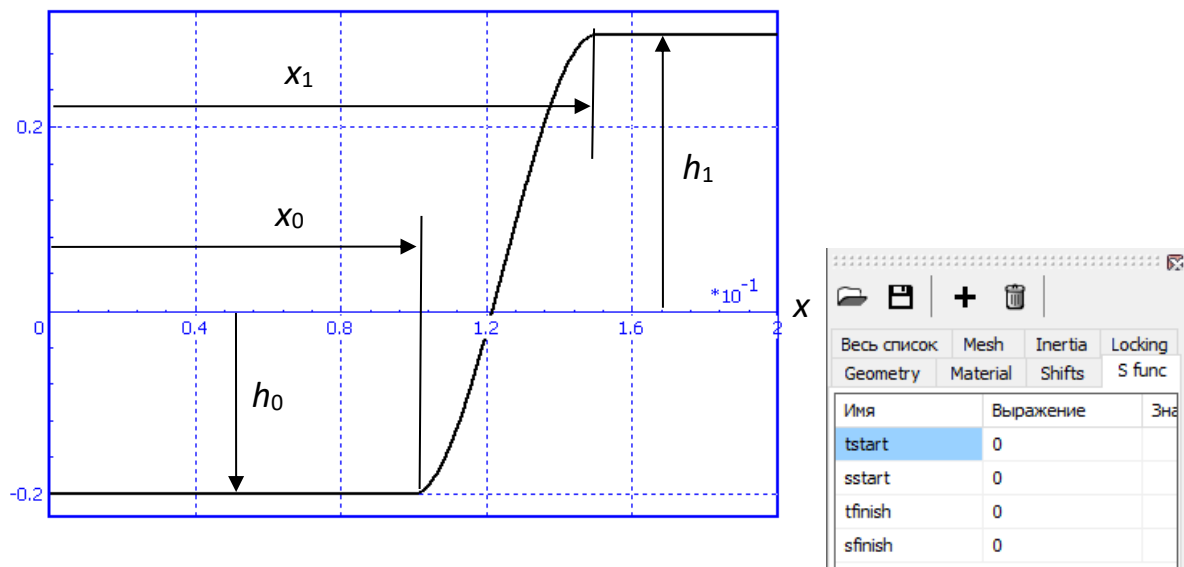


Figure 33.31. Step function and identifier tab

33.3.1.7. Development of wheel model for database

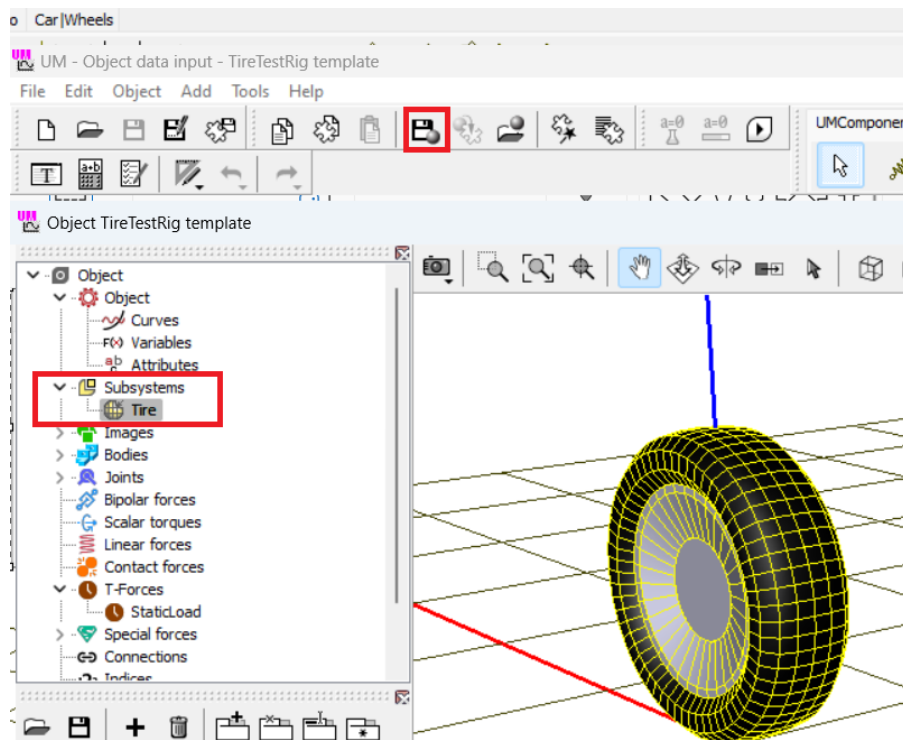



Figure 33.32. Creating a file with main elements of flexible tire

It is recommended to create a file containing the main elements of the tire for each of the developed FE tire model. This file is included in the database and allows you to simplify and speed up the process of adding flexible tires to a wheeled vehicle model, Sect. 33.4.3 *Adding FE tire to wheeled vehicle models*. Follow the steps below.

- Open a template of flexible tire in UM Input program.
- Go to the flexible wheel subsystem tab, Figure 33.32.
- Use the button  on the tool panel or the **Edit | Copy to file** menu command to save the subsystem in file [Name of subsystem].sbs in the tire model directory. The file contains the following elements:
 - subsystem with flexible tire;
 - graphic object for the wheel rim;
 - the force element **Hub coupling** which will be used for coupling of the wheel rim with the corresponding wheel of a wheeled vehicle, Sect. 33.4.3 *Adding FE tire to wheeled vehicle models*.
 - identifiers parameterizing the listed elements.

33.3.2. Structure of a tool for setting parameters and studying tire properties in UM Simulation program

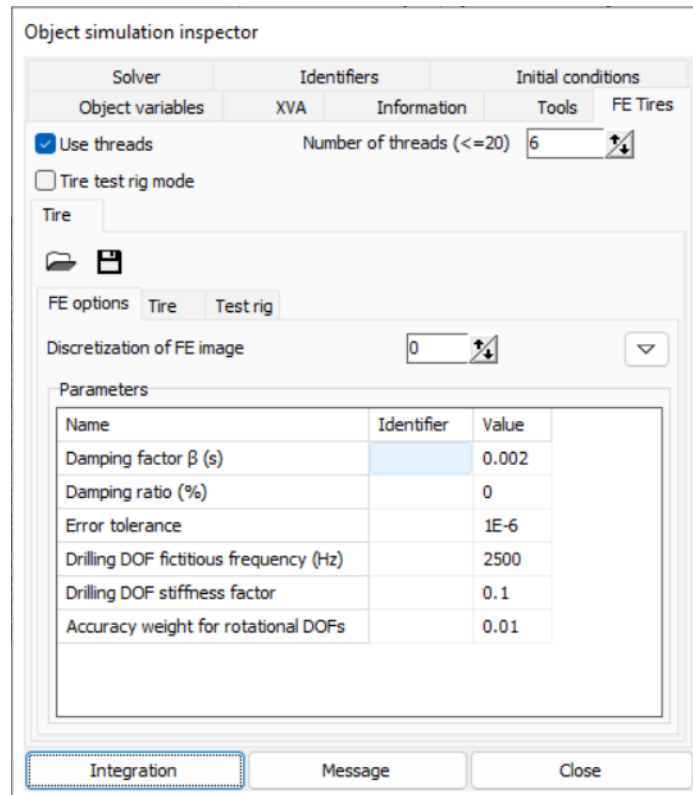


Figure 33.33. Tabs for development and study of flexible tire model

After creating a model template for a specific tire, the user should open it in the UM Simulation program. The main tools for setting parameters and studying the properties of the tire are located on the **FE Tires** tab of the Object simulation inspector, Figure 33.33. Let us look at the functions of tabs.

FE options. The tab includes parameters specifying the damping parameters, some parameters of the finite element model and solver.

Tire/Parameters. The page contains tabs for setting

- tire geometry
- material properties
- tire pressure
- tire/road contact parameters
- some additional parameters.

Tire/Models. Here a model from database can be assigned. If there are reduced models, the type is specified: full, upper or side reduction.

Tire/Reduction. Tool for development of full and reduced tire models and saving to files for tire database.

Test rig. The parameters of the virtual test rig are set here and the results of standard virtual tire tests are displayed: the dependence of the lateral, longitudinal forces and aligning moment on the corresponding slips, etc.

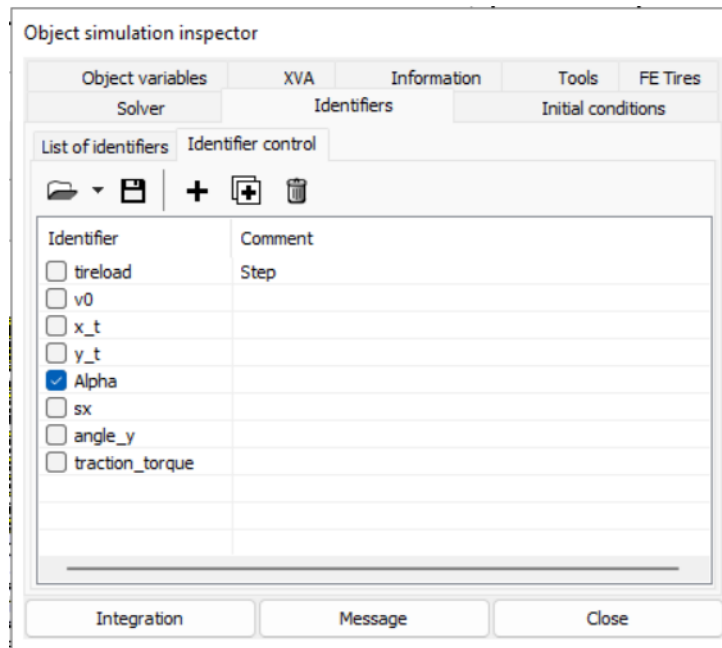


Figure 33.34. Tool for identifier control

The **Identifiers | Identifier control** tab is an important tool for enabling various tire testing modes. It allows setting dependencies on the time of lateral, longitudinal shift, lateral slip, load, speed, etc.

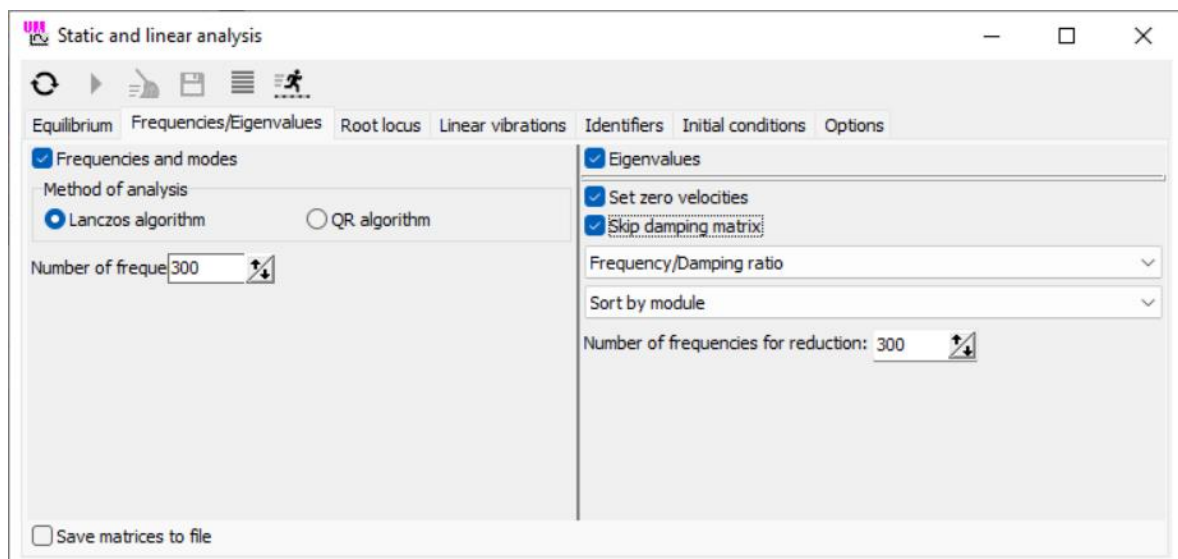
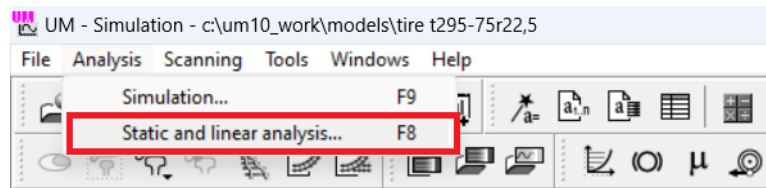


Figure 33.35. Computation of natural frequencies and eigenvalues

The calculation of tire frequencies and eigenvalues (to estimate damping) is carried out in the static and linear analysis tool, accessible through the main menu command **Analysis | Static and Linear Analysis**, Figure 33.35.

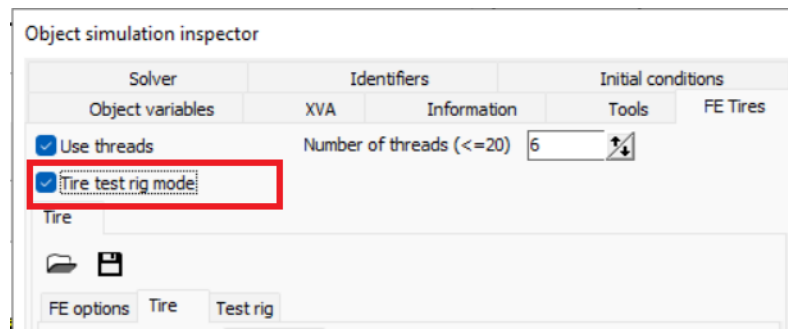


Figure 33.36. Enabling Test rig mode

Tire dynamic modeling is performed in two modes:

- mode of standard integration of equations of motion for given values of identifiers and other parameters;
- test rig mode, in which curves of the dependence of tire-road interaction forces on slip-page, static stiffnesses in different directions and other indicators are calculated. To enable this mode, use the corresponding key on the **FE Tires** tab, Figure 33.36. The main parameters of the mode are set on the **Test rig** tab.

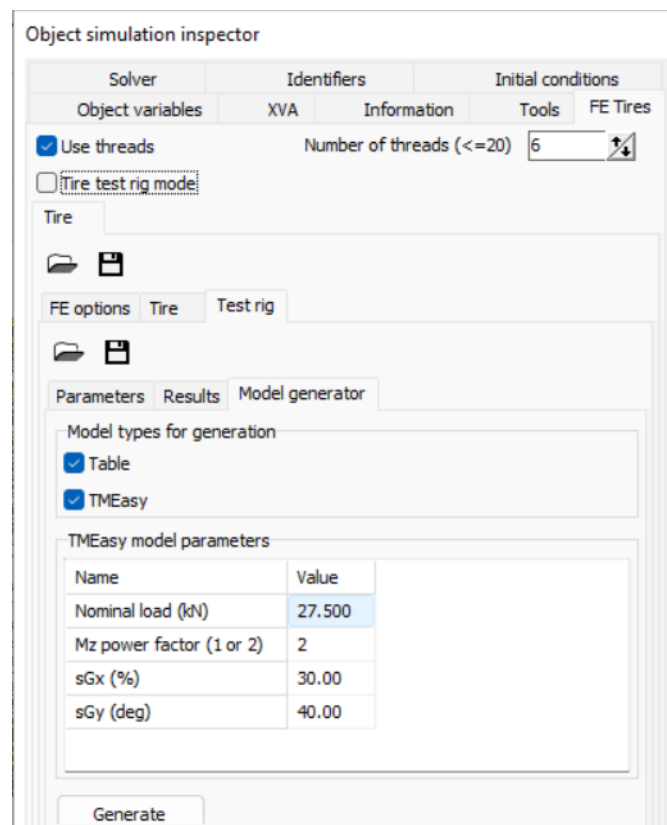


Figure 33.37. Generator of massless tire models

A generator of simplified massless tire models (tabular and TMEasy) becomes available after performing calculations in test rig mode.

33.3.3. Setting tire geometry and material

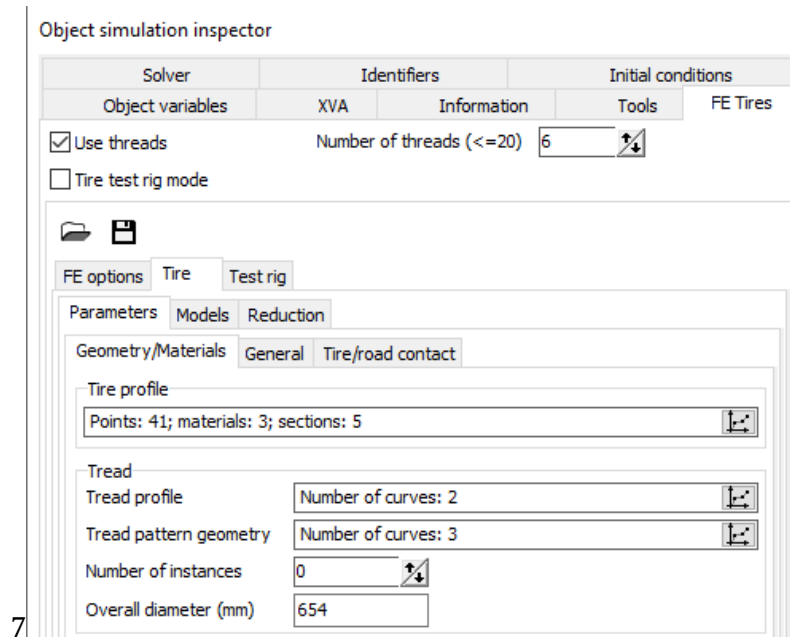


Figure 33.38. Tab for tire geometry and material

The tab **Tire | Parameters | Geometry/Materials** is used to set the tire geometry and material, Figure 33.38.

33.3.3.1. Tire profile and material

33.3.3.2. Setting tread geometry

Tread	
Tread profile	Number of curves: 2
Tread pattern geometry	Number of curves: 3
Number of instances	0
Overall diameter (mm)	654

Figure 33.39. Tread geometry description

The tread geometry is described in the **Tread** group, Figure 33.39.

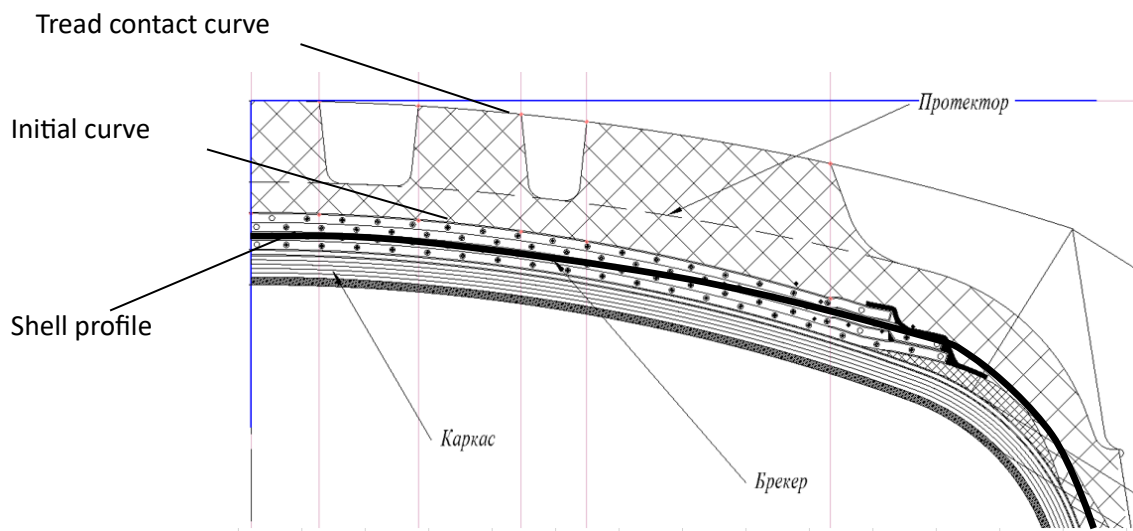


Figure 33.40. Boundary curves of tread material

Tread profile curves. Two boundary curves of tread material are specified, Figure 33.40:

- contact curve, which is the outer curve of the tread and defines the contact surface;
- initial curve, which is the boundary of the breaker.

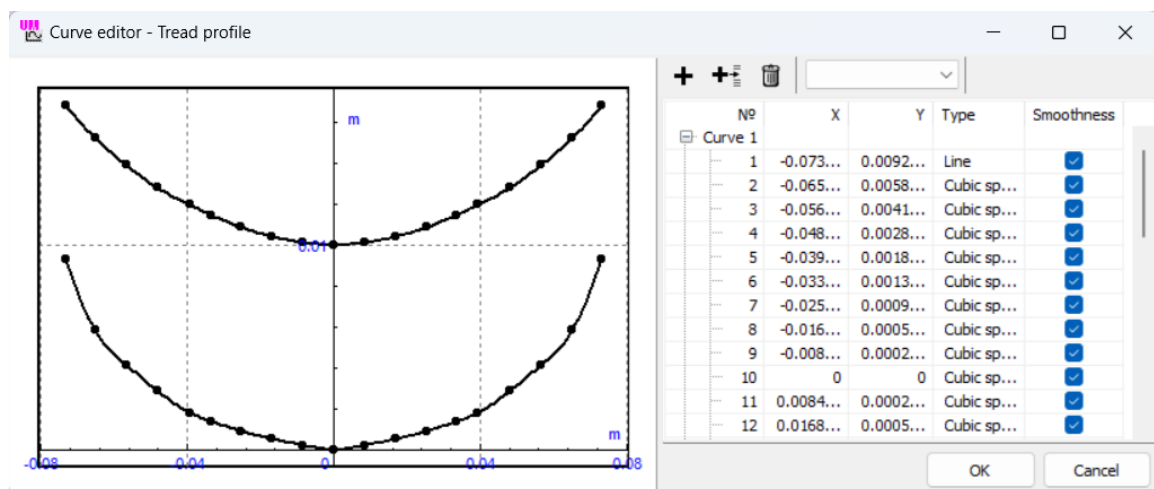



Figure 33.41. Setting contact and initial curves of tread

Curves are set using the curve editor by clicking on the button , Figure 33.41. Coordinates of points are specified in meters. Unlike Figure 33.40, the ordinate axis is directed upward from the contact curve, so that the initial curve is above the contact curve. The origin of system of coordinates for defining curves is located at the center of the contact curve.

The **Overall diameter** of tire specifies the position of central point of the contact curve relative to the wheel center. In fact, this parameter is the maximal diameter of the tread surface for tire with zero internal pressure.

Tread pattern.

Two styles have been developed to set the tread pattern, typical examples of which are shown in Figure 33.42.

The **first style** has a structure of surface revolution with grooves running along the entire contact surface of the tire, Figure 33.42 left. To define such a tread, the user should specify a set of segments on the abscissa axis with breaks corresponding to the grooves, Figure 33.43. The value of the **Number of instances** parameter in this case should be zero, as in Figure 33.47.

The **second style** assumes a periodic structure of the tread pattern along the surface of the tire, Figure 33.42 right. The period is specified by the **Number of instances** parameter, in the example of buses in Figure 33.43 this number is equal to 29. One period of drawings is created using the curve editor, Figure 33.44. To help in creating this type of pattern, a rectangle is automatically drawn in the window. The height of the rectangle is equal to the period taking into account the radius of the tire shell.

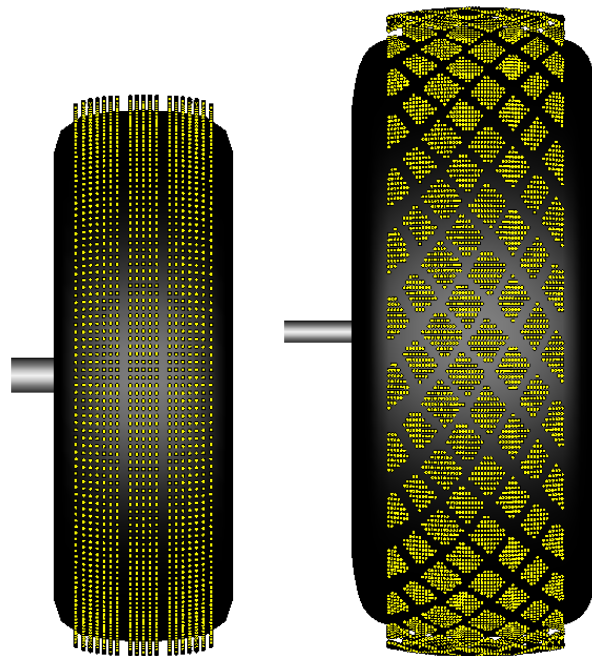


Figure 33.42. Examples of two tread pattern styles

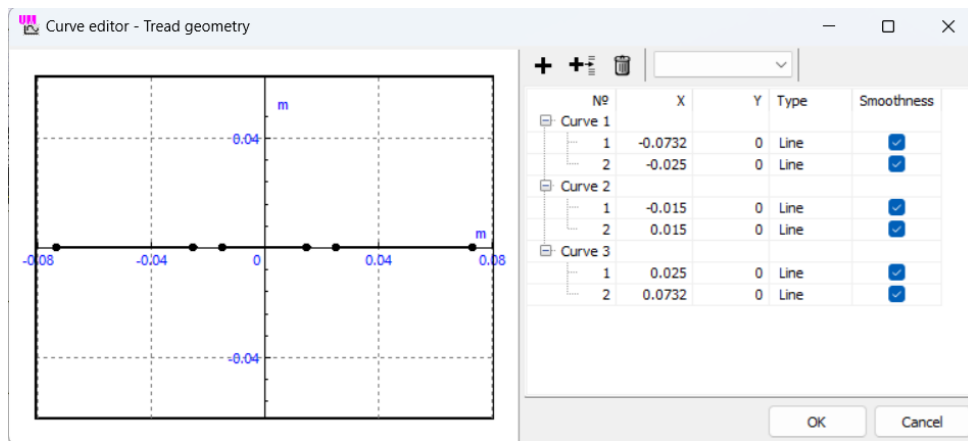


Figure 33.43. Setting the tread pattern geometry of the first type

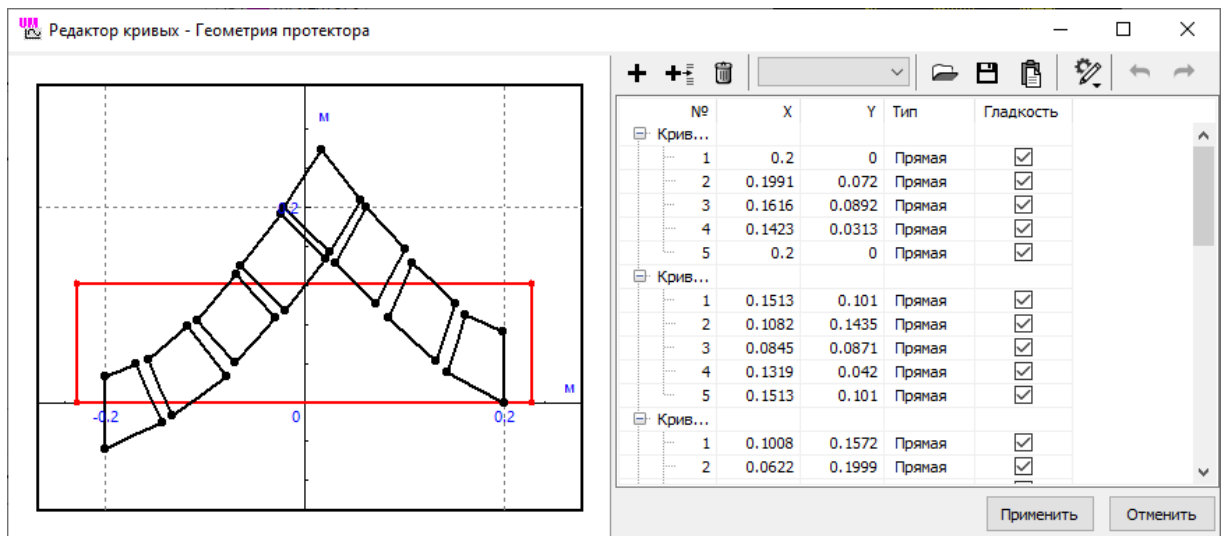


Figure 33.44. Setting the tread pattern geometry of the second type

33.3.4. Setting finite element parameters

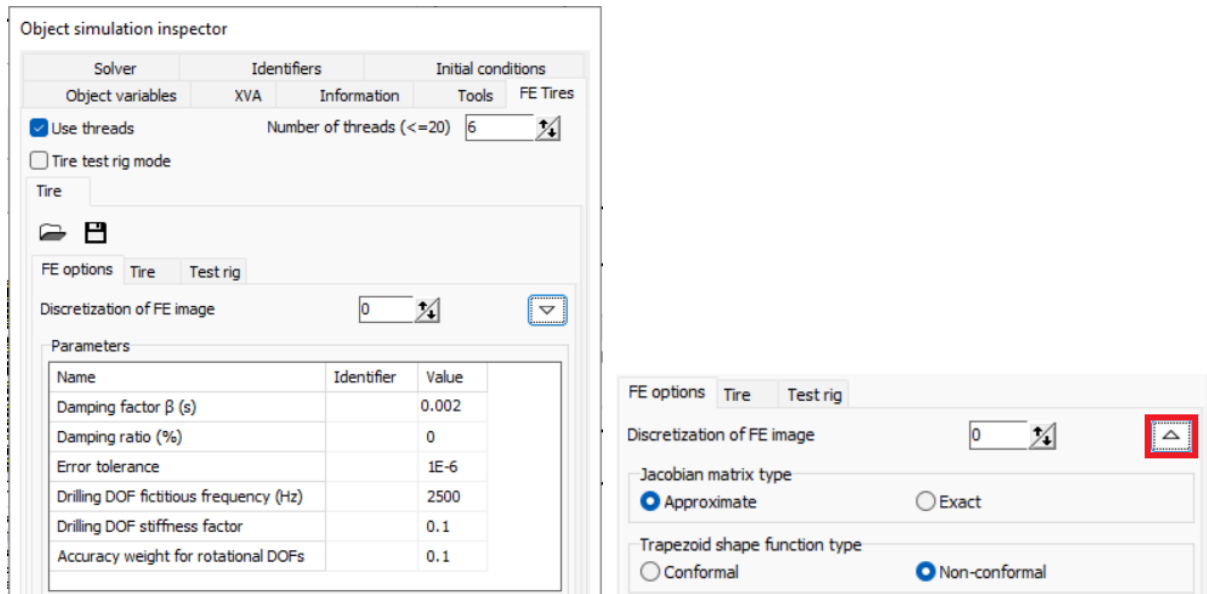



Figure 33.45. Parameters of tire FE

Some parameters of tire finite elements are set on the **FE options** tab, figure 33.45. Auxiliary parameters are hidden and accessible by clicking the button .

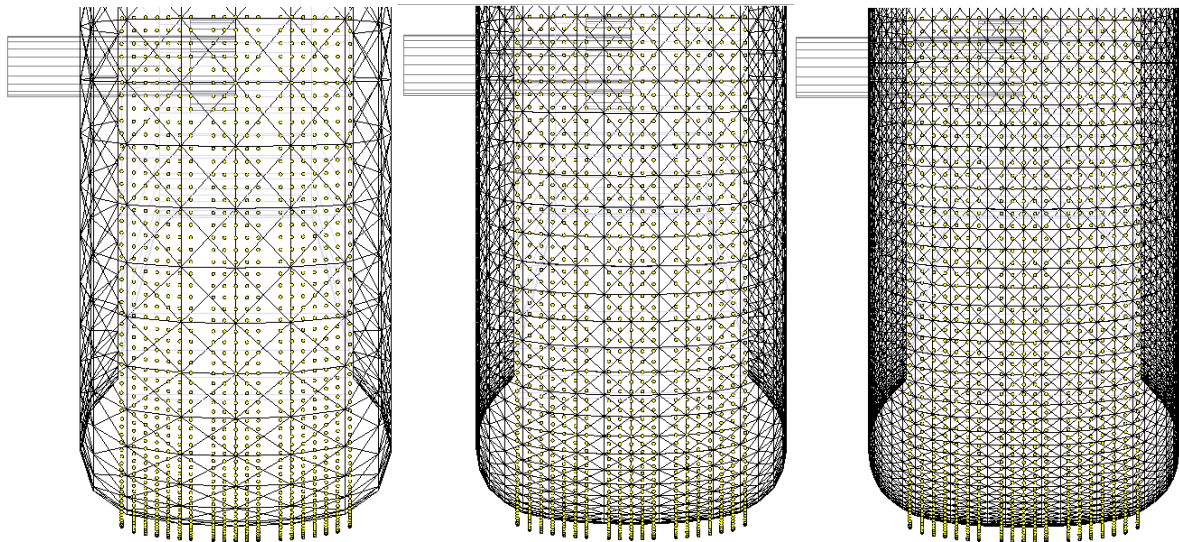


Figure 33.46. Discretization of FE image

The parameter **Discretization of FE image** affects only the tire image in the animation window. With a zero value of the parameter, the trapezoidal FE is drawn with two triangles, with a value of 1 - eight, 2 - eighteen, etc. The parameter does not affect the mechanical properties of the model.

The **Parameters** group contains numerical values of some FE models parameters, which can be parameterized by identifiers if necessary.

Damping factor β , s – the main parameter of the internal damping of the tire, proportional to the stiffness matrix, see Sect. 33.2.8 *Estimation of the influence of damping proportional to the stiffness matrix on eigenvalues*.

Damping ratio (%) – an auxiliary damping in equations (33.1). The δ parameter allows you to set the damping ratio within each of the elements. The disadvantage of such a damping is its dependence on the meshing as well as weak influence on high tire frequencies. For this reason, it is not recommended to use this parameter without special research for a specific tire model and mesh. This parameter has a noticeable effect on the properties of the tire at values of the order of 100% and above.

Error tolerance – specifies the accuracy of solving equations (33.2) in computation of the Craig-Bampton coordinates on the given absolute coordinates of nodes to within a finite element when constructing equations of motion (33.1). The equations are solved for each FE at each integration step. Increasing accuracy entails slowing down the simulation process.

Drilling DOF fictitious frequency (Hz), Drilling DOF stiffness factor – auxiliary parameters for the construction of a non-degenerate FE stiffness matrix. It is not recommended to change the default values.

Accuracy weight for rotational DOFs – reduces the influence of the angular degrees of freedom on the automatic control of the integration step size. With a value less than 1, it allows speeding up the simulation process due to an acceptable loss of accuracy. Recommended values are in the range [0.01, 1]. When specifying reduced accuracy, a comparison of the simulation results with the value 1 is required.

Two additional options hidden by default:

Jacobian matrix type – specifies the method for solving Eq. (33.2).

Trapezoid shape function type – sets the choice of shape functions for FE.

It is recommended to use the default values for these parameters.

33.3.5. Setting tire/road contact parameters

Name	Identifier	Value
Tread Young's modulus		3160000
Contact damping coefficient		0.1
Friction coefficient f0		0.8
Friction coefficient f		0.55
Stribeck velocity threshold		3
Friction power parameter		1
Discretization X contact per element		5
Discretization Y contact per element		5

Figure 33.47. Tread/road contact parameters

Tread and contact parameters are available on the tab **Tire | Parameters | Tire/road contact**, Figure 33.47. Here the parameters of the brush model of tire contact with the road are set, see Sect. 33.2.5 *Contact interaction of tire with road*.

Tread Young's modulus – parameter of the material from which the protector is made, Pa;

Contact damping coefficient – coefficient of linear damping for each of the bristle $d_n = d_\tau$ in the contact force models (33.5), (33.6), Ns/m;

The following four parameters define the Stribeck model for the dependence of the friction coefficient on sliding velocity (33.7)

$$f(v_s) = f_\infty + (f_0 - f_\infty)e^{-(v_s/v_{str})^\delta}$$

Friction coefficient f0 – static coefficient of friction f_0 ;

Friction coefficient f – coefficient of friction for large sliding velocity f_∞ ;

Stribeck velocity threshold v_{str} ;

Friction power parameter δ .

Two parameters determine the number of rows of bristles in the longitudinal and transverse directions for one FE which can be in contact with the road,

Discretization X contact per element (n_x)

Discretization Y contact per element (n_y)

Thus, the maximum number of bristles for a FE is $(n_x + 1)(n_y + 1)$ however the actual number may be less due to grooves and channels in the tread geometry.

33.3.6. Setting numerical value of tire pressure

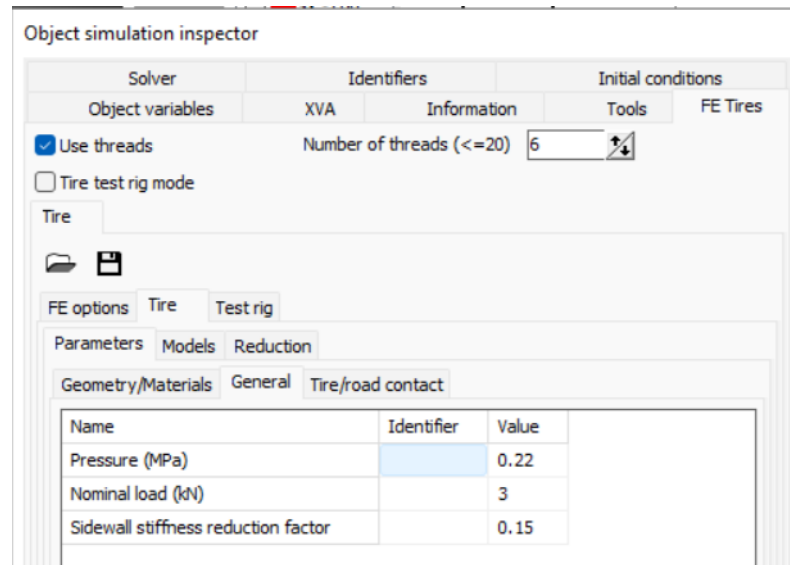


Figure 33.48. Tire pressure parameter

The tire inflation pressure is set on the tab **Tire | Parameters | General**. The numerical value of pressure is given in megapascals.

33.3.7. Setting stiffness reduction factor for tire with reduced side-walls

The parameters for the stiffness reduction of a tire with a reduced sidewall are set on the tab in Figure 33.48: the nominal load and the reduction factor, Sect. 33.2.7 *Reduced tire models*. Computation of the reduction factor is described in Sect. 33.3.13.4.4 *Evaluation of the parameters of a modified tire model with a reduced sidewall*. The non-modified reduced model is used for zero value of the factor.

33.3.8. Settings for solver when modeling wheels with FE tire

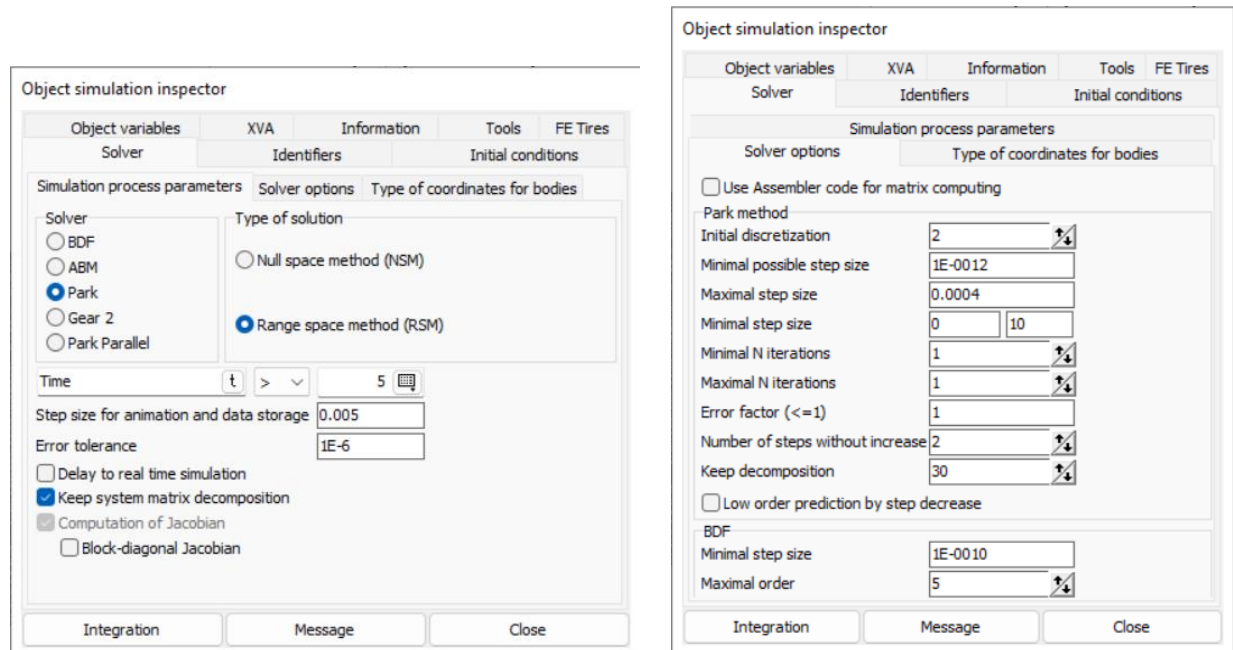


Figure 33.49. Solver settings

Typical settings for the solver are presented in Figure 33.49 left. The following checked option is important:

Keep system matrix decomposition – the option significantly (up to ten times) speeds up the simulation process by keeping the system matrix and its decomposition at several integration steps. The related parameter **Keep decomposition** (Figure 33.49, right) specifies the maximum number of integration steps on which the matrix decomposition is kept. A large value (about 100) is recommended for applications with an unmoved tire. When running, the matrix should be recalculated more often, keeping for 10-20 steps.

Approximately double speed up of the simulation process can be achieved by parallelizing calculations on multi-core processors, Figure 33.50. Enabling the parallel calculation mode and setting the number of involved cores is done on the **FE tire**. Check the **Use threads** option and set the number of threads. Practice shows that it is most effective to use 4-6 threads depending on processor type.

Also, approximately one and a half to two times the speed up is achieved by reducing the **Accuracy weight for rotational DOFs** from 1 to 0.1, Figure 33.50. The limit value of this parameter is 0.01. In any case, the effect of decreasing this parameter on the simulation results should be tested by comparison of simulation results with different values of this parameter.

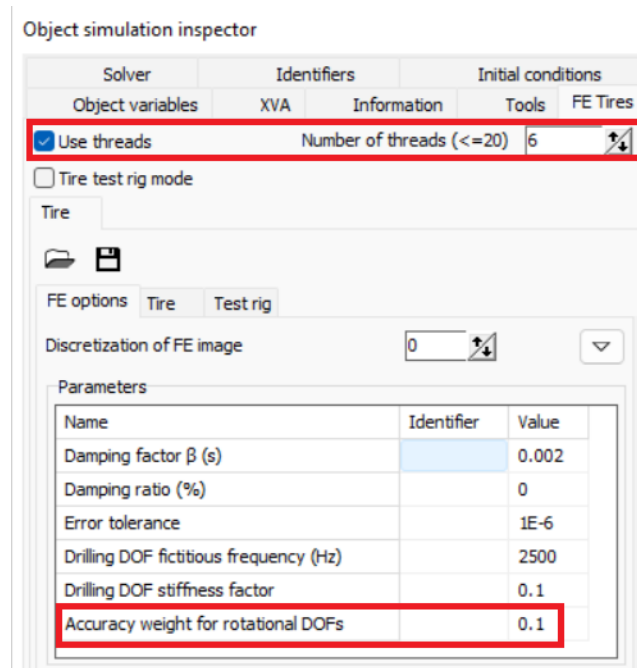


Figure 33.50. Settings that speed up the simulation process

Note. In the current UM version, the Park parallel method is not available for simulation of FE tires. In the future, this restriction will be removed.

33.3.9. Animation of tire-road contact forces

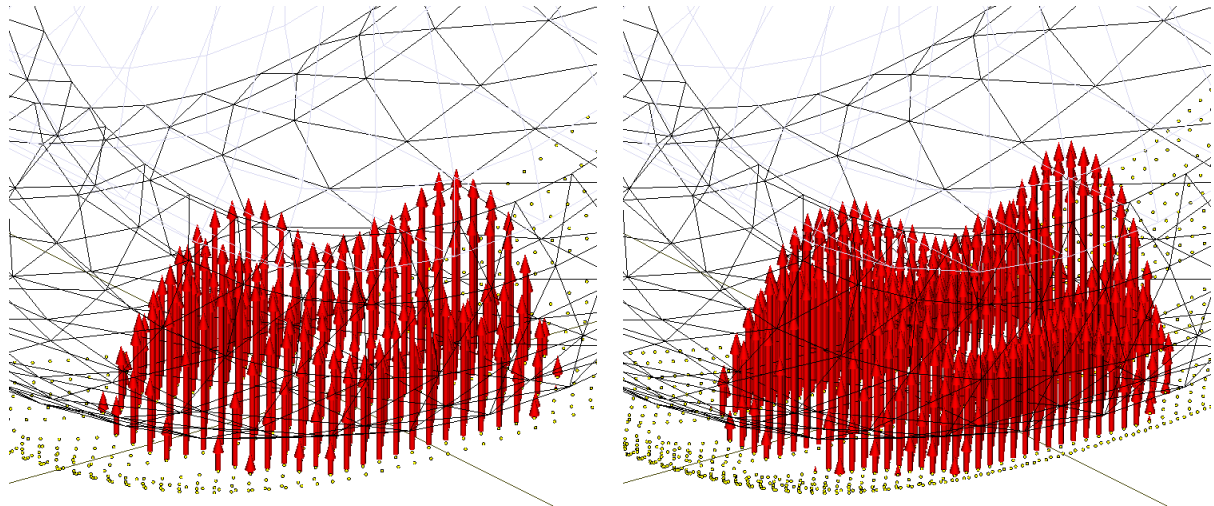


Figure 33.51. Vectors of normal forces (actually pressures) for different numbers of bristles within the element

When modeling the dynamics of an FE tire, the vectors of normal forces and friction forces can be drawn in the animation window. To ensure that the height of the force vectors does not change when the number of bristles within one FE changes (Figure 33.51), pressure vectors are actually drawn, i.e. forces are divided by the area per bristle: the area of the FE is divided by the theoretical number of bristles within the FE including those excluded from contact due to grooves and contact boundaries.

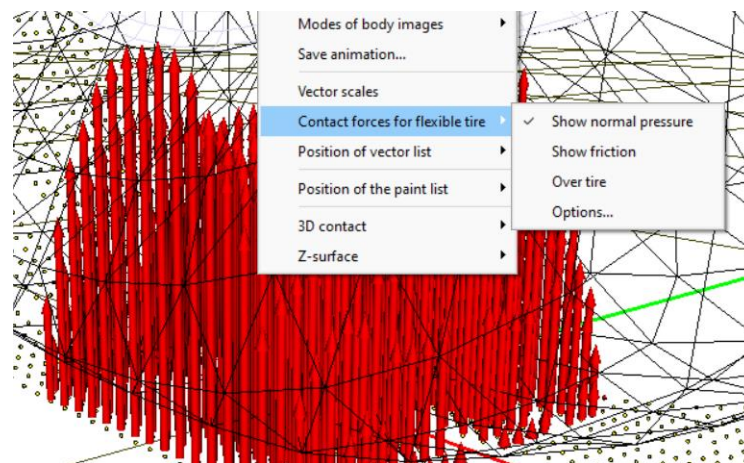


Figure 33.52. Fragment of an animation window with a popup menu

Let us look at the contact force animation settings available through the popup menu of the animation window, Figure 33.52. Here is the list of the menu commands.

Show normal pressure – turns on/off the animation of normal forces in contact.

Show friction – turns on/off the animation of friction forces in contact.

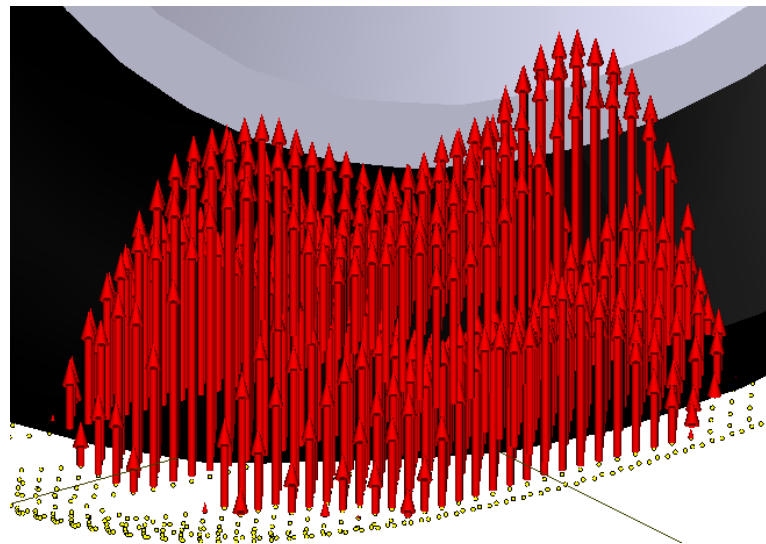


Figure 33.53. Enabled "over tire" mode in solid graphics mode

Over tire – a mode in which force vectors are shown on top of the graphic image of the tire. This mode is recommended when the tire is drawn in solid graphics, Figure 33.53, otherwise the vectors will be hidden by the shell and you have to convert the graphic image of the tire into wireframe graphics, as in Figure 33.52.

Options – the command opens the window with options, Figure 33.54.

On the **Scale** tab, the pressure value at which the vector will have a length equal to the characteristic size (usually 1 m) is indicated. On the **Size** tab, the user can change the radii of the vector image and the length of the arrow in mm. **Visibility, colors** – turn animation on and off, change the style and colors of vectors. The entered parameters are written to the configuration file and applied in all animation windows when the model is read again.

Figure 33.56 shows three variants of animation:

- normal and friction forces are visible,
- normal forces only,
- friction forces only.

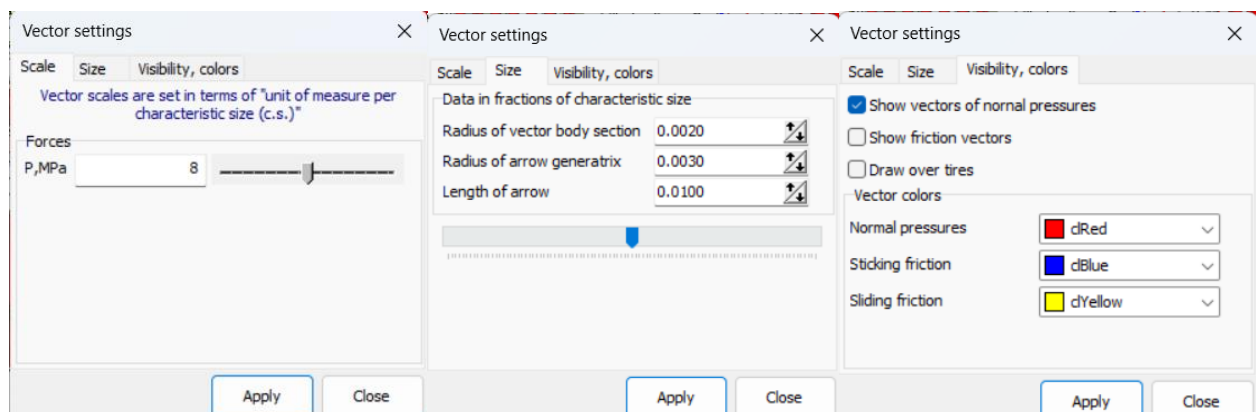


Figure 33.54. Options for force animation

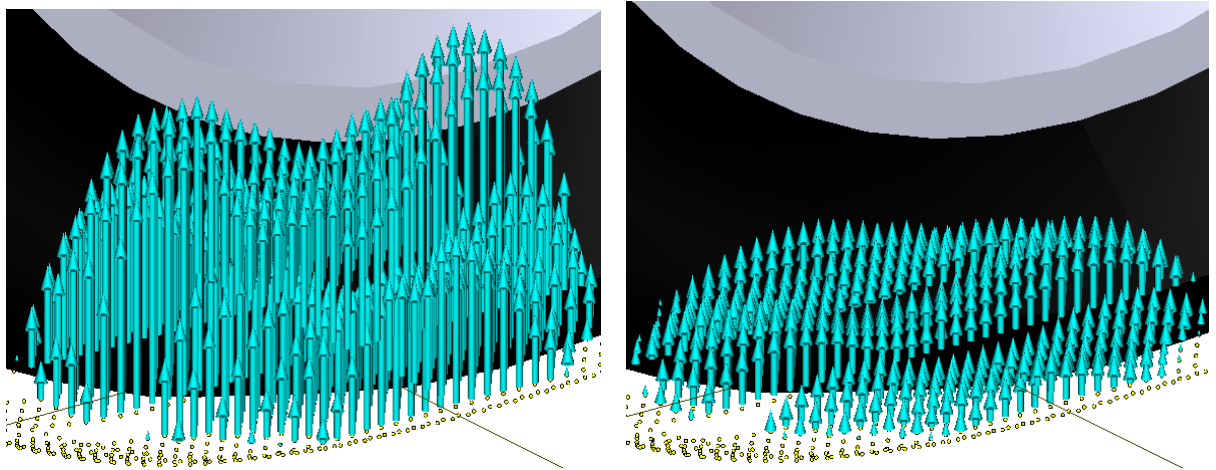


Figure 33.55. Change of scale and color

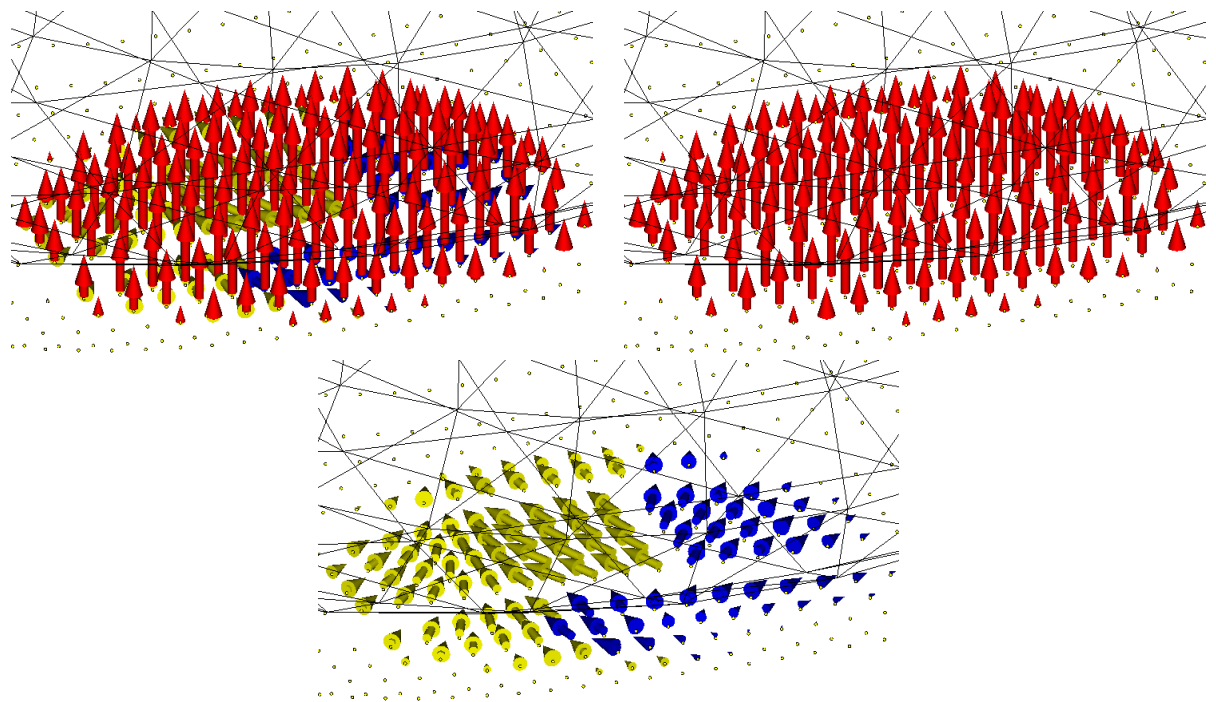


Figure 33.56. Force animation options

33.3.10. Wizard of variable: FE tire nodal variables

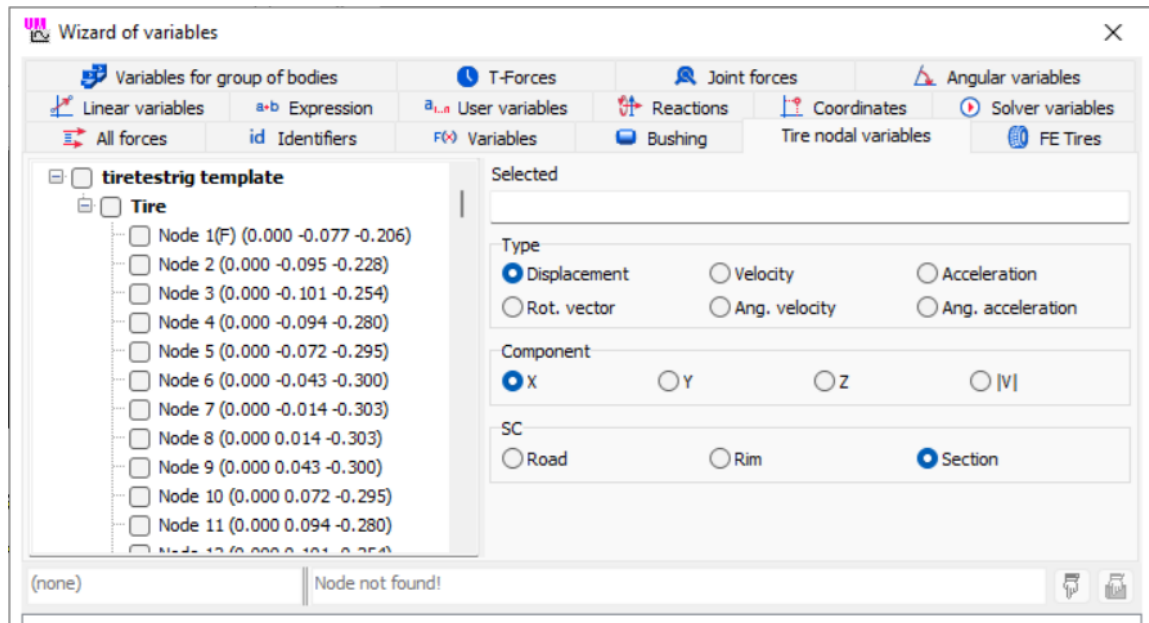


Figure 33.57. Tire nodal variables

Kinematic nodal variables for FE tire are created on the **Tire nodal variables** tab of the wizard of variables, Figure 33.57.

The variables refer to the elastic displacements of the nodes relative to the wheel rim. The vector of elastic displacement, the velocity and acceleration of the node relative to the disk, as well as the rotation vector, angular velocity and angular acceleration of the SC node relative to the wheel disk (**Type** group) are calculated. The variable is equal to the projection of the vector onto the axis of one of the three coordinate systems (**SC** group) or the modulus of the vector. The projection or magnitude of the vector is specified in the **Component** group.

The following systems of coordinates are used:

Road – the inertial system of coordinates SC0;

Rim – SC associated with the disk rim;

Section – CS of the section to which the node belongs. SCs of this type rotate together with the wheel rim and form a constant angle about the transverse axis Y with the rim SC. In particular, for the first section, the SC of the section coincides with the SC of the wheel rim. The Z axis of the section passes through the center of the wheel.

Note. The nodes rigidly connected to the wheel rim are marked with the letter F in the list in Figure 33.57. These nodes are fixed relative to the wheel rim, and all variables for them are equal to zero.

33.3.11. Wizard of variable: FE tire road contact variables

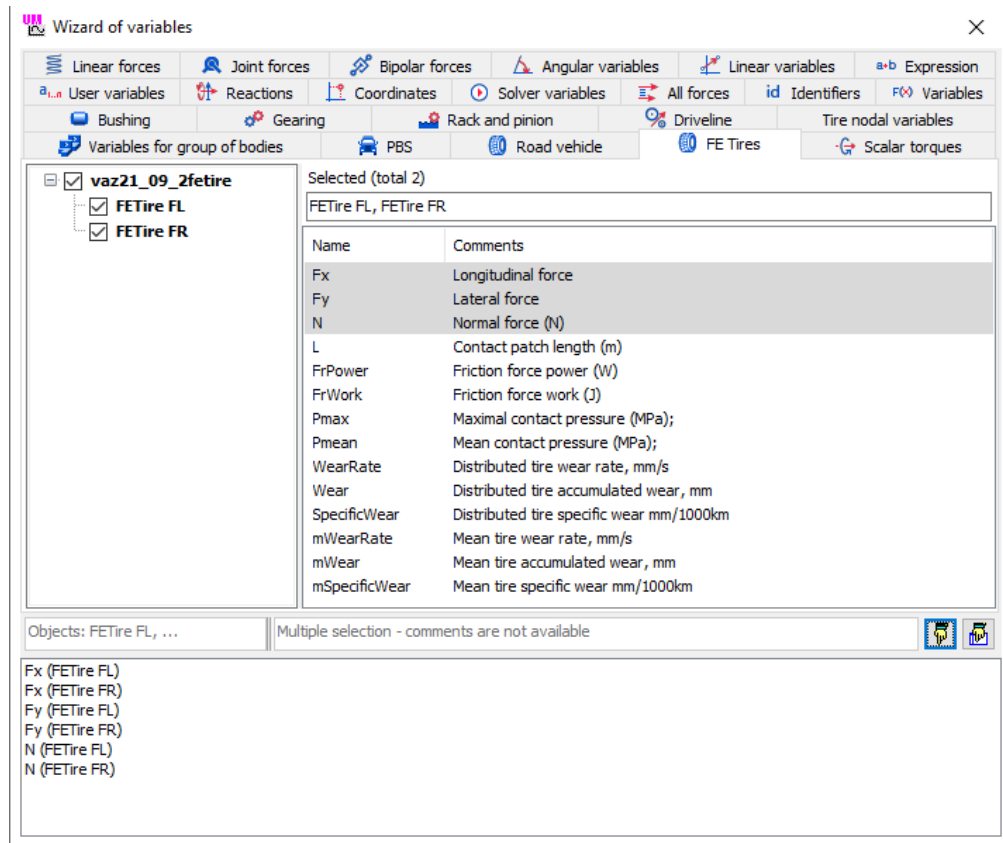


Figure 33.58. Variables for FE tire

Variables associated with the FE tire are placed on a separate tab of the **Wizard of variables**, Figure 33.58.

F_x – the total longitudinal force acting from the road on the tire (N).

F_y – the total lateral force acting from the road on the tire (N).

N – the total normal force acting from the road on the tire (N).

L – length of the contact patch, mm.

FrPower – the total power of friction forces (watts), only the bristles in which slippage occurs in contact with the road are taken into account. In sticking mode, friction force does no work.

FrWork – total work of friction forces since the start of simulation (joules); calculated as the time integral of the variable **FrPower**.

P_{max}, **P_{mean}** – the maximum and average normal tire pressure on the road. The pressure is equal to the force divided by the area associated with a bristle.

WearRate – wear rate curve of the tire material distributed along section of the tire rolling surface, mm/s.

Wear – accumulated tire wear curve, mm.

SpecificWear – curve for the tire wear rate related to the 1000km mileage, mm/1000km.

mWearRate, **mWear**, **mSpecificWear** – mean values of the corresponding distributed variables.

33.3.12. Creation and assignment of files for full and reduced FE tire models

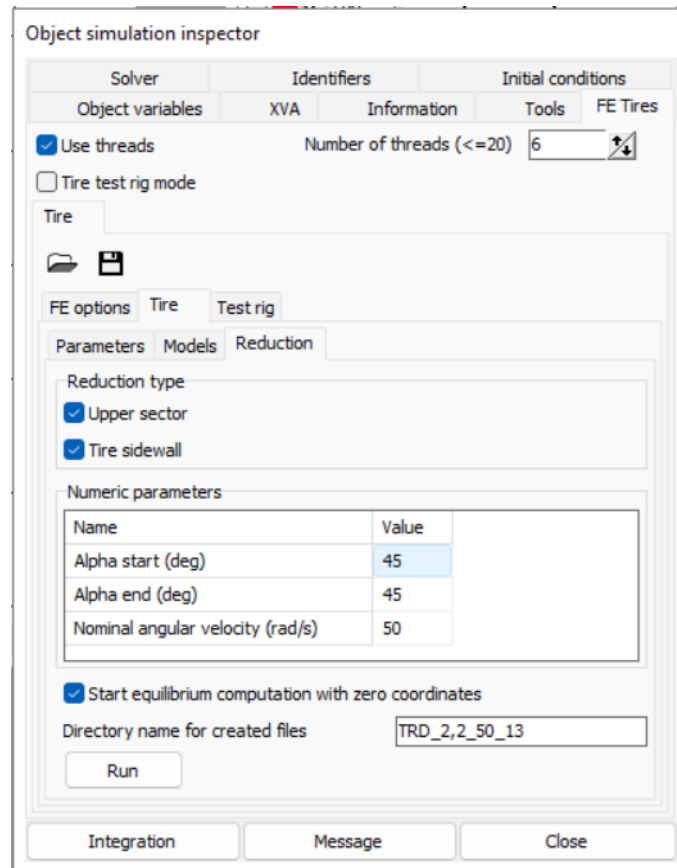


Figure 33.59. Tab for development of full and reduced FE tire models

To use FE tire models in studying the dynamics of wheeled road vehicles and monorail trains, the user should create a file of a complete description of the tire model *.tire and, optionally, files of reduced models *.tsr, *.tur, Sect. 33.2.7 *Reduced tire models*. For this purpose, the **Tire | Reduction** tab of the Simulation inspector is used, Figure 33.59.

The file extensions of the reduced models are explained as follows:

*.tsr – tire side reduction;

*.tur – tire upper reduction.

Consider the workflow for development of models.

1) Specify the values of the parameters on which the model depends:

- Inflation pressure, Sect. 33.3.6 *Setting numerical value of tire pressure*
- Material parameters, Sect. 33.3.3 *Setting tire geometry and material*
- FE mesh parameters.

Please note that material damping parameters are not taken into account when constructing reduced tire models and can be changed when studying vehicle dynamics.

2) If necessary, change the name of the directory where the files are written. By default, the directory name includes (Figure 33.59):

- abbreviation TRD – tire reduction data;
- pressure in bar (2.2 bar in Figure 33.59);

- number of finite elements along the tire circumference (50 in Figure 33.59);
- number of finite elements in the tire section (13 in Figure 33.59).

The created files are placed in this directory and have the same name, for example, for the TRD_2,2_50_13 directory the file names are:

TRD_2,2_50_13.tire, TRD_2,2_50_13.tsr, TRD_2,2_50_13.tur.

3) Set the reduction parameters

- **Reduction type** – the reduced models being created are indicated; if no reduction type is specified, then only *.tire for the full tire model file is created;
- Option **Start equilibrium computation with zero coordinates**

When calculating reduced models, the tire must be in a state of equilibrium at a given inflation pressure in the absence of contact interaction with the road. If the option to use zero coordinates is enabled, the program determines the equilibrium state automatically by solving nonlinear equilibrium equations. If the program cannot solve the equations, then the user should turn off this mode and calculate the equilibrium state of the tire by integrating the equations of motion: see Sect. 33.3.13.2.1 *Computation of equilibrium for linear analysis*, subsection **Calculation of equilibrium by the integration method**.

- **Numeric parameters**

If necessary, change the numerical parameters:

- initial and final reduction angles α_{start} , α_{end} used when reducing the upper part of the tire, Sect. 33.2.7 *Reduced tire models*;
- nominal angular velocity – angular velocity of the tire at the rated speed of the wheeled vehicle; the parameter is used in reduced models to construct the dependence of the models on the inertia forces associated with tire rotation.

- 4) Start the calculation process. After its successful completion, check the presence of the created files. A directory with the given name is created in the directory of the current object. When starting the calculation, an existing directory with the same model name is automatically deleted by the program.

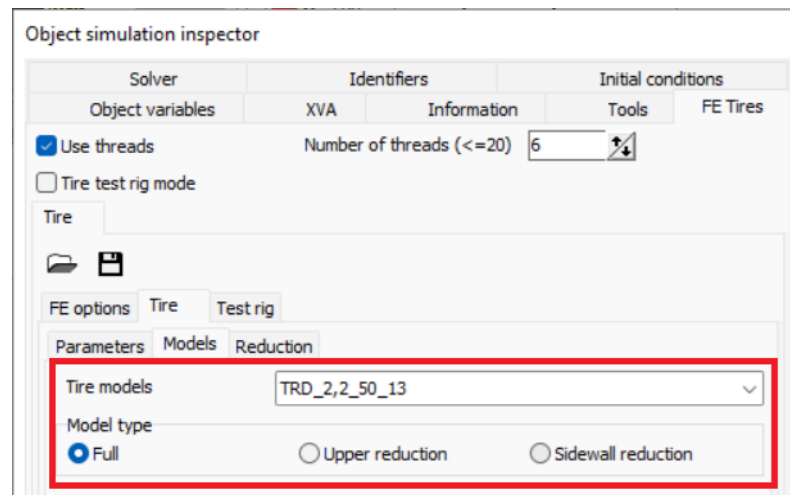


Figure 33.60. Setting FE tire model

The created tire models can be assigned to the current tire on the **Tire | Models** tab to run tests, Figure 33.60. The *.tire description file is loaded using the **Tire models** list, which contains all detected models in the current object directory. After loading the model, the user can set the reduction type if there are corresponding previously created files.

33.3.13. Main tests for studying tire properties

33.3.13.1. Auxiliary tools

33.3.13.1.1. Macros for identifiers

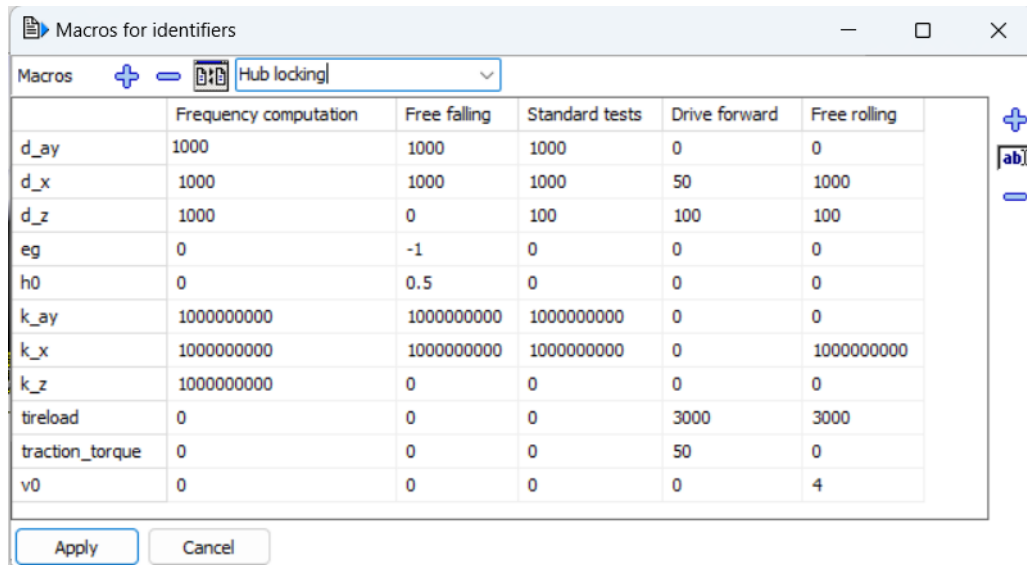



Figure 33.61. Tool for creating and modification of macros for identifiers

Use the **Macros for identifiers** tool to change quickly the values of a group of identifiers, Figure 33.61. The tool is available via the menu command **Tools | Identifier macros**. Assigning identifiers using a macro occurs on the **Identifiers** tab by clicking on the button , Figure 33.62. After selecting a macro, a confirmation window appears in which the user can exclude the assignment of some identifiers or cancel it completely. For example, the *tireload* identifier in Figure 33.62 is unassigned, and the current value of 2000 will be left.

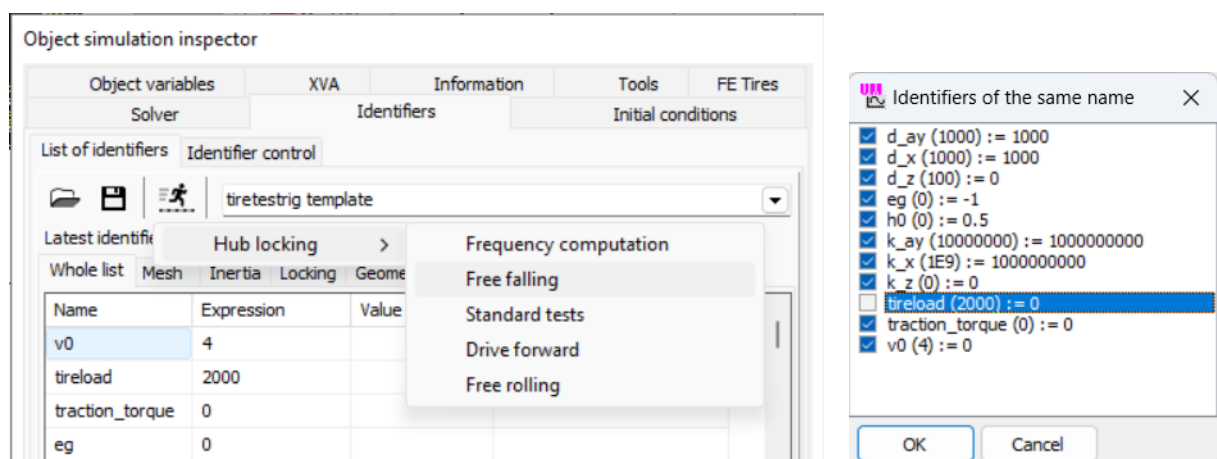


Figure 33.62. Example of assigning identifiers using a macro

Macros in Figure 33.61 implement various methods of fixing the hub. Consider the macros with explanations.

Frequency computation – all degrees of freedom of the hub are locked by forces. This case is used in frequency analysis, Sect. 33.3.13.2 *Computation of frequencies and eigenvalues*.

Free falling – only the vertical degree of freedom is unlocked. It is used in test 33.3.13.3 *Estimation of internal damping by coefficient of restitution of bouncing tire*.

Standard test – the vertical degree of freedom is unlocked, but the vertical damping is left. The macro is used in most static and tests described below.

Drive forward – three degrees of freedom of the hub are freed: vertical, longitudinal and rotation around the transverse axis. The macro is used in the test described in Sect. 33.3.13.5 *Wheel rolling forward under constant torque*. Damping is left in the longitudinal direction to limit the speed of longitudinal movement.

Free rolling – two degrees of freedom of the hub are freed: vertical and rotation around the transverse axis. A non-zero load and velocity of the Ground body are specified. This macro is used in test Sect. 33.3.13.6 *Evaluation of effective tire radius for freely rolling wheel*.

33.3.13.1.2. List of variables for tests

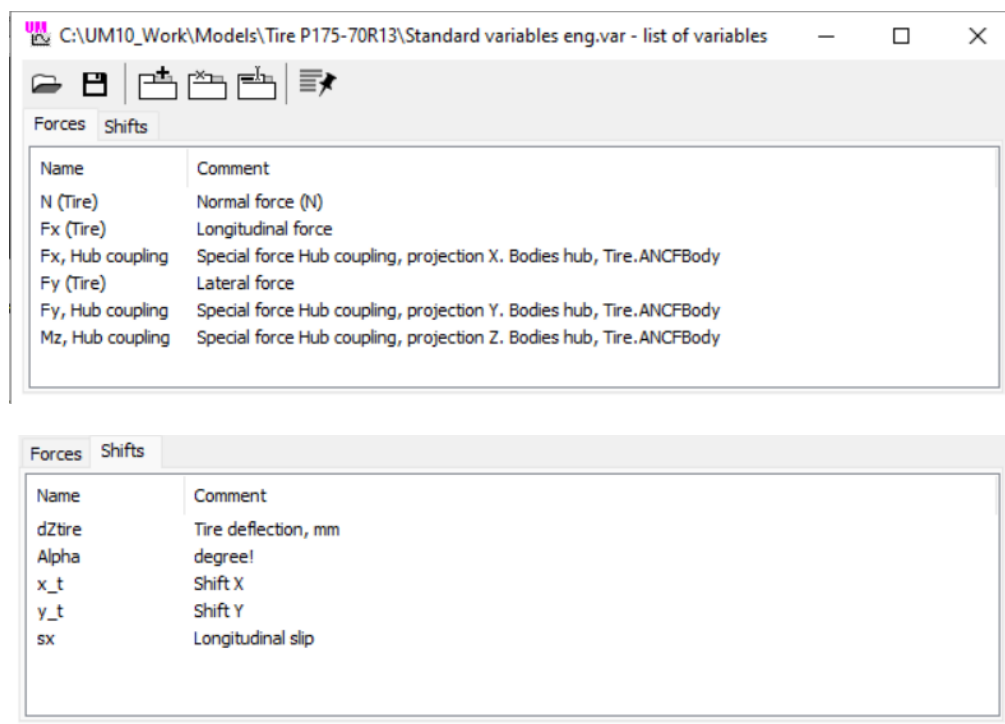



Figure 33.63. List of standard variables

The *Standard variables.var* file is opened using the menu command **Tools | List of variables** or by the button . The list contains several variables that are useful when constructing graphs in simulation tests with the wheel model, Figure 33.63.

Let us list the tests that use these variables.

N (Tire), dZTire – tire pressure on the road and tire deflection

The variables are used to plot the dependence of the static deflection of the tire on the load, , Figure 33.83, Sect. 33.3.13.4.2 *Dependence of tire deflection on load*. Let us focus on the **dZTire**

variable equal to the tire deflection. Open this variable in the wizard using the corresponding pop-up menu command by right-clicking on the variable, Figure 33.64. The variable is of type **Expression** and is formed by two operators. The first multiplies the joint coordinate X1.2 of the hub by -1000 changing the sign and converting to mm. The second operator determines the actual tire deflection by subtracting the absolute value of the coordinate at the moment of touching the road converted to mm 4.13. This last number (4.13mm) was obtained in the wheel drop test, where we simultaneously plotted the coordinate X1.2 and the normal force N. At the moment when the force appeared, the value of the coordinate was equal to -0.00413m. If the user analyzes his own wheel model, he must modify this expression by replacing the value 4.13 with the value corresponding to the case being studied.

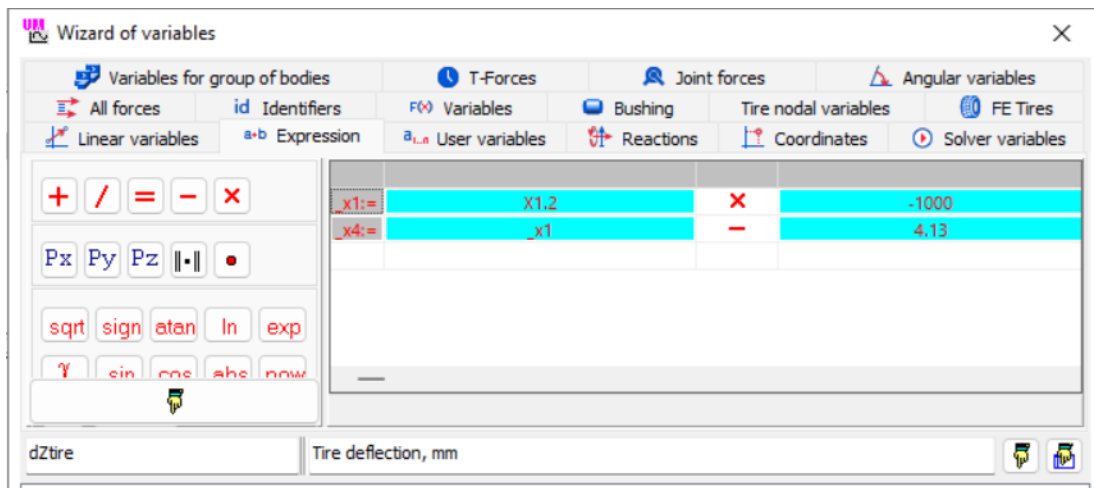


Figure 33.64. Tire deflection variable

33.3.13.1.3. Identifier control tool

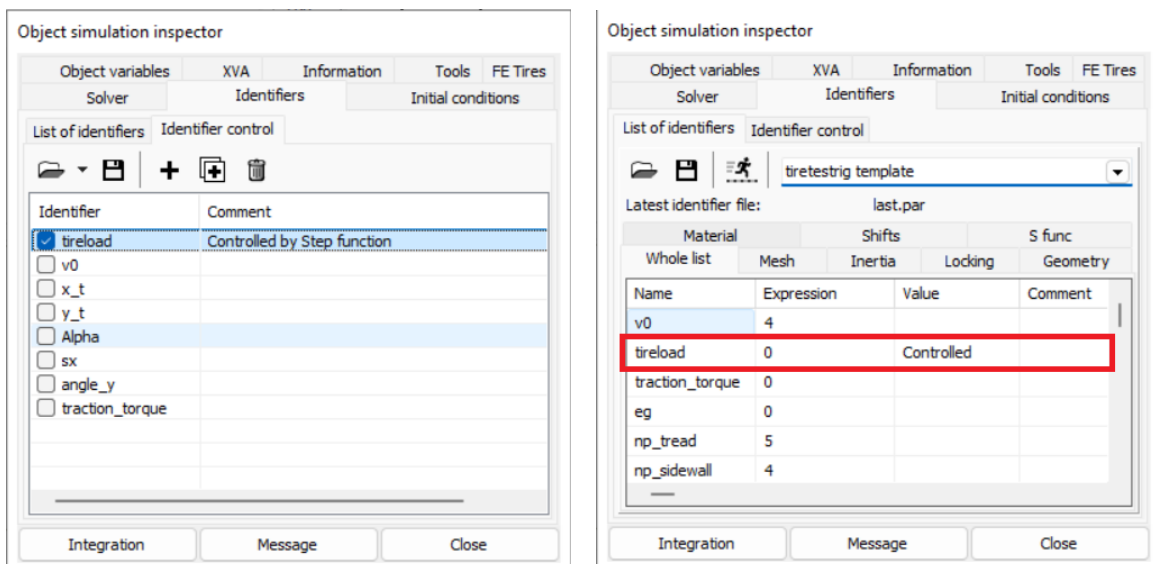


Figure 33.65. Identifier control

The identifier control provides the main tool for performing tests of various types with a wheel, Figure 33.65. The tool allows setting a dependency for one or more identifiers during the

simulation process. The dependency can be a specified time function or a variable created using the Wizard of variables. When a control is active, as for the *tireload* identifier in Figure 33.65 on the left, then the value specified on the Identifier List tab is ignored, and the identifier is marked with the **Controlled** comment (Figure 33.65 on the right). Identifier controls that are not checked are not active and their current values from the identifier list are used for the current simulation.

Consider controls. To open the control description, double click on the corresponding line of the list. A detailed description of the tool can be found in [Chapter 4](#), Sect. *Identifier control*.

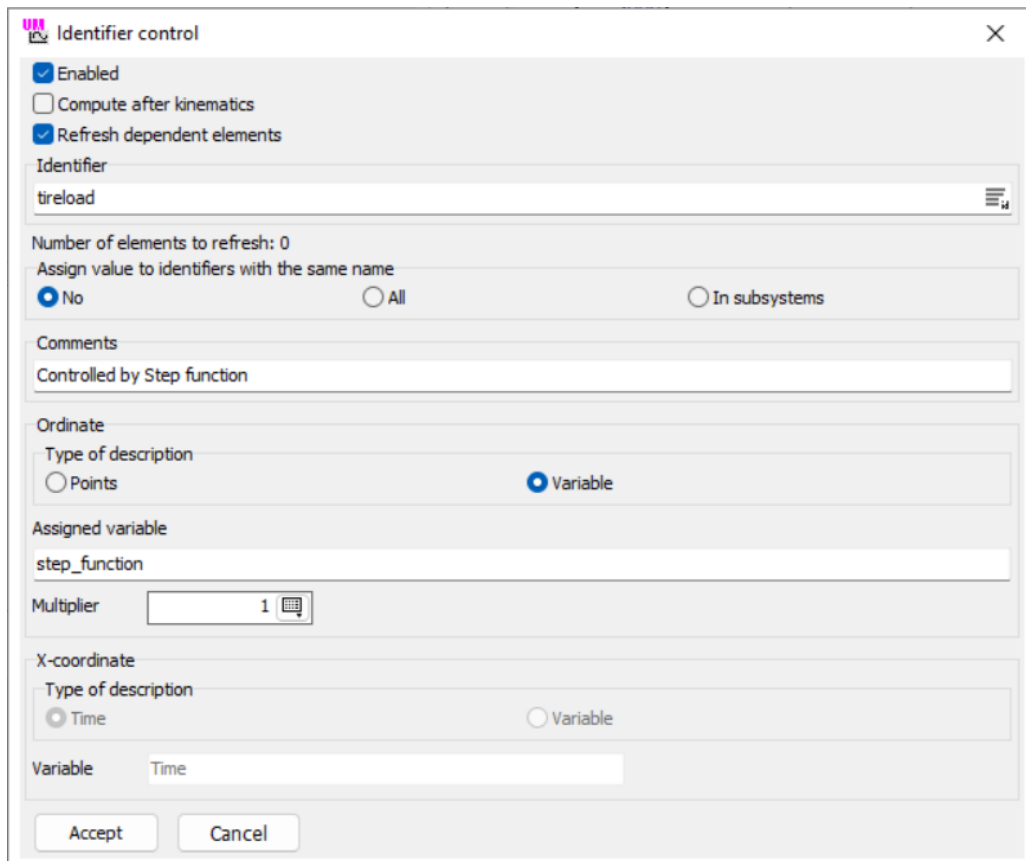


Figure 33.66. Control for identifier *tireload*

Tireload – the dependence of the wheel load on time is set using the *step_function* variable (Figure 33.66) included in the model in the UM Input program, Sect. 33.3.1.6 *List of variables*. The variable is available in the Wizard of variable (Figure 33.67) and is dragged into the corresponding box in Figure 33.66 using the mouse.

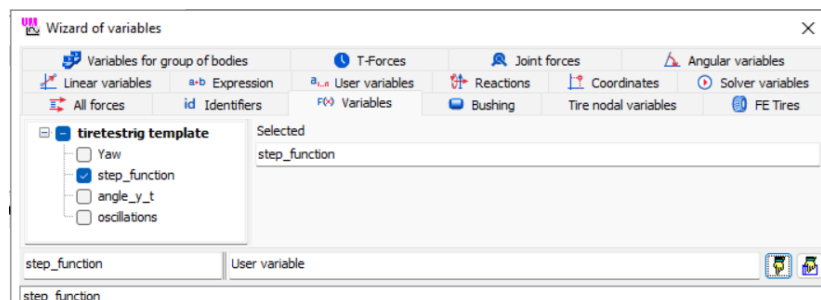


Figure 33.67. Variable *step_function* in wizard of variables

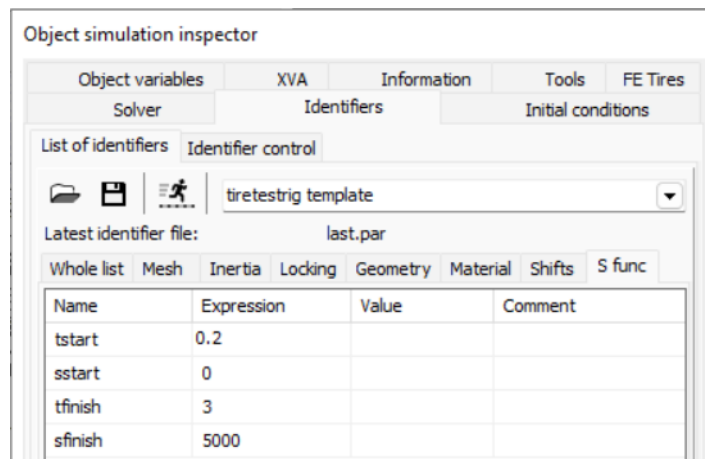


Figure 33.68. Example of *step_function* parameters for tire load

This control is used in the test II. 33.3.13.4.2 *Dependence of tire deflection on load*. An example of the step function parameters for tire load control is shown in Figure 33.68.

x_t – time dependence of the longitudinal shift of the hub specified by the *step_function* variable; used to evaluate the longitudinal static stiffness of a tire in the test 33.3.13.4.3 *Static tests for evaluation tire stiffness for lateral, longitudinal wheel shifts and rotation about vertical axis* .

y_t – time dependence of the lateral shift of the hub specified by the *step_function* variable; used to evaluate the lateral static stiffness of a tire in the test 33.3.13.4.3 *Static tests for evaluation tire stiffness for lateral, longitudinal wheel shifts and rotation about vertical axis* .

Alpha – time dependence of the hub rotation about the vertical axis specified by the *step_function* variable; used to evaluate the angular static stiffness of a tire in the test 33.3.13.4.3 *Static tests for evaluation tire stiffness for lateral, longitudinal wheel shifts and rotation about vertical axis* .

33.3.13.1.4. Use of preliminary created configurations

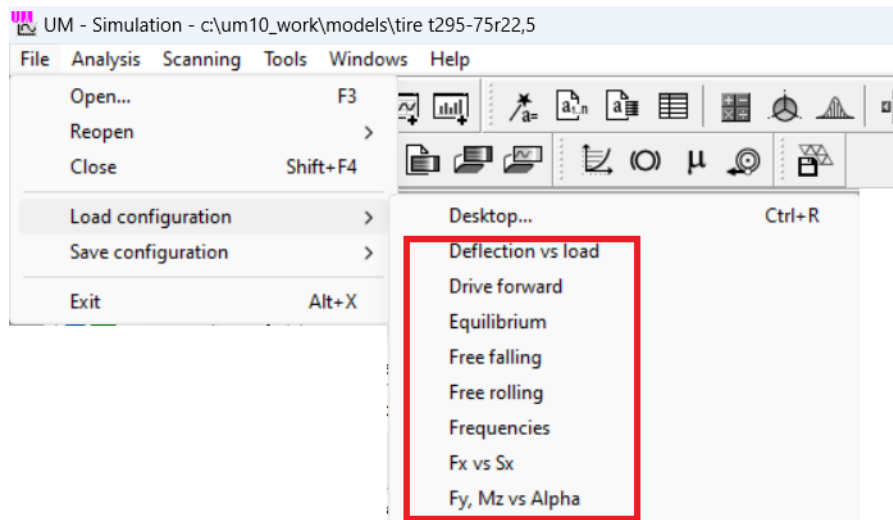


Figure 33.69. Example of full configurations

For all the tests described below, full configurations have been prepared (Figure 33.69) including *.par identifier files, *.icf desktop settings and numerical methods, *.xv initial conditions and *.ancl tire model parameters. These files can be used with a new tire model developed by the user, subject to a number of conditions.

- 1) The initial conditions file *.xv, as a rule, is incompatible with the new tire model and will be automatically ignored by the program if the number of coordinates in this file does not match the current number. For this reason, it is advisable to create files of the equilibrium state of the tire at the desired load values and use them in tests.
- 2) Some identifiers in the *.par file may be incompatible with the new model, for example, the wheel radius *rwheel* or identifiers that parameterize the graphic image of the wheel rim and the FE mesh. It is necessary to rename such identifiers in the UM Input program. Sect. 33.3.1.5 *List of identifiers*.

33.3.13.2. Computation of frequencies and eigenvalues

The tool for computation of the frequencies and eigenvalues of the linearized equations of motion of the wheel is available from the main menu command **Analysis | Static and linear analysis**, Figure 33.35.

To perform this calculation, the hub must be fixed and the coordinates of the tire nodes must correspond to equilibrium without taking into account the contact interaction of the tire with the road. It is important to note that **the contact interaction with the road in this mode is automatically disabled by the program.**

Note. Calculation of frequencies and natural values in a non-equilibrium tire position leads to incorrect results.

33.3.13.2.1. Computation of equilibrium for linear analysis

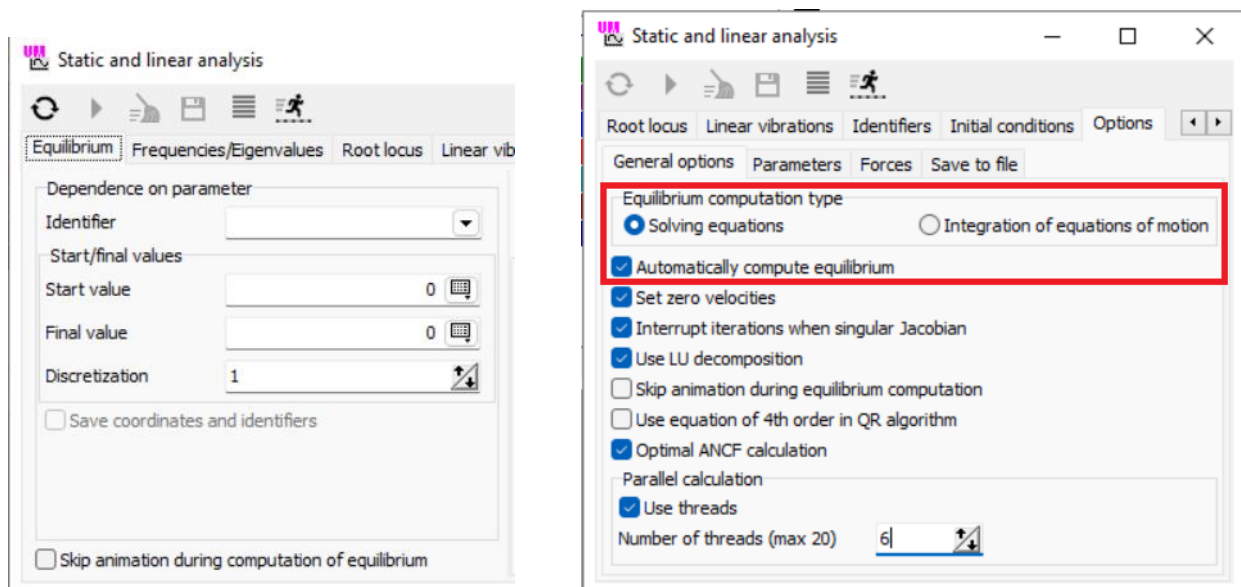



Figure 33.70. Tab for computation of equilibrium (left) and options

In the case when the tire inflation pressure is zero, equilibrium corresponds to zero coordinate values of the nodes, so you just need to set zero coordinate values on the initial conditions tab. If the pressure is non-zero, then two approaches can be used to calculate equilibrium, and the second approach is used if the first method does not give a positive result, that is, the direct solution to the equilibrium equations diverges.

Automatic computation of equilibrium.

- Set zero values of coordinates on the **Initial conditions** tab.
- Open the **Equilibrium** tab and run computation by the  button, Figure 33.70 left. If the calculation completes successfully, set the automatic calculation type as the main one on the **Option** tab, Figure 33.70 right.
- If the calculation iterations do not converge, then the second approach should be used to calculate the equilibrium.

Computation of equilibrium by integration of equations of motion.

- The hub is fixed in all degrees of freedom.

- Zero value for coordinates are set.
- If contact with the road is possible at zero coordinates, then contact interaction is disabled: the number of contacts within the element is set equal to zero.
- The standard modeling process is started from the Simulation inspector. To assess the model state, a graph of the kinetic energy of the system can be used.

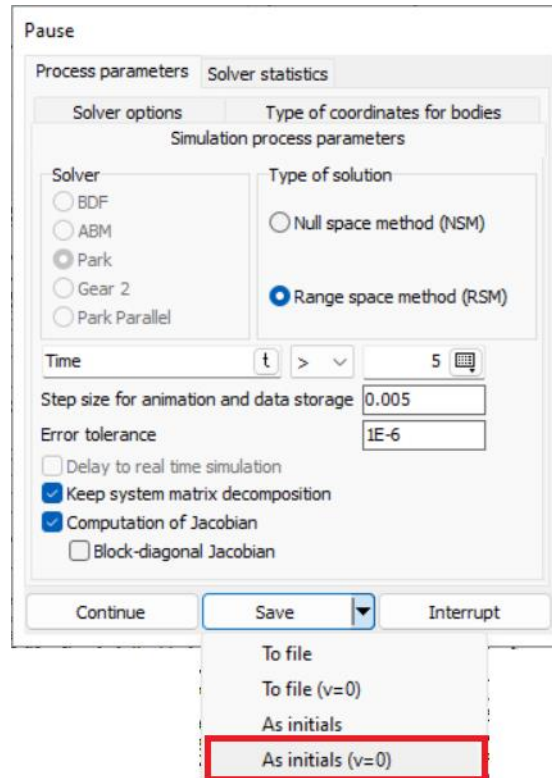


Figure 33.71. Assigning current coordinates and zero velocities as initial conditions

- After the model reaches a position close to equilibrium, the user should use the Save button in the pause mode so that using the **As initials (v=0)** command the current coordinate values and zero velocities are accepted as initial conditions, Figure 33.71.

33.3.13.2.2. Computation of frequencies

Since the tire has a large number of degrees of freedom, the Lanczos method is used to calculate frequencies, which allows calculating a given number of lower frequencies and modes. The number of frequencies is indicated in the upper left part of the window, Figure 33.72. For the same reason, an approximate calculation of eigenvalues is used, based on the reduction of matrices of linearized equations using the calculated modes; the number of frequencies and corresponding modes is indicated in the right part of the window and cannot exceed the calculated number of eigenfrequencies. For example, both numbers in Figure 33.72 are equal to 300.

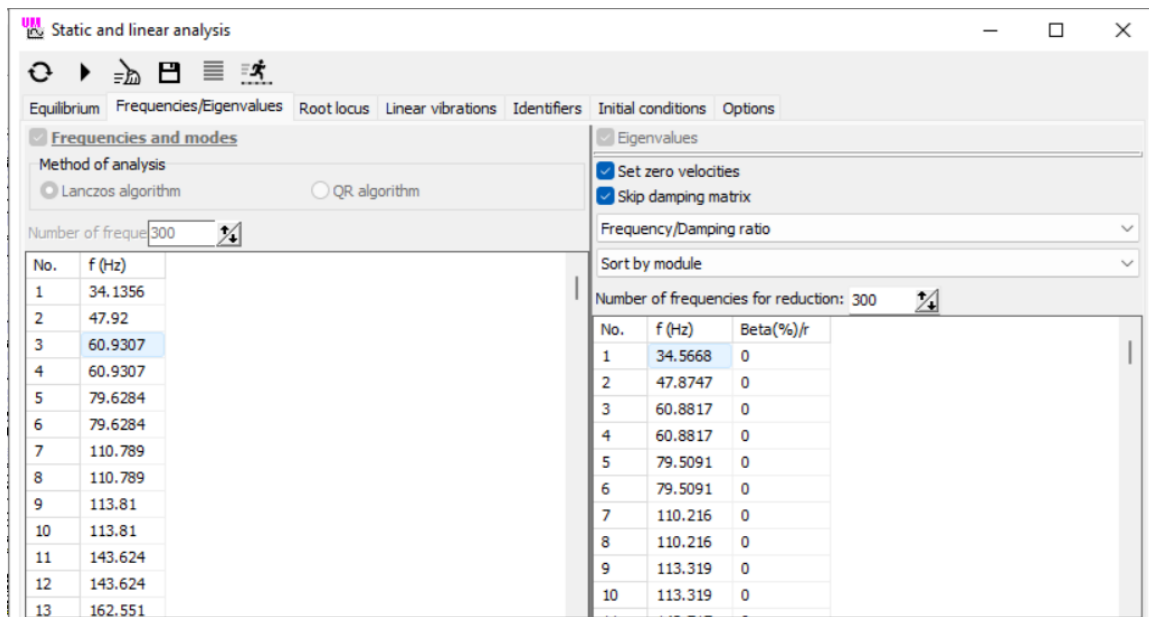



Figure 33.72. Computation of frequencies for nonzero tire pressure

To compute the natural frequencies of the tire in the equilibrium position, open the **Frequencies/Eigenvalues** tab. Since the stiffness matrix at non-zero tire pressure is not symmetrical, the natural frequencies are calculated approximately due to the forced symmetrization of the matrix, see Sect. 33.2.6 *Tire pressure*. More exact frequency values can be obtained by calculating the natural values of the tire equations with the dissipation matrix turned off (Figure 33.72 right). The calculation is started by clicking on the button .

As it can be seen from the calculation results in Figure 33.72, the approximate natural frequencies on the left side are close to their refined values on the right. In the absence of pressure, the frequencies almost coincide, Figure 33.73.

Frequencies in Figure 33.72 and Figure 33.73 are calculated for the same tire, only the pressure differs. In the first case, $p=2.2$ bar, in the second one $p=0$. The comparison demonstrates how strongly frequencies depend on pressure.

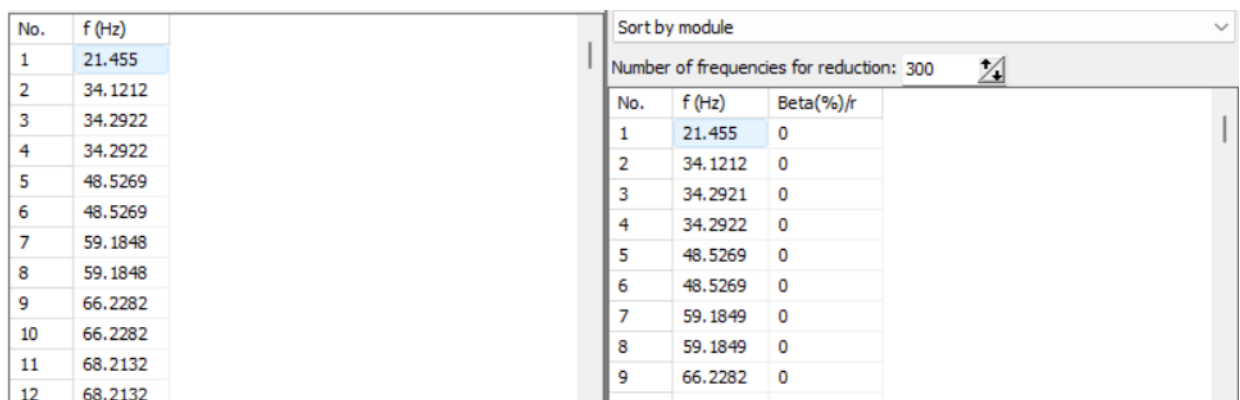


Figure 33.73. Computation of frequencies for zero tire pressure

33.3.13.2.3. Computation of eigenvalues taking into account damping

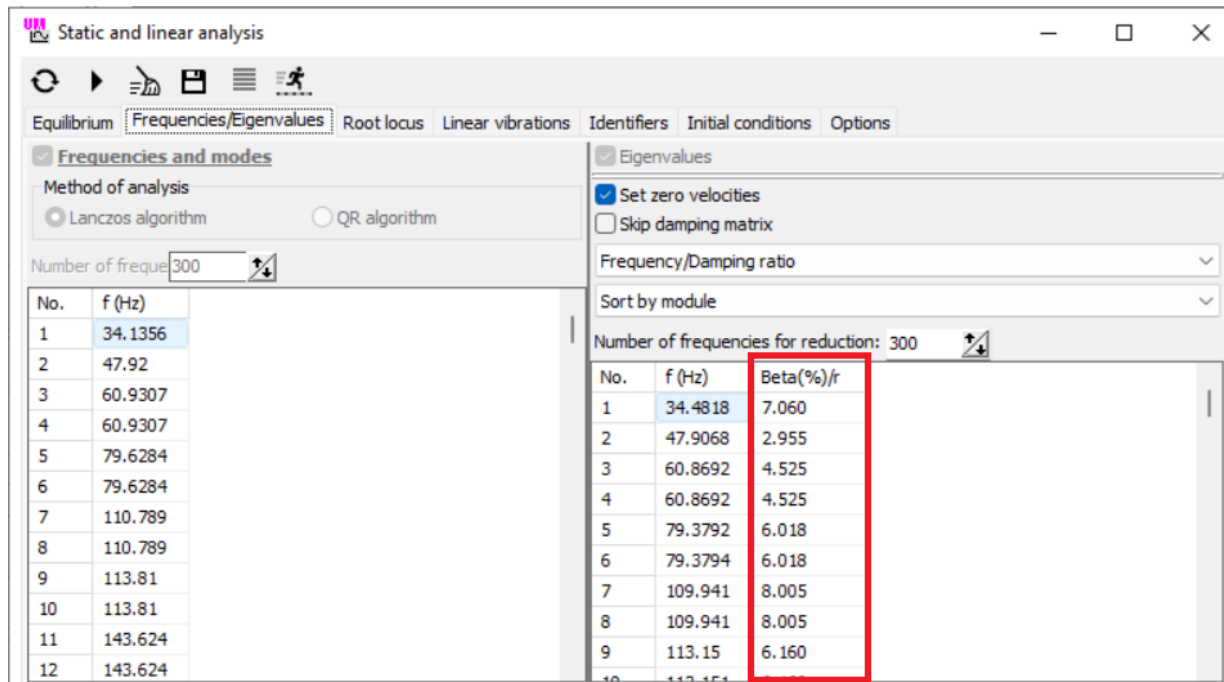




Figure 33.74. Computation of eigenvalues for non-zero pressure taking into account damping

To assess the degree of damping, the eigenvalues of the tire equations should be calculated taking into account the damping forces. To do this, the user should enable accounting of the damping matrix, clear the results of the previous calculation using the button  and perform the calculation by clicking on the button , Figure 33.74. The second column Beta in the eigenvalue results corresponds to the damping ratio for the corresponding frequency.

The calculation is performed approximately by reducing the matrices using the natural modes. Note that the results of the theoretical study given in Sect. 33.2.8 *Estimation of the influence of damping proportional to the stiffness matrix on eigenvalues*, when taking into account the tire pressure, are approximate. In the absence of pressure, the theoretical results are accurate.

Results in Figure 33.75 were obtained for the same tire as in Fig. 33.69, but for zero pressure. The value of the damping parameter in the calculations is $\beta=0.0005s$, which gives the critical frequency (33.12) $\omega^* = 4000rad/s = 636.62$ Hz. According to Eq. (33.13), this gives for the first frequency 21.455Hz a theoretical damping ratio of 0.0337 or 3.37%, for a second frequency 34.1212Hz the theoretical damping ratio is 0.0536 or 5.36%. Both results practically correspond to the calculated values in Figure 33.75.

Frequency damping ratio in Figure 33.74 is lower than in Figure 33.75 for most frequencies, which is a consequence of the fact that the internal damping in the tire is built without taking into account the pressure (see Eq. (33.1)).

Thus, a theoretical analysis of the effect of damping on tire frequencies in Sect. 33.2.8 *Estimation of the influence of damping proportional to the stiffness matrix on eigenvalues* gives accurate values in the absence of inflation pressure and allows the user to qualitatively and quantitatively evaluate this influence in the presence of pressure.

No.	f (Hz)
1	21.455
2	34.1212
3	34.2922
4	34.2922
5	48.5269
6	48.5269
7	59.1848
8	59.1848
9	66.2282
10	66.2282
11	68.2132
12	68.2132

No.	f (Hz)	Beta(%) / r
1	21.4428	3.369
2	34.0723	5.351
3	34.2424	5.381
4	34.2424	5.382
5	48.3855	7.617
6	48.3865	7.617
7	58.9279	9.297
8	58.9291	9.297
9	65.867	10.403
10	65.8705	10.403

Figure 33.75. Computation of eigenvalues for zero pressure taking into account damping

33.3.13.3. Estimation of internal damping by coefficient of restitution of bouncing tire

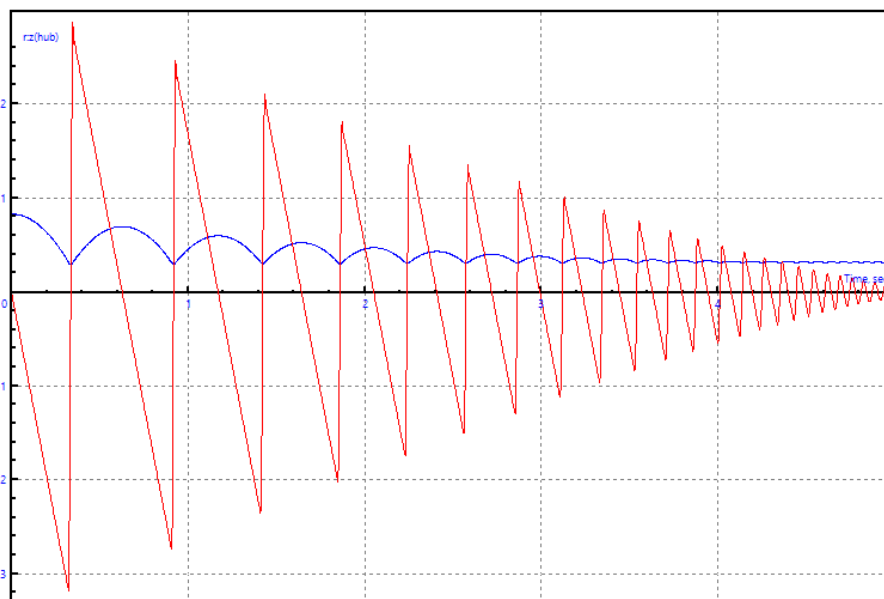


Figure 33.76. Vertical coordinate and velocity of wheel center for $\beta = 0.002s$

Consider a test that can be used to estimate the internal damping from experimental data. The wheel in the initial position is raised so that the distance from the bottom surface of the tire to the road surface is approximately 0.5 m. To do this, the h_0 identifier is set to 0.5. The wheel falls from this position under the influence of gravity and makes a series of bounces. Graphs of the vertical coordinate of the wheel center and its velocity are plotted for different values of the damping parameter β , Figure 33.76,. Simulation time is 5s.

The main test results are the restitution coefficient v_{i+1}/v_i and the height ratio h_{i+1}/h_i at bounce. Here v_i is the velocity at the moment the tire touches the road, that is, the minimum negative values in plots, and h_i is the maximum rise of the center of the tire above the position corresponding to the tire touching the road. Theoretically, these quantities are related to each other by the formula following from the law of conservation of energy:

$$h_{i+1}/h_i = (v_{i+1}/v_i)^2$$

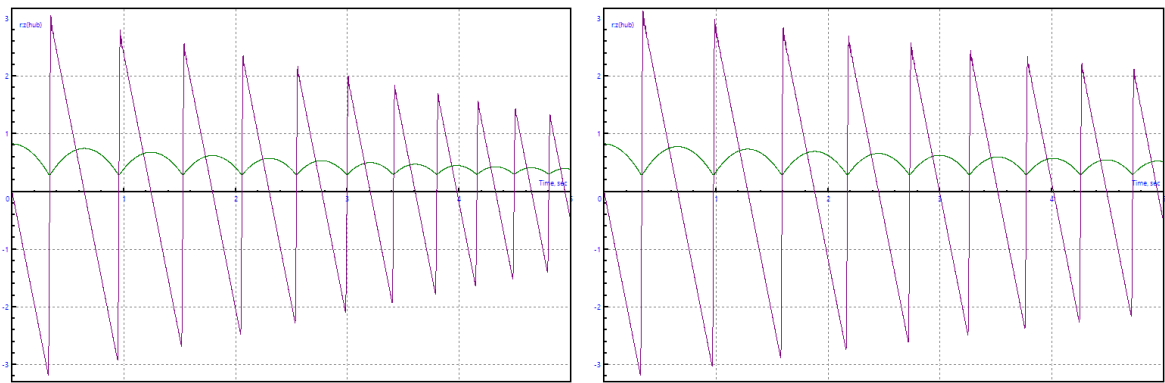


Figure 33.77. Wheel bouncing at $\beta = 0.001s$ and $\beta = 0.0005s$

The values of the coefficients for three damping parameters are given in Table 33.1. During five bounces, the recovery coefficient is almost constant, and the relationship between the coefficients almost exactly matches the theory, for example, $0.906 \approx 0.952^2$.

Table 33.1

β, c	v_2/v_1	v_3/v_2	v_4/v_3	v_5/v_4	v_6/v_5	h_2/h_1	h_3/h_2	h_4/h_3	h_5/h_4	h_6/h_5
0.002	0.862	0.858	0.860	0.862	0.862	0.740	0.739	0.740	0.740	0.740
0.001	0.918	0.918	0.921	0.919	0.920	0.844	0.845	0.847	0.845	0.847
0.0005	0.950	0.954	0.951	0.952	0.954	0.904	0.907	0.907	0.906	0.907

33.3.13.4. Static tests

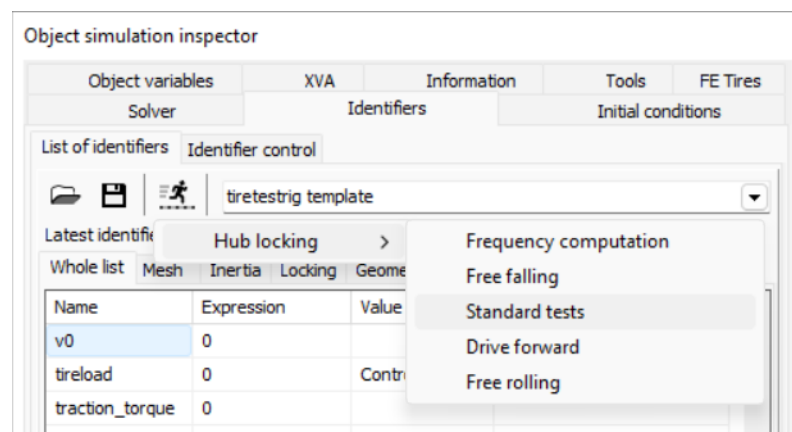


Figure 33.78. Macro for standard tests

These tests are performed while the rotation and longitudinal motion of the tire are blocked. To set the required values for locking identifiers, use the **Standard test** macro, Figure 33.78.

33.3.13.4.1. Equilibrium at a given tire load

The test is typically used to calculate the initial coordinates at equilibrium for a tire on the road at a given load. Before performing the test, the user should

- Set identifier values using the standard macro (Figure 33.78) and set the specify the desired load by the *tireload* identifier (N);
- check that all identifier controls are disabled;
- set zero values for the coefficient of static and sliding friction (the step can be skipped);
- if necessary, set zero values for all coordinates.

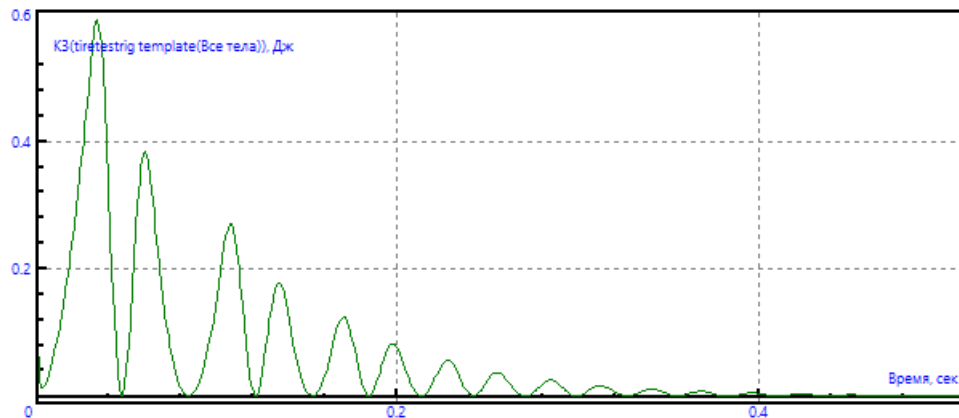


Figure 33.79. Drop of kinetic energy

Writing coordinates to the initial conditions file in pause mode

Start the simulation process. When the tire reaches a position close to the equilibrium one (Figure 33.79), save the calculated coordinate values to a file (Figure 33.80) with a name reflecting the state of the current model, for example, Equilibrium3kN50x11.xv - here it is clear from the file name that the tire mesh is 50 to 11 FE, and the load is 3 kN. **It is important to remember that when the mesh is changed, the initial condition files become incompatible and cannot be read.**

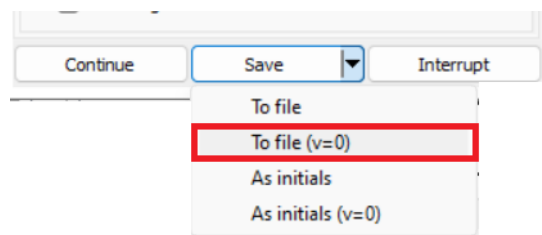


Figure 33.80. Writing coordinates to the initial conditions file in pause mode

33.3.13.4.2. Dependence of tire deflection on load

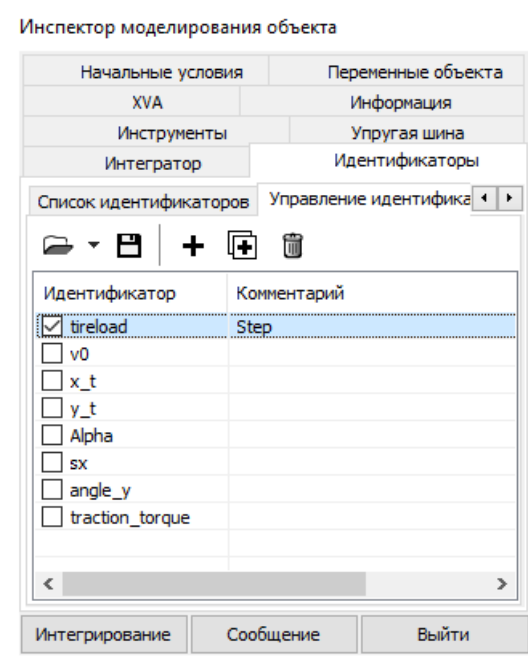


Figure 33.81. Identifier control for tire load

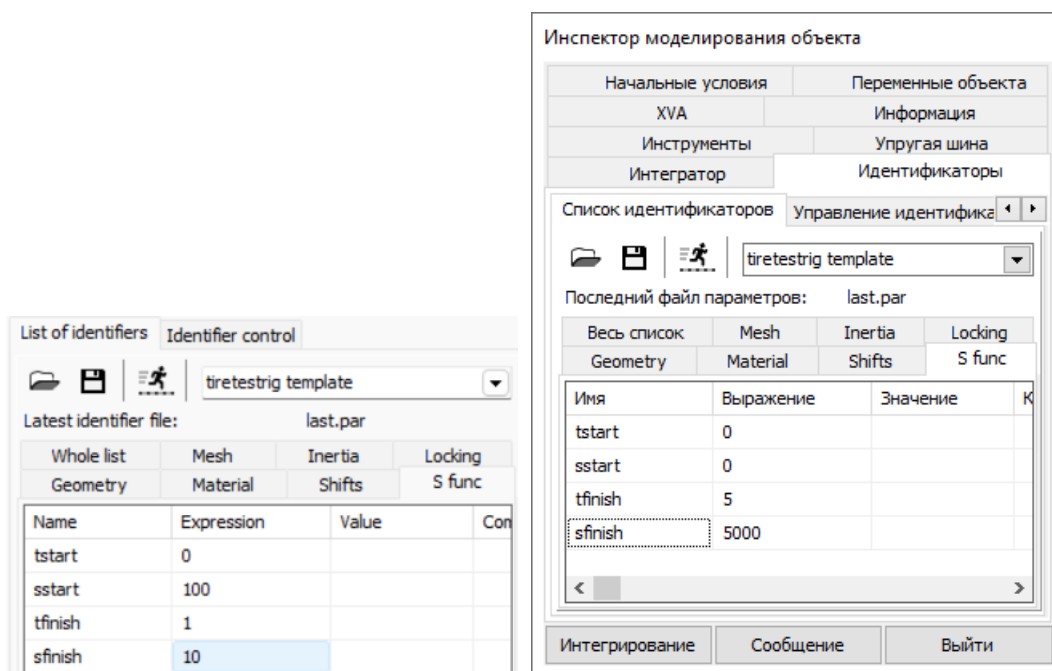


Figure 33.82. Load control parameters for equilibrium calculations at minimum load (left) and for test execution (right)

A standard test to evaluate the static deflection of a tire as a function of load and compare the computer model with the results of static bench tests.

Before running the test, the user should load the **Deflection vs load** configuration and make the necessary modifications related to the parameters of the tire under the test, or perform preliminary test preparation, which consists of the following steps.

- 1) configure identifiers using the **Standard tests** macro (Figure 33.78);

- 2) enable the *tireload* identifier control, which use the *step_function* variable (Figure 33.81 left, Sect. 33.3.1.6 *List of variables*);
- 3) bring the tire into equilibrium at minimum load by performing simulation with values of the step function parameters similar to those shown in Figure 33.82 left; in the example in the figure, the force decreases from 100N to 10N; The parameters of the step function are described in Sect. 33.3.1.6 *List of variables*;
- 4) set the *step* function parameters for running the test (Figure 33.81 right); the load in the example in the figure increases from the minimum value for which equilibrium is calculated to the maximum.
- 5) set zero values for the coefficient of static and sliding friction (optional operation);
- 6) generate variables corresponding to tire deflection and load.

Run simulation. An example of simulation result is shown in Figure 33.83.

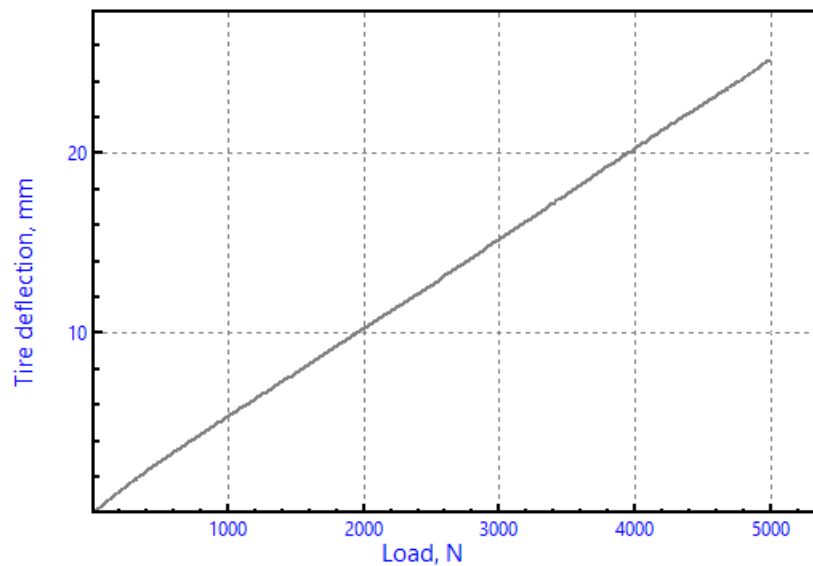


Figure 33.83. Tire static deflection vs. load

33.3.13.4.3. Static tests for evaluation tire stiffness for lateral, longitudinal wheel shifts and rotation about vertical axis

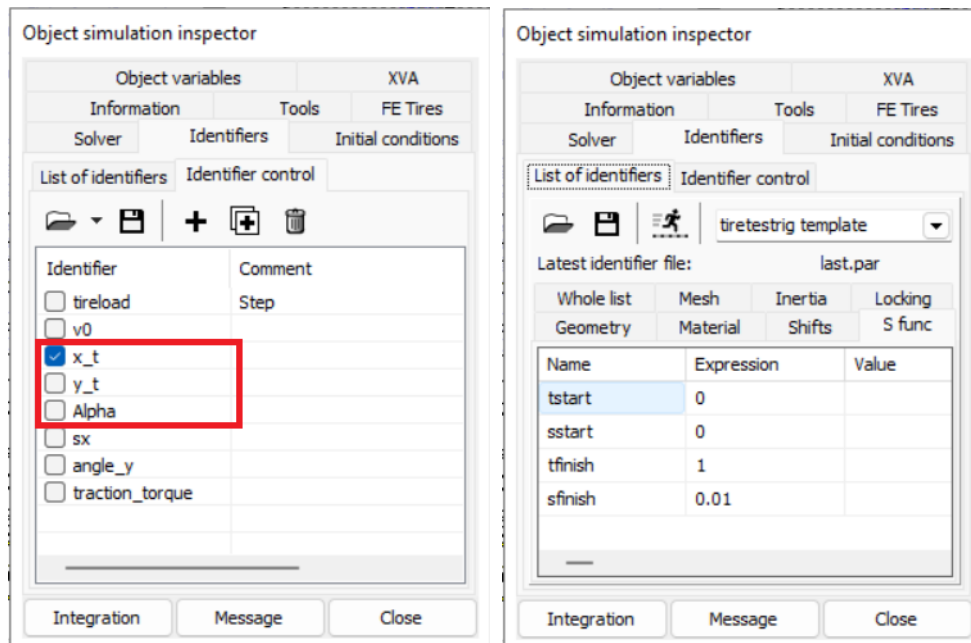


Figure 33.84. Identifier controls for shifts and rotation

Standard tests for assessing the static coefficients of lateral, longitudinal and angular stiffness of a tire and comparing the computer model with the results of bench tests.

Before performing the tests, the user should load one of the configurations **Longitudinal shift**, **Lateral shift**, **Static rotation Z** and make the necessary modifications related to the characteristics of the tire under test, or perform a preliminary test preparation, consisting of the following steps:

- 1) specify identifiers using the **Standard tests** macro (Figure 33.78);
- 2) set the desired value to the load identifier for which the test is performed using the *tireload* identifier, the value is set in N);
- 3) enable identifier management *x_t*, *y_t*, *Alpha* - depending on the type of test (Figure 33.84 left); assign values for the parameters of the step functions (Figure 33.84 right); shifts are specified in meters, so the value of *sfinish* should be on the order of 0.01, the angle is specified in degrees, so the value should be on the order of 0.5;
- 4) read or calculate the initial coordinates corresponding to the given load; it is recommended to set zero friction when compute the equilibrium position for the giver load;
- 5) set non-zero values for the coefficient of static and sliding friction (mandatory operation);
- 6) create a graphical window in which to place the variables corresponding to the desired shift (rotation) and force (moment) from the list **Standard variables** and accept the displacement as an abscissa.

If a full configuration is loaded then step 4 is usually required.

Start the simulation process. An example of the result is shown in Figure 33.85.

33.3.13.4.4. Evaluation of the parameters of a modified tire model with a reduced sidewall

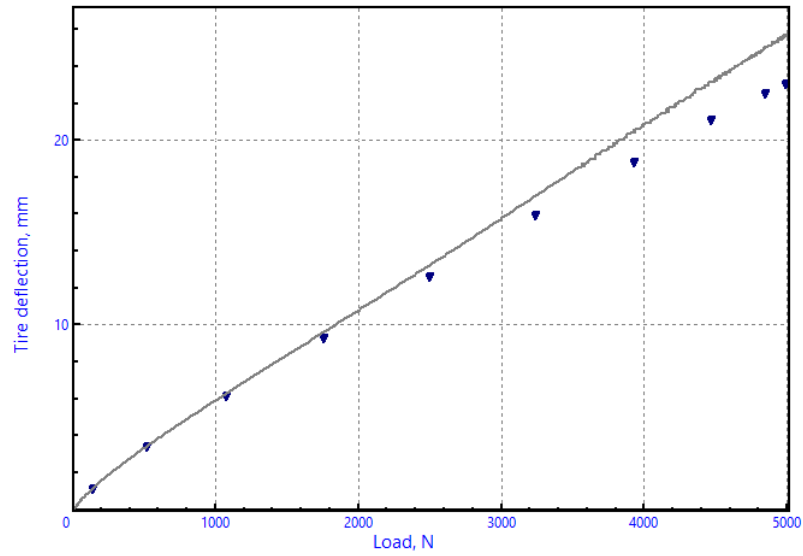


Figure 33.87. Comparison of tire deflection versus load for the full model (solid line) and the model with a reduced side parts (marker)

The tire model with a reduced side part has increased rigidity in tests of the dependence of deflection on load at rated and increased loads, Sect. 33.2.7 *Reduced tire models*. Figure 33.87 presents the corresponding test for the Dunlop 195/70 R15 tire. At a load of 2.5 kN, close to the nominal one, the deflection of the reduced model is approximately 5% less than that of the full model, and at a higher load of 5 kN the difference is more than 10%.

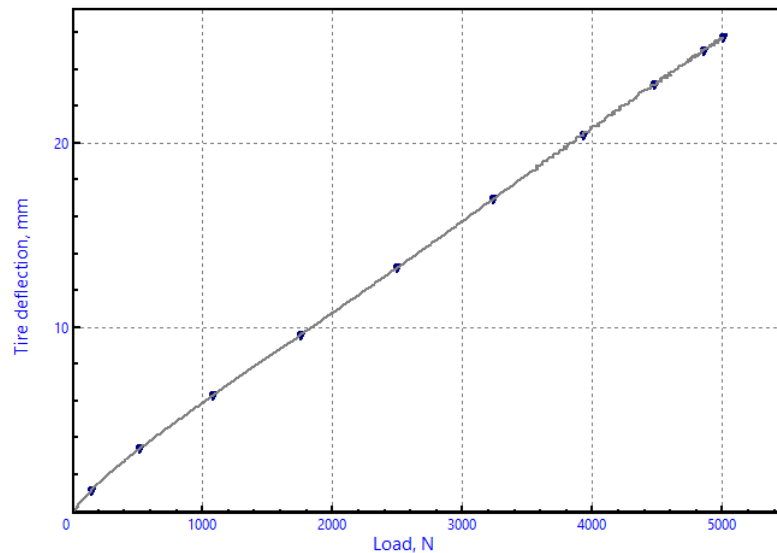


Figure 33.88. Comparison of tire deflection versus load for the full model (solid line) and the modified model with reduced side parts (marker)

To improve the model, UM implements a modification of the model with reduced side parts, Figure 33.88. The stiffness matrix of the reduced part in the modified model is multiplied by a reduction factor, which depends on the current and nominal load values, as well as on the stiffness reduction factor, which should be calculated for each tire model. The parameters of the modified model are set on the **Tire | Parameters | General** tab, Figure 33.89.

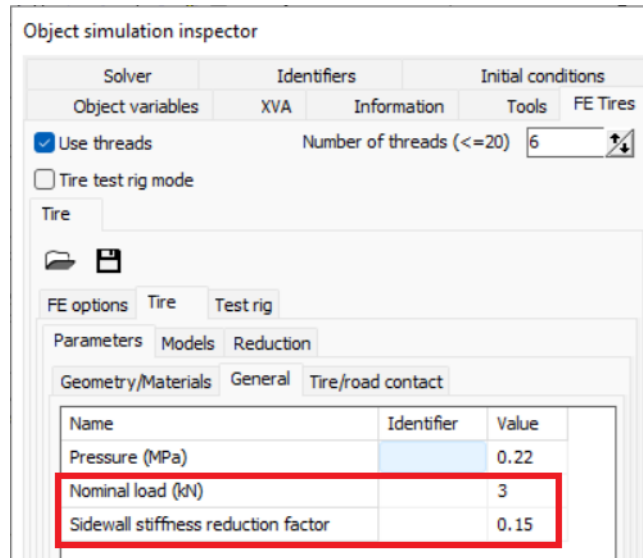


Figure 33.89. Parameters of the modified FE tire model with reduced side parts

Download the full configuration **RS stiffness reduction**. The parameters correspond to the test for calculating the dependence of tire deflection on load, that is, only the vertical motion of the tire is unlocked. Load the desired tire model on the **Tire | Models** tab, Figure 33.90. Set the nominal tire load and identifier *sr_reduction* parameterizing the stiffness reduction factor, Figure 33.91. The identifier control for the factor must be activated.

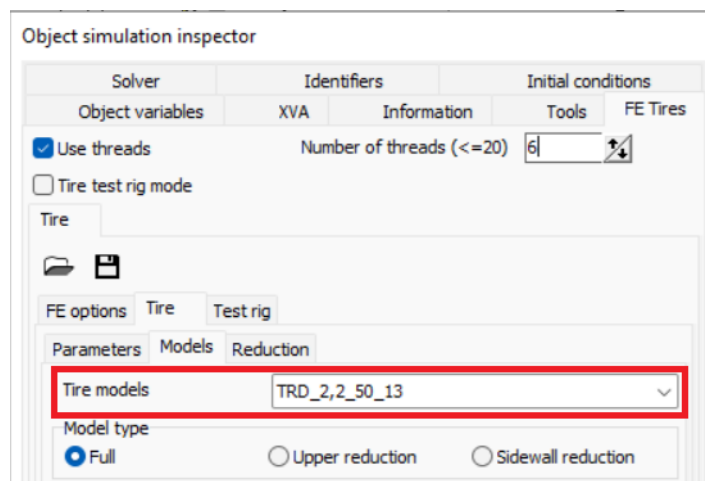


Figure 33.90. Loading tire model

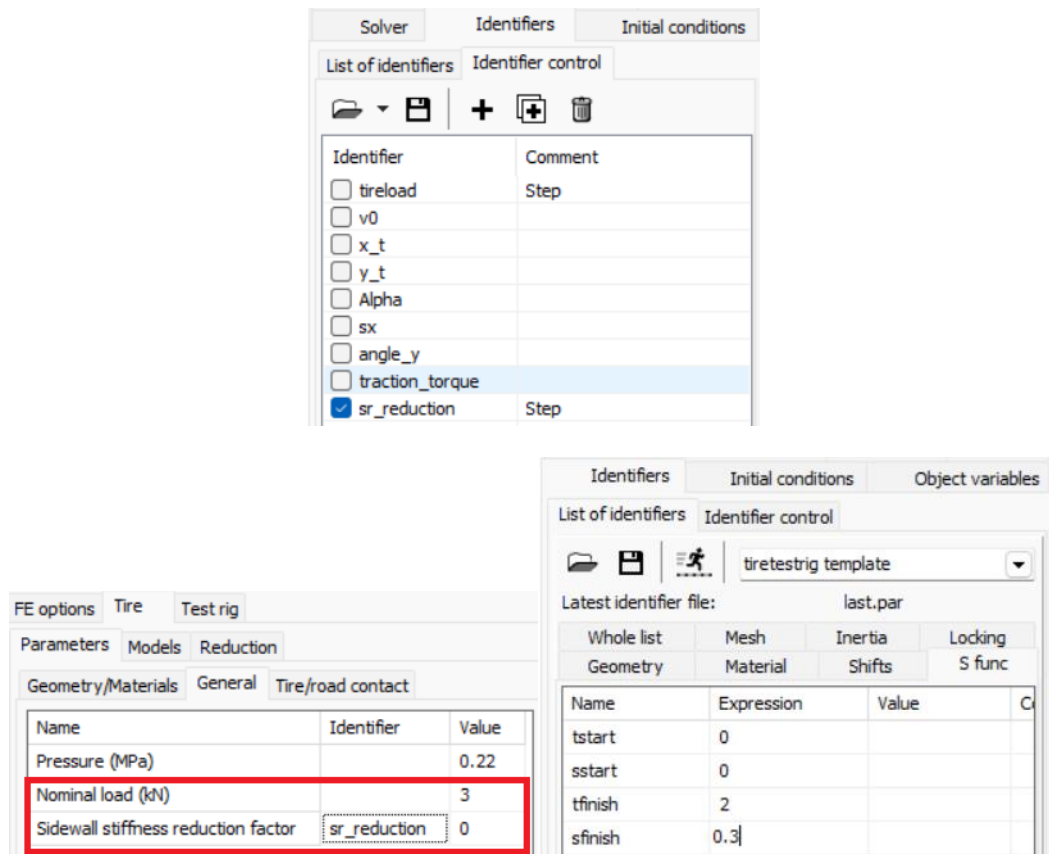


Figure 33.91. Identifier control for evaluation of sidewall stiffness reduction factor

The test consists of two steps: calculating the static deflection under maximal tire load and determining the value of the stiffness reduction factor:

- 1) set the model type: **full**;
- 2) set the tire load close to the maximum (approximately double the nominal) using the *tireload* identifier;
- 3) enable the identifier control (Figure 33.91) and run the simulation the main result of which is the dependence of tire deflection on time, Figure 33.92; the identifier *sr_reduction* is set as X-value in this window;

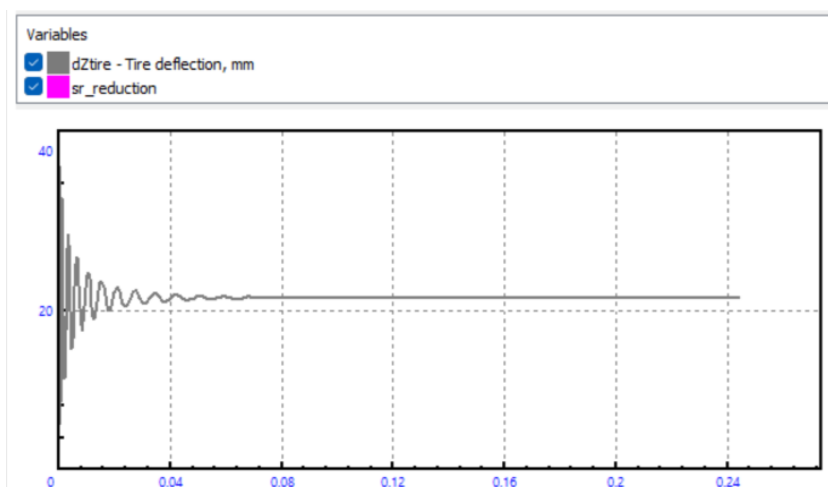


Figure 33.92. Tire deflection versus reduction factor for full model

- 4) copy the plot as static variable in the pause mode; please note that the reduction factor value is ignored in the full tire model, it affects the behavior of the tire model with reduced sidewalls only;

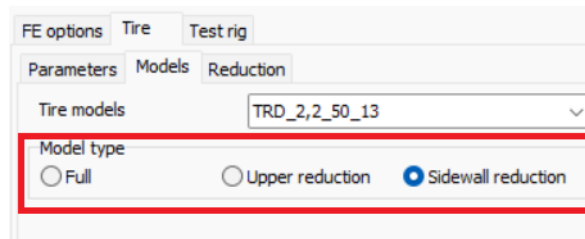


Figure 33.93. Activation of tire model with reduced sidewall

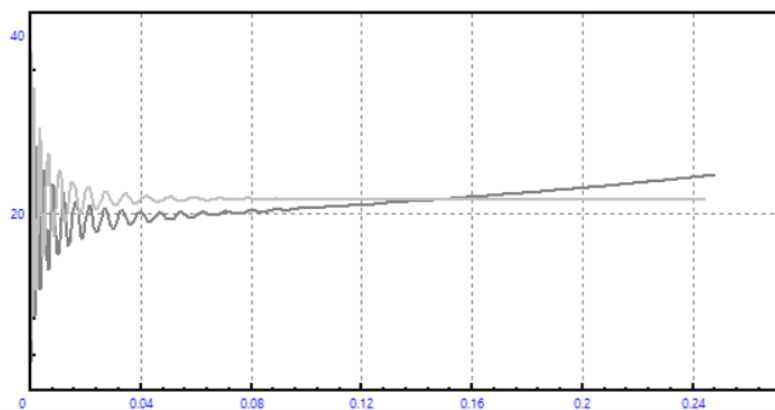



Figure 33.94. Comparison of deflection for full and reduced tire models

- 5) interrupt the simulation;
- 6) set the tire model with **reduced sidewall**, Figure 33.93;
- 7) run simulation and stop it when the deflection curves intersect, Figure 33.94.
- 8) the X value corresponding to intersection of graphs is the value of the reduction factor we need; it is about 0.15 in Figure 33.94.
- 9) set the calculated factor value to 0.15 in the parameters tab in Figure 33.91;
- 10) use the button  to renew the file *tire in the directory containing the reduced tire model; in our case the directory is TRD_2,2_50_13 and the file is TRD_2,2_50_13.tire, Figure 33.95.

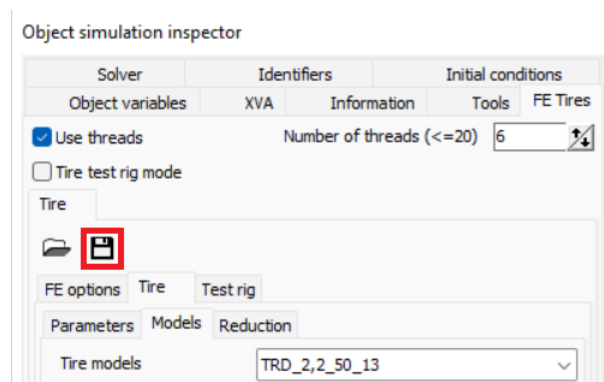


Figure 33.95. Saving tire parameters to file

33.3.13.5. Wheel rolling forward under constant torque

The wheel in this test rolls forward under the influence of a constant torque applied to the hub. The torque is specified by the *traction_torque* identifier. To limit the speed, linear longitudinal damping is used, specified by the identifier *d_x*. The velocity identifier of the **Ground** body is zero $v_0=0$.

Load the full **Drive forward** configuration before performing the test and make the necessary modifications related to the parameters of the tire under test or perform preliminary test preparation consisting of the following steps:

- 1) disable all identifier controls, Figure 33.65;
- 2) configure identifiers using the **Drive forward** macro;
- 3) set the desired value for the load identifiers *tireload* and *torque_torque*;
- 4) create variables: angular speed of rotation of the hub relative to the lateral axis and velocity of longitudinal movement of the hub; place the variables in the graphics window;
- 5) read or calculate the initial coordinates corresponding to the given load.

When the simulation starts, the wheel begins to roll, the speeds tend to the maximum values, Figure 33.96.

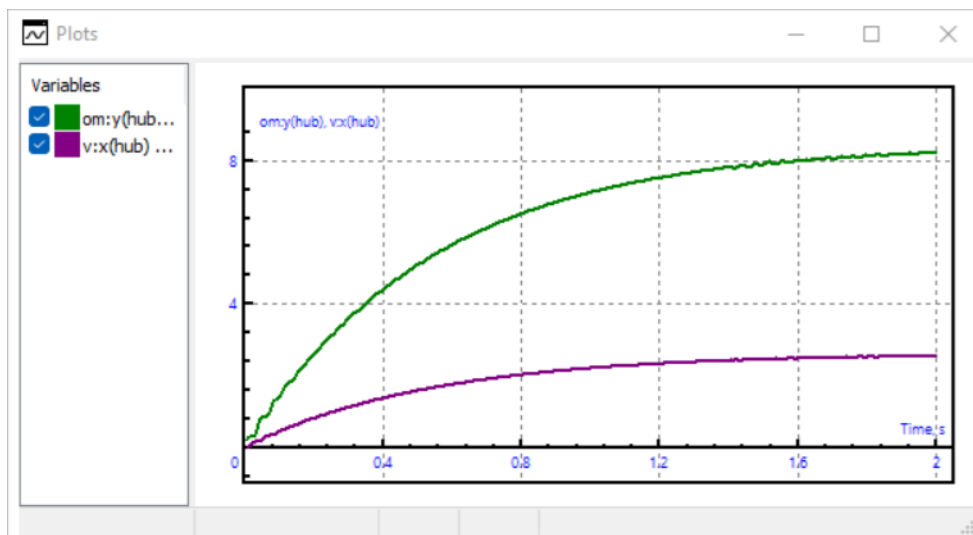


Figure 33.96. Longitudinal and angular velocity versus time

33.3.13.6. Evaluation of effective tire radius for freely rolling wheel

Consider an expression for the longitudinal tire slip [6]:

$$s_x = \left(1 - \frac{v}{r_{e0}\omega}\right) \times 100\%.$$

Here v, ω is the longitudinal and angular velocities of the wheel, r_{e0} is the effective radius of free rolling of the wheel. The radius is determined by the value of the angular velocity ω_0 in the absence of resistance to movement, while longitudinal slip is equal to zero ($s_x = 0$). Thus,

$$r_{e0} = \frac{v}{\omega_0}.$$

This test is used to determine the effective radius of freely rolling wheel.

Load the full **Free rolling** configuration before performing the test and make the necessary adjustments related to the characteristics of the tire being tested, or perform preliminary test preparation consisting of the following steps:

- 1) disable all identifier controls, Figure 33.65;
- 2) configure identifiers using the **Free rolling** macro, Figure 33.62;
- 3) set the desired value for the load identifiers *tireload* velocity v_0 ; v_0 sets the velocity of longitudinal motion of the **Ground** body, which is “pulled” under the wheel and simulates the roller of the test rig, Sect. 33.3.1.2.2 *Auxiliary body Ground*;
- 4) create variables: angular velocity of rotation of the hub relative to the lateral axis; place the variables in the graphics window;
- 5) read or calculate the initial coordinates corresponding to the given load.

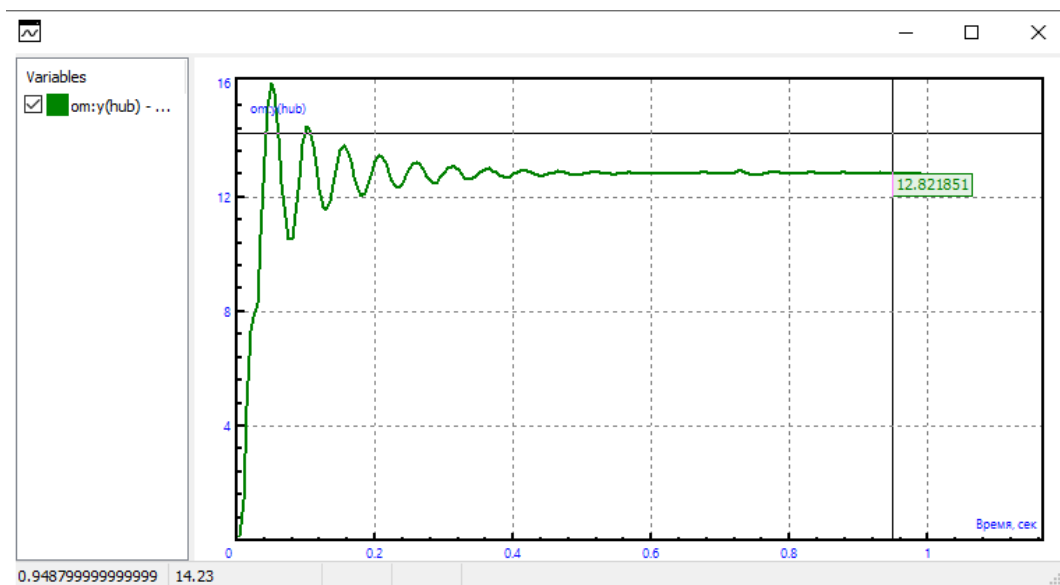


Figure 33.97. Angular velocity of hub vs time

Run the simulation. After some time, the angular velocity of the wheel becomes almost constant, Figure 33.97. To estimate the value of the effective radius, pick up the steady-state value of the angular velocity and divide the longitudinal velocity v_0 by this value. In the example in Figure 33.97 we have

$$r_{e0} = \frac{4}{12.82} = 0.312\text{m}.$$

According to the general theory of wheel rolling, this radius is slightly larger than the distance from the wheel center to the road under a given load of 3 kN, which is 0.310m in our example.

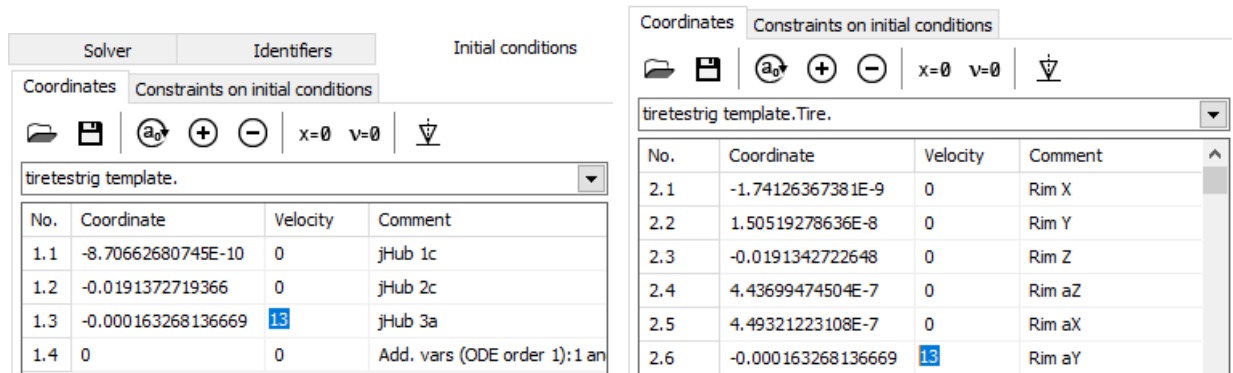


Figure 33.98. Setting the initial angular velocity of the hub and wheel

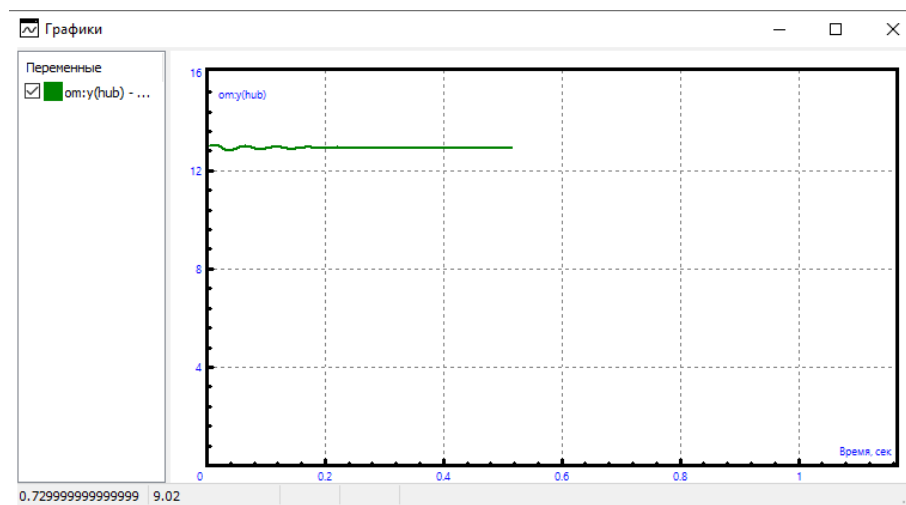


Figure 33.99. Dependence of the angular velocity of the hub on time at a non-zero initial angular velocity

Note that the transient process in Figure 33.97 can be significantly reduced if the hub and wheel are given an initial angular velocity close to the steady state. For example, the angular velocity, with an initial velocity of 13 rad/s (Figure 33.98) is shown in Figure 33.99.

33.3.13.7. Dependence of longitudinal force on slippage

The test determines the dependence of the longitudinal force F_x on the longitudinal slip s_x . Before performing the test, the user should load the full **F_x vs S_x** configuration and make the necessary adjustments related to the parameters of the tire being tested, or perform preliminary test preparation, which consists of the following steps.

- 1) Enable the identifier control for s_x , $angle_y$, Figure 33.100, left; the first of them corresponds to the longitudinal slip, and the second to the hub rotation angle. The identifier s_x is included in the expression for the dependence of the angular velocity on the longitudinal velocity, Figure 33.100, right. The variable $angle_y_t$ is assigned to the identifier $angle_y$; the variable $angle_y_t$ is equal to the integral of the angular velocity om_y , Sect. 33.3.1.6 *List of variables*.

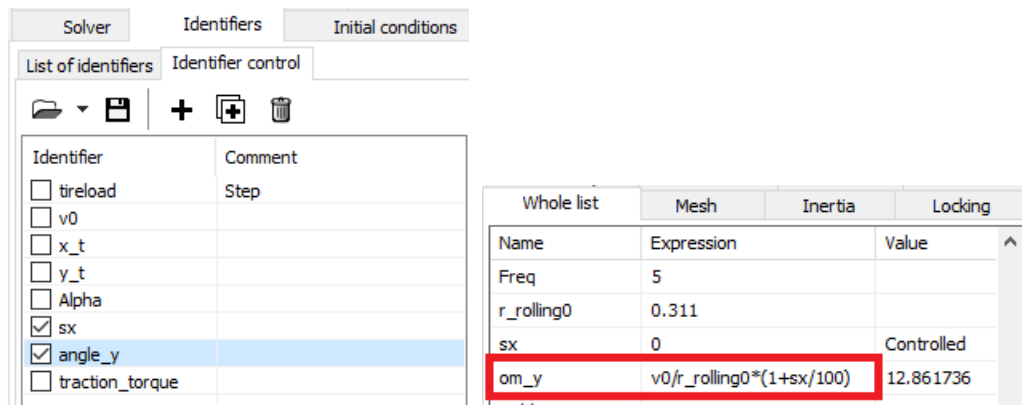


Figure 33.100. Options for test ‘Fx vs Sx’

2) Set up identifiers using a macro **Standard tests**.

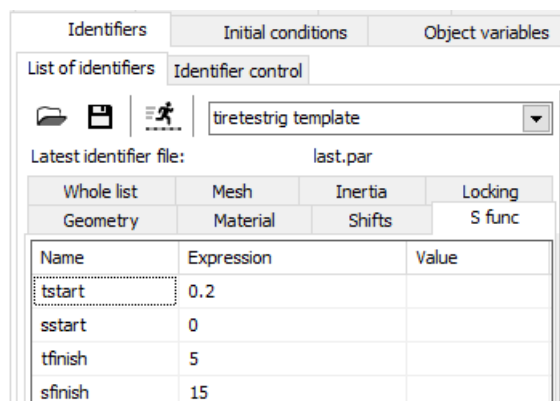


Figure 33.101. Example of setting parameters for controlling the slip identifier *sx*

- 3) The *step_function* is used for control the identifier *sx*. The user must specify identifiers corresponding to arguments of this function, Figure 33.101.
- 4) Set the required value for the identifiers of load *tireload*, speed *v0* and the identifier of the effective radius of the free rolling wheel *r_rolling0*, Sect. 33.3.13.6 *Evaluation of effective tire radius for freely rolling wheel*.
- 5) Open a new graphic window and copy into it the variables corresponding to the longitudinal force and slippage from the list of variables (Sect. 33.3.13.1.2 *List of variables for tests*). Plot the *sx* variable along the abscissa using the pop-up context menu.
- 6) Read or calculate the initial coordinates corresponding to the given load.

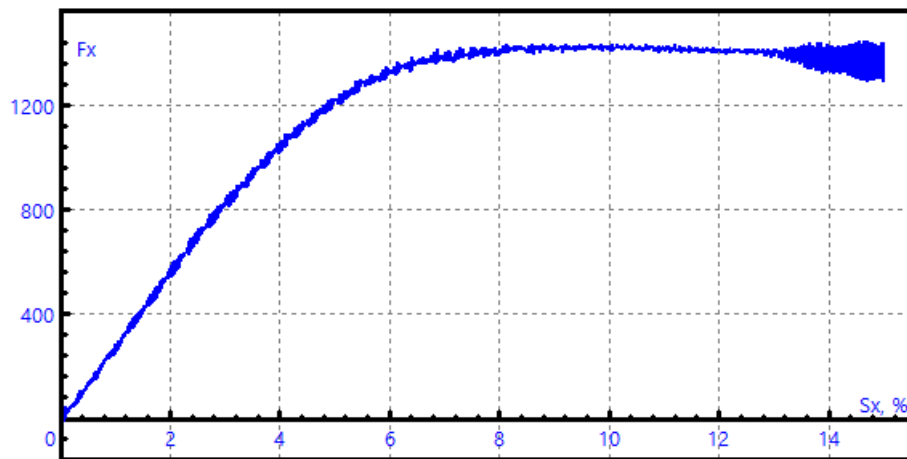


Figure 33.102. Example of dependence of longitudinal force on slippage

Run the simulation. A typical calculated dependence of the longitudinal force on slip is shown in Figure 33.102. The oscillations at big values of s_x correspond to the classical self-excited oscillations in the presence of a decreasing friction characteristic, Sect. 33.2.5 *Contact interaction of tire with road*.

33.3.13.8. Dependence of lateral force and aligning moment on lateral slip

The test determines the dependence of the lateral force F_y and the aligning moment M_z on the lateral slip angle $s_y = \alpha$ measured in degrees. Before performing the test, the user should load the full configuration **Fy, Mz vs Sx** and make the necessary adjustments related to the parameters of the tire being tested, or perform preliminary test preparation consisting of the following steps.

- 1) Enable control of the *Alpha, angle_y* identifiers; the first of them corresponds to the lateral slip angle, which equals in our case to the rotation of the wheel plane around the vertical axis; the second identifier is equal to the rotation angle of the hub.
- 2) Set up identifiers using a macro **Standard tests**.
- 3) To control the lateral slip identifier *Alpha*, the *step_function* variable is used. The user should specify the identifiers corresponding to the arguments of this function, Figure 33.101.
- 4) Set the required value to the load identifiers *tireload*, speed v_0 and the identifier of the effective radius of the free rolling wheel $r_{rolling0}$, Sect. 33.3.13.6 *Evaluation of effective tire radius for freely rolling wheel*.
- 5) Open two graphic windows and copy the variable F_y (the lateral force acting on the tire) and the angle *Alpha* from the list of variables to the first window, as well as the aligning moment M_z and the angle *Alpha* to the second window. Plot the variable *Alpha* along the abscissa in both windows using the pop-up context menu.
- 6) Read or calculate the initial coordinates corresponding to the given load.

Run the simulation. Typical calculated dependences of the lateral force and the aligning moment on the angle are shown in Figure 33.103, Figure 33.104. Oscillations at big values of *Alpha*

correspond to the development of self-excited oscillations in the presence of a falling friction characteristic.

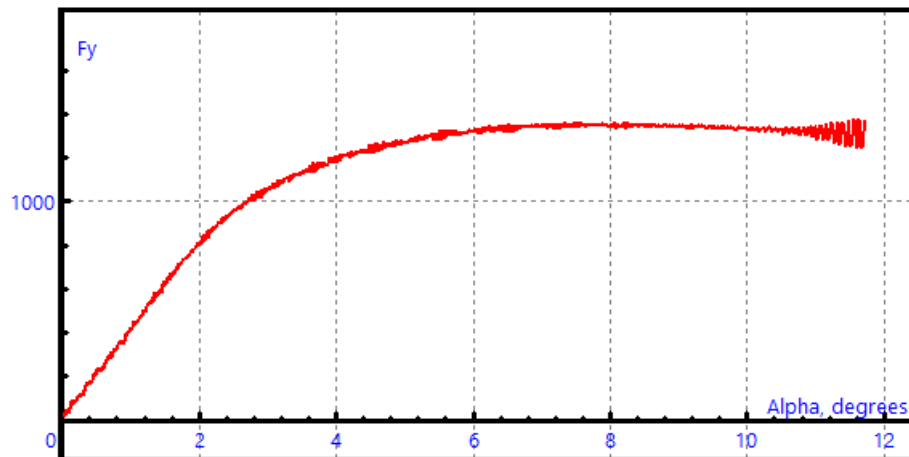


Figure 33.103. Example of dependence of transverse force on lateral slip angle

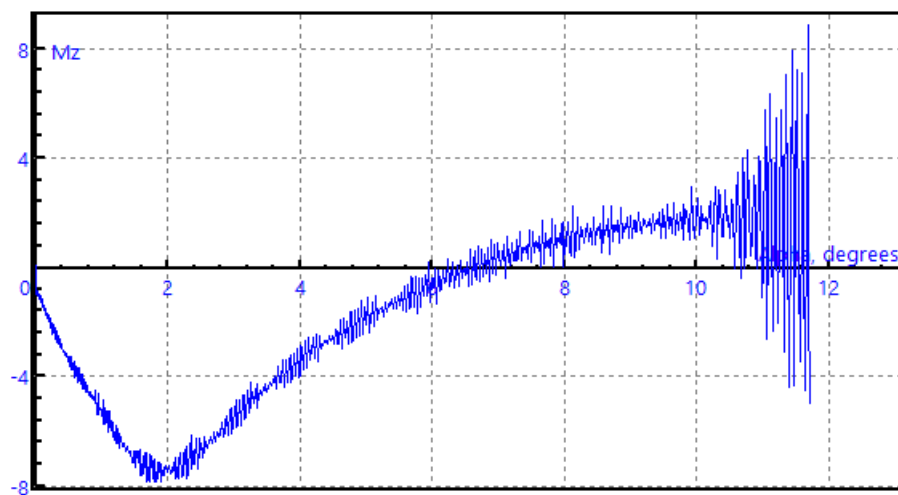


Figure 33.104. Example of dependence of aligning moment on lateral slip angle

33.3.14. Test rig mode: automatic study of tire properties

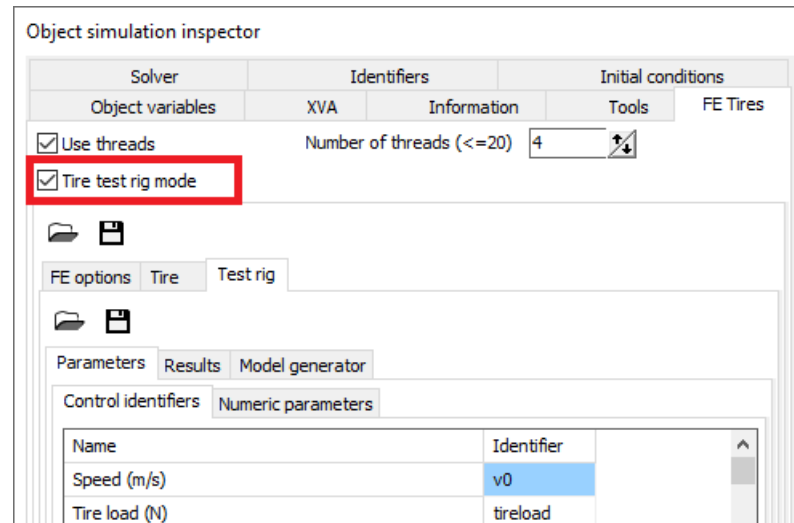


Figure 33.105. Tire test rig mode

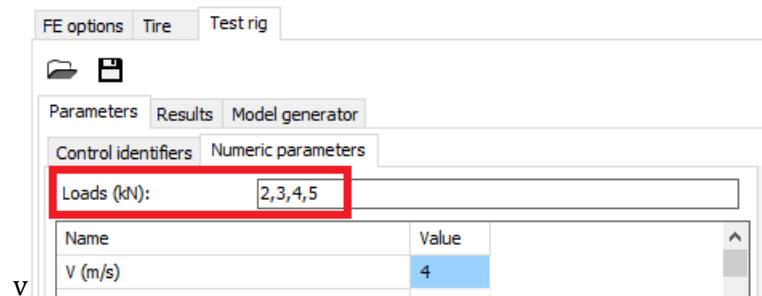


Figure 33.106. List of tire loads

The virtual test rig mode is the main tool for investigating the properties of a tire, Figure 33.105. This mode automatically performs a series of static and dynamic tests described above to obtain dependencies of the steady-state values of various tire performances depending on the load, Figure 33.106. The parameters and results of the test execution are displayed in the **Test rig** tab of the object simulation inspector. **The test rig mode is available provided that a single tire template is used**, 33.3.1 *Tire template in UM Input program*.

33.3.14.1. List of tests

Minimum Load Equilibrium Calculation

This is the first test performed at the beginning of the process to evaluate the tire's unloaded radius. The tire is balanced with a load equal to the tire's weight, see Sect. 33.3.13.4.1 *Equilibrium at a given tire load*.

Next, a series of tests is performed for each load from the list specified by the user, Figure 33.106. The series consists of five static and three dynamic tests.

1) Calculation of equilibrium under a given load

The tire is brought into equilibrium by a given load, see Sect. 33.3.13.4.1 *Equilibrium at a given tire load*. The test results include the tire's static deflection and the coordinate values at the equilibrium position. The coordinates are used as initial conditions for the remaining tests in the

series. The next four tests are performed to evaluate the static stiffnesses of the tire, see Sect. 33.3.13.4.3 *Static tests for evaluation tire stiffness for lateral, longitudinal wheel shifts and rotation about vertical axis*.

2) Calculation of longitudinal static stiffness of the tire

3) Calculation of lateral static stiffness of the tire

4) Calculation of vertical static stiffness of the tire

5) Calculation of static stiffness of the tire when turning around the vertical axis

Now let's look at the dynamic tests of the series.

6) Freely rolling tire

The test is used to calculate the effective radius of the tire when free rolling, Sect. 33.3.13.6 *Evaluation of effective tire radius for freely rolling wheel*.

7) Calculation of the dependence of the lateral force and aligning moment on the lateral slip angle

See Sect. 33.3.13.8 *Dependence of lateral force and aligning moment on lateral slip*.

8) Calculation of the dependence of longitudinal force on longitudinal slippage

See Sect. 33.3.13.7 *Dependence of longitudinal force on slippage*.

During the last two tests, steady-state values of forces and torque are determined with a step-wise increase in the lateral slip angle α and longitudinal slip s_x . Each step has two stages: first, the value of α or s_x is brought to the desired value using the s-function; the execution time of this stage is called the **Transition time**. In the second stage, transient processes are calmed down, and at the end of the stage the average steady-state value of the variables is calculated, the averaging time **TWindow** = 0.1s. Averaging is necessary due to possible self-excited oscillations, Figure 33.102, Figure 33.103.

Note.

It is recommended to disable the gravity force when performing tests in which a non-zero load on the wheel is specified. To disable it, set the zero value to the identifier $eg=0$. In this case, the steady-state value of the total normal force of interaction of the tire with the road is exactly equal to the specified load. The enabled gravity force corresponds to $eg=-1$.

33.3.14.2. Test rig parameters

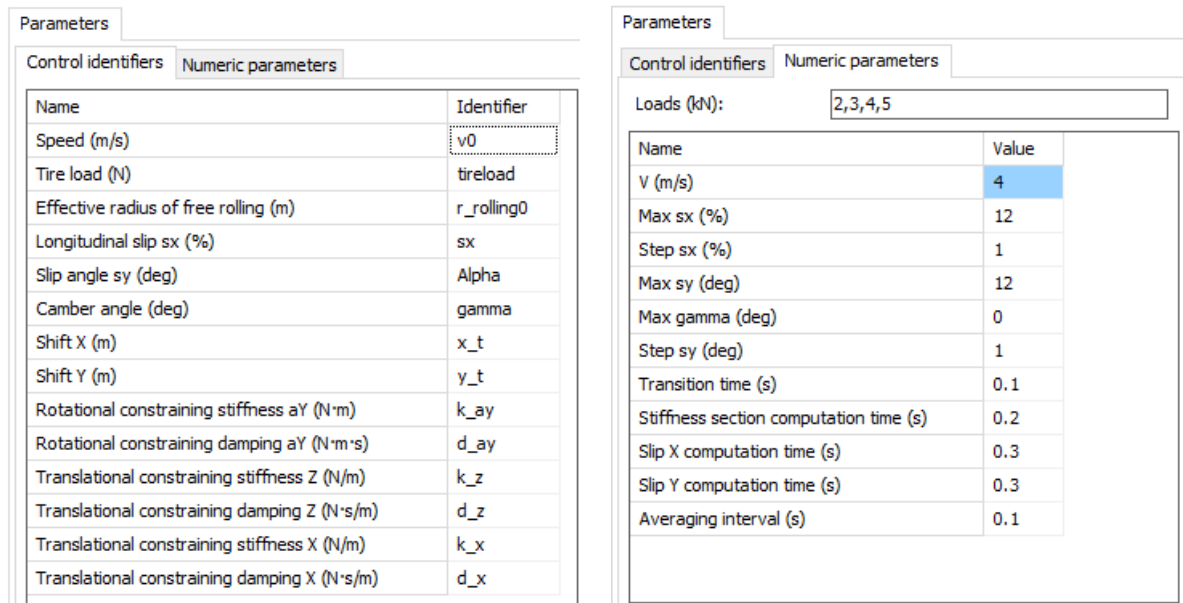


Figure 33.107. Tabs with test rig parameters

Parameters of the virtual test rig are located on two tabs, Figure 33.107.

Control Identifiers

The tab contains model identifiers that are used by the program to automatically manage the test series, Figure 33.107 left. If the user uses a tire template, the identifiers are already assigned.

Numeric parameters

The tab (Figure 33.107, right) contains the numerical values of the test setup parameters specified by the user.

- **List of loads:** loads in kN for which calculations are made; the loads are specified in ascending order separated by commas.
- **V (m/s):** speed value when performing tests with a rotating tire, i.e. tests 6)-8) in Sect. 33.3.14.1 *List of tests*.
- **Max s_x (%):** maximum value of longitudinal slip $s_{x,max}$.
- **Step s_x (%):** step with which the results of dependence on longitudinal slip are calculated. For example, with a step value of 1%, the longitudinal force F_x will be calculated for values $s_x = 1\%, 2\%, 3\% \dots s_{x,max}$.
- **Max s_y (deg):** maximum value of the lateral slip angle $\alpha_{max} = s_{y,max}$.
- **Step s_y (deg):** step with which the results of the lateral slip dependence are calculated. For example, with a step value of 1° , the lateral slip resistance force (lateral force) F_y will be calculated for values $s_y = 1^\circ, 2^\circ, 3^\circ \dots s_{y,max}$.
- **Transition time (s):** time interval T_t , during which the slip value s_x or s_y increases by the step value, Figure 33.108.
- **Stiffness section computation time (s):** the time interval during which the calculation of the static stiffness of the tire in one of the directions is performed, i.e. each of tests 2)-5) in Sect. 33.3.14.1 *List of tests*.

- **Slip X computation time (s):** time interval T_{cx} during which the value of slip s_x is kept constant in test 7), Sect. 33.3.14.1 *List of tests*, Figure 33.108.
- **Slip Y computation time (s):** time interval T_{cy} during which the value of slip s_y is kept constant in test 8), Sect. 33.3.14.1 *List of tests*, Figure 33.108.
- **Averaging interval (s):** time interval T_a during which the average values of the variables are calculated, Figure 33.108.

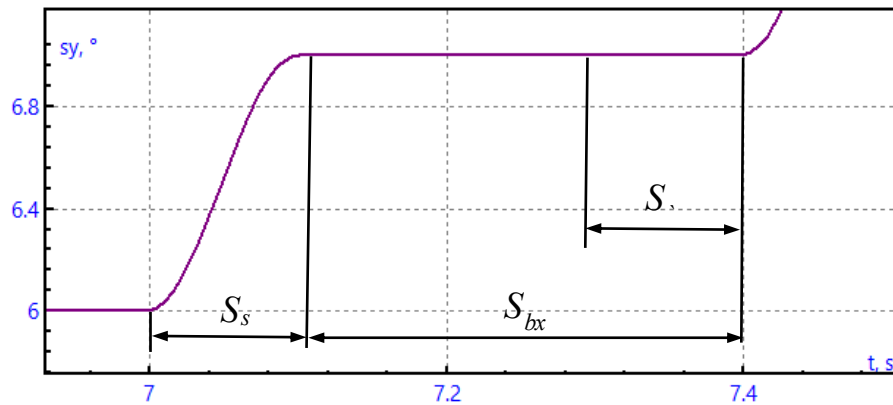


Figure 33.108. Fragment of controlled change of s_y

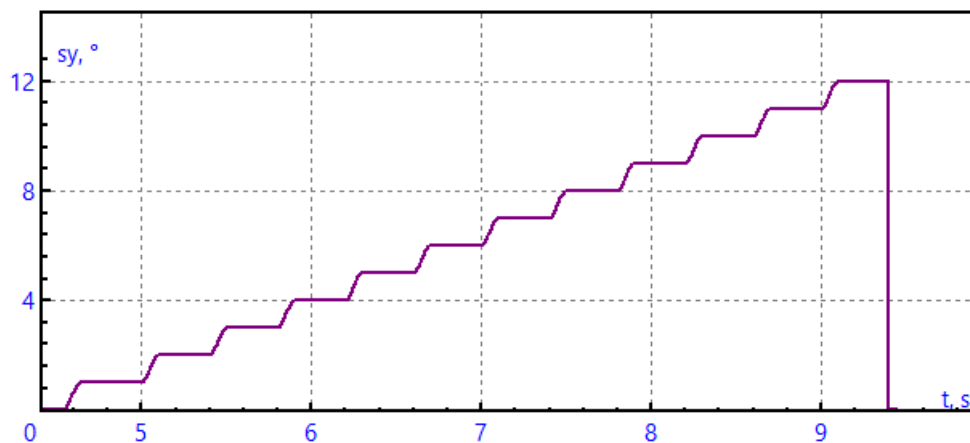


Figure 33.109. An example of controlled change in the dependence of the lateral slip angle s_y on time

Figure 33.109 shows an example of changing the lateral slip angle when performing test 8). The maximum value of the angle is 12° , the step is 1° . Time constants: $T_t=0.1s$, $T_{cy}=0.3s$, the time interval over which the calculated values are averaged is $T_a=0.1s$.

33.3.14.3. Performing a series of tests

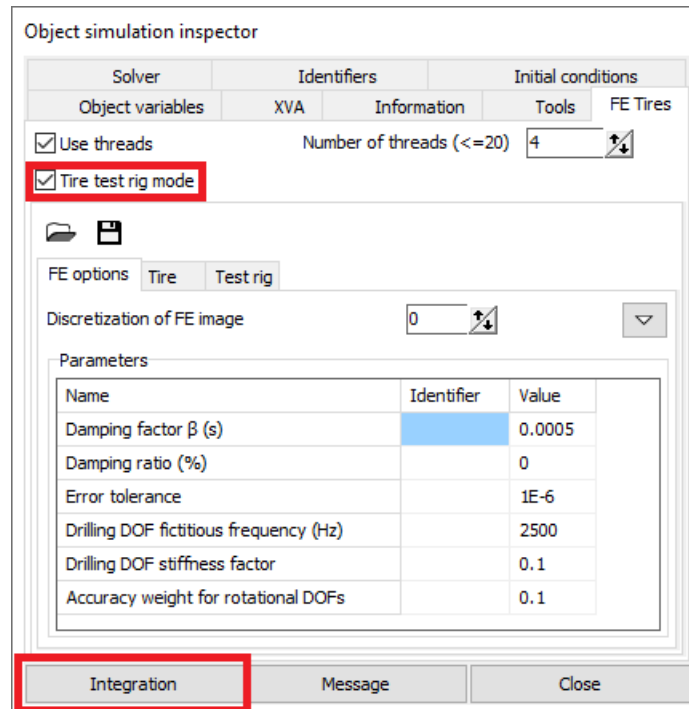


Figure 33.110. Starting the test rig mode

Test start

- Set sufficient simulation time in the **Solver** tab, for example 30s.
- Set test parameters, Sect. 33.3.14.2 *Test rig parameters*.
- Enable the test rig mode, Figure 33.110.
- Run the simulation be the **Integration** button.

When starting the simulation, the program automatically performs some necessary actions:

- disables all identifier controls;
- sets zero values for velocities.



Figure 33.111. Process window

The process of performing a series of tests

During the simulation, the program automatically performs a series of tests described in Sect. 33.3.14.1 *List of tests*. The process is displayed in a special window indicating the current wheel load and a comment on the current test, Figure 33.111. It is recommended to use graphic windows displaying the calculated values during the test. For example, Figure 33.112 shows an example of plotting a lateral force graph during the execution of the corresponding test; the graph corresponds to the change in lateral slip in accordance with Figure 33.109.

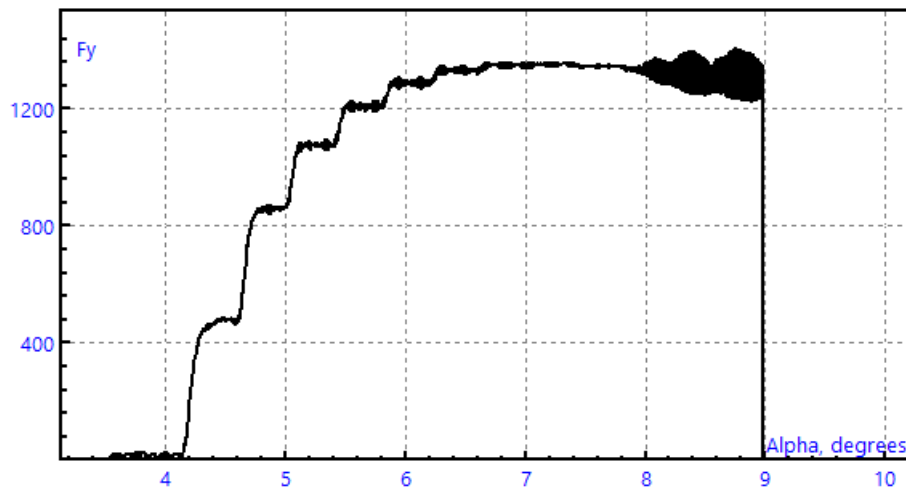


Figure 33.112. Lateral force versus the lateral slip angle

Completion of the test

The calculation can be interrupted by the user at any time with a transition to the pause mode. After that the user can either continue the calculation or *interrupt it with the loss of uncalculated data*. The calculation is completed automatically after the entire series of tests. If too small simulation time is set, the program goes into the pause mode, the time can be increased and the process can be continued.

After completing the calculation, the results should be saved to a file *.trr (test_rig_results) Figure 33.113.

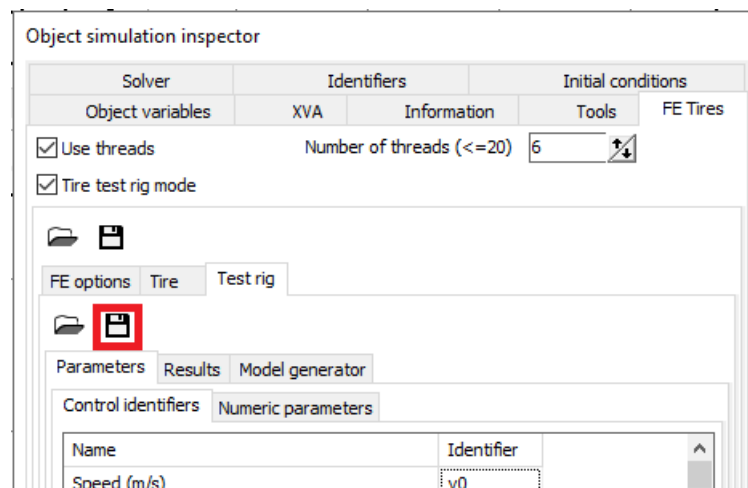


Figure 33.113. Saving test results to a file *.trr

33.3.14.4. Test rig simulation results

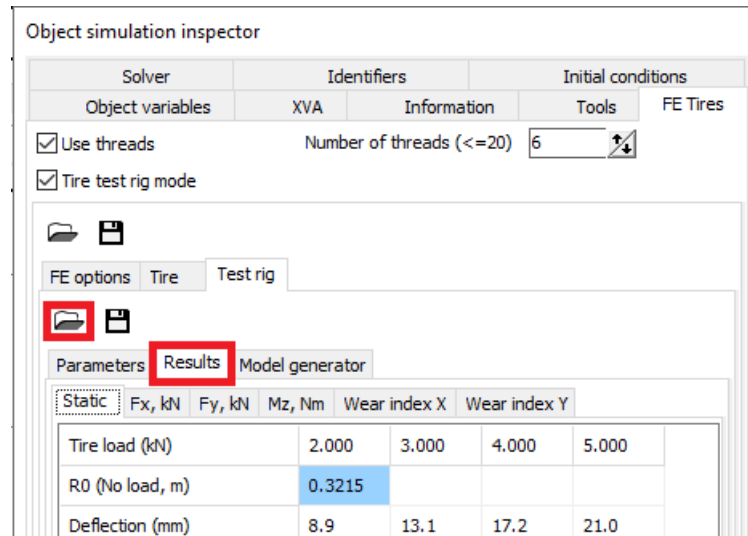



Figure 33.114. The button for reading result files *.trr and the tab containing the results

The results of the virtual test rig simulation are available in two cases: either immediately after the calculation is completed, or after reading the results file using the button , Figure 33.114. The **Results** tab appears, which contains tables of the numerical values of the calculated variables.

Along with the tables, the results can be obtained in graphical windows via the pop-up menu using the right mouse button, Figure 33.115.

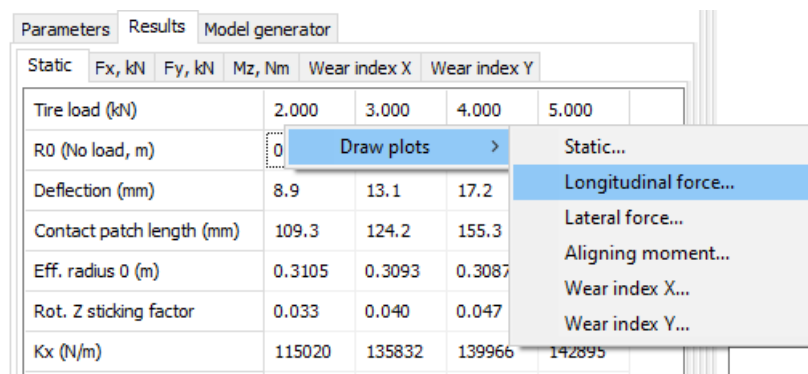


Figure 33.115. Pop-up menu for plotting results

Static results

The table contains mainly the results of static tests, Figure 33.116.

R0 (No load, m) – radius of the wheel without load, result of the initial test of the wheel balance under light load, see Sect. 33.3.14.1 *List of tests*.

Deflection (mm) – static deflection of a tire under a given load, Figure 33.117.

Contact patch length (mm) – the size of the tire contact patch in the longitudinal direction.

Eff. radius 0 (m) – effective radius of the tire when free rolling, Sect. 33.3.13.6 *Evaluation of effective tire radius for freely rolling wheel*.

Static	Fx, kN	Fy, kN	Mz, Nm	Wear index X	Wear index Y
Tire load (kN)	2.000	3.000	4.000	5.000	
R0 (No load, m)	0.3215				
Deflection (mm)	8.9	13.1	17.2	21.0	
Contact patch length (mm)	109.3	124.2	155.3	170.5	
Eff. radius 0 (m)	0.3105	0.3093	0.3087	0.3084	
Rot. Z sticking factor	0.033	0.040	0.047	0.054	
Kx (N/m)	115020	135832	139966	142895	
Ky (N/m)	124648	133441	139610	142197	
Kz (N/m)	238606	242988	259951	288155	
Kaz (N*m)	1189	2138	3121	3937	

Figure 33.116. Example of table with static results

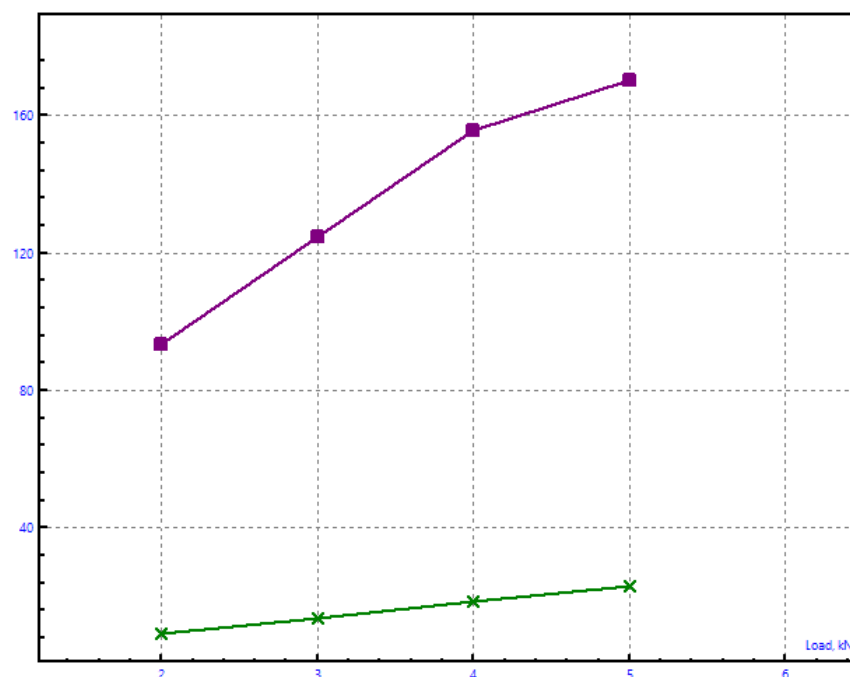


Figure 33.117. An example of the dependence on the load of the contact patch length (upper graph) and the static deflection of the tire

Rotation Z sticking factor – variable k_{mz} with the dimension of length, which is used to estimate from above the maximum value of the moment $M_{z,max}$ of resistance to rotation of a standing wheel around the vertical axis. An approximation of the moment is calculated using the formula

$$M_{z,max} \approx k_{mz} f_0 N,$$

where f_0 is the coefficient of static friction, N is the wheel load. In the example in Figure 33.116 we have $k_{mz} = 0.041m$ for the load $N=3000N$. For $f_0 = 0.8$ we obtain $M_{z,max} \approx 98Nm$. Figure 33.118 shows an example of a graph of the dependence of the moment M_z on time when simulating the rotation of a standing wheel about the vertical axis, see Sect. 33.3.13.4.3 *Static tests for evaluation tire stiffness for lateral, longitudinal wheel shifts and rotation about vertical axis*. In the graph, the maximum value of the moment $M_{z,max} \approx 96Nm$, which agrees well with the obtained estimate.

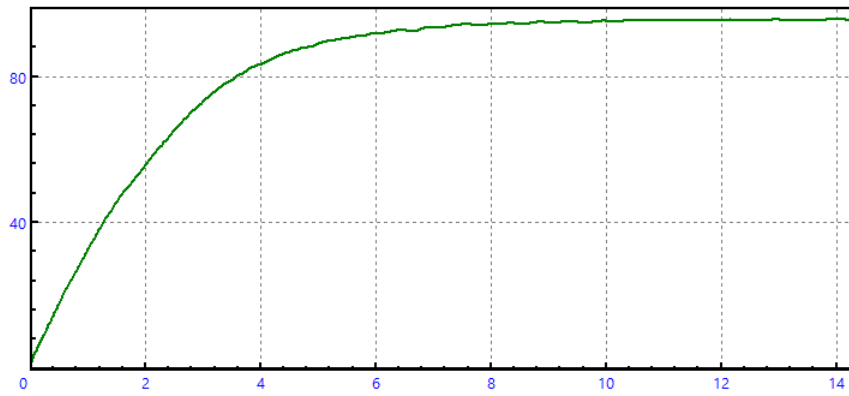


Figure 33.118. Example of the time dependence of the moment of resistance to rotation

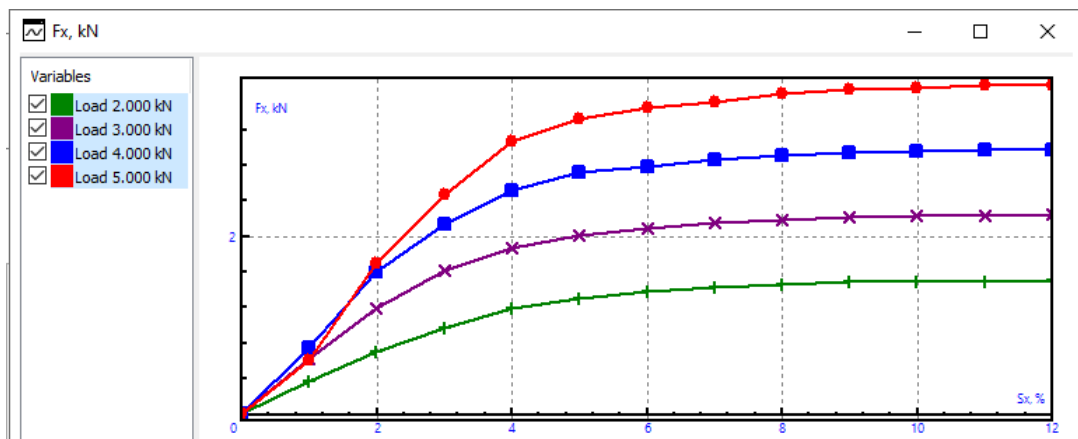


Figure 33.119. Example: longitudinal force vs. slip

Longitudinal force, kN – dependence of the longitudinal force F_x (kN) on slip s_x (%), Figure 33.119.

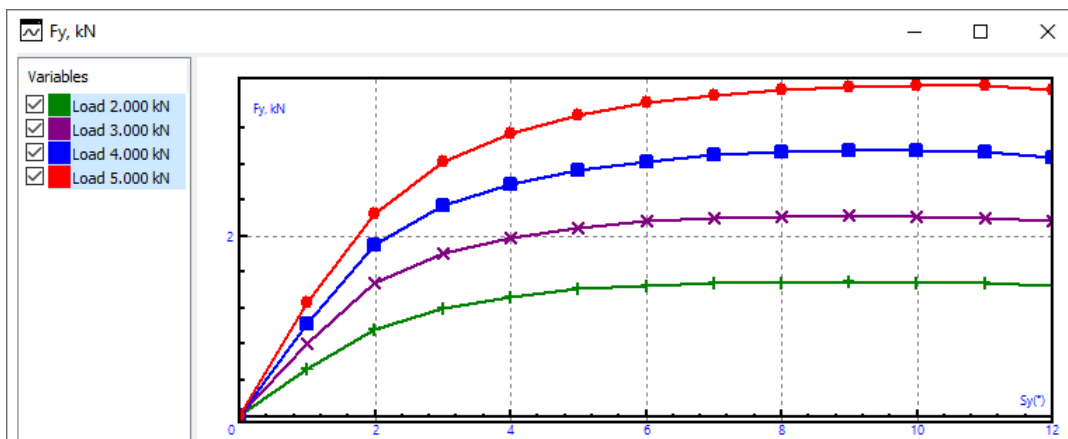


Figure 33.120. Example: lateral force vs. slip

Lateral force, kN – dependence of the tire-road interaction force F_y (kN) on the lateral slip angle Figure 33.120.

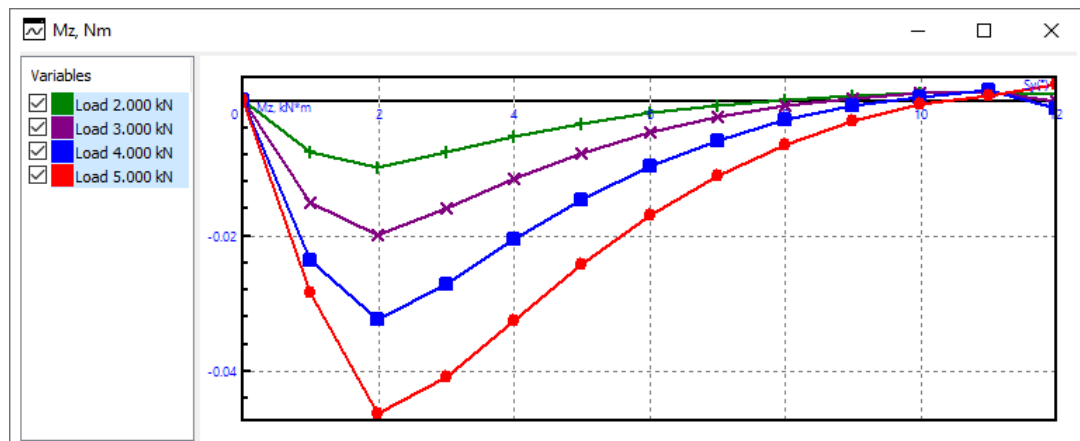


Figure 33.121. Example: aligning moment vs. lateral slip

Aligning moment, Nm – dependence of aligning moment M_z on lateral slip s_y (°), Figure 33.121.

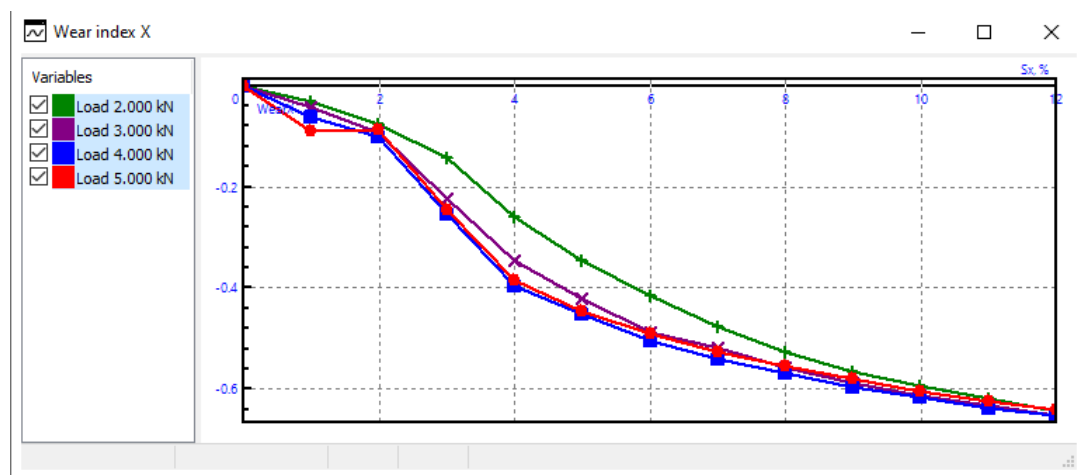


Figure 33.122. Wear index w_x vs. slip s_x

Wear index X – dependence of wear index w_x on longitudinal slip s_x , Figure 33.120.

The unitless wear index is computed according to the formula

$$w_x = \frac{N_{fr}}{Nv(s_x/100)},$$

where N_{fr} is the power of friction forces in contact between the tire and the road (i.e. only sliding friction forces are taken into account, the adhesion friction forces do not perform work), N is the load on the wheel, v is the longitudinal speed.

Wear index Y – dependence of wear index w_y on longitudinal slip s_y , Figure 33.123.

The unitless wear index is computed according to the formula

$$w_y = \frac{N_{fr}}{Nv(s_y 2\pi/180)}.$$

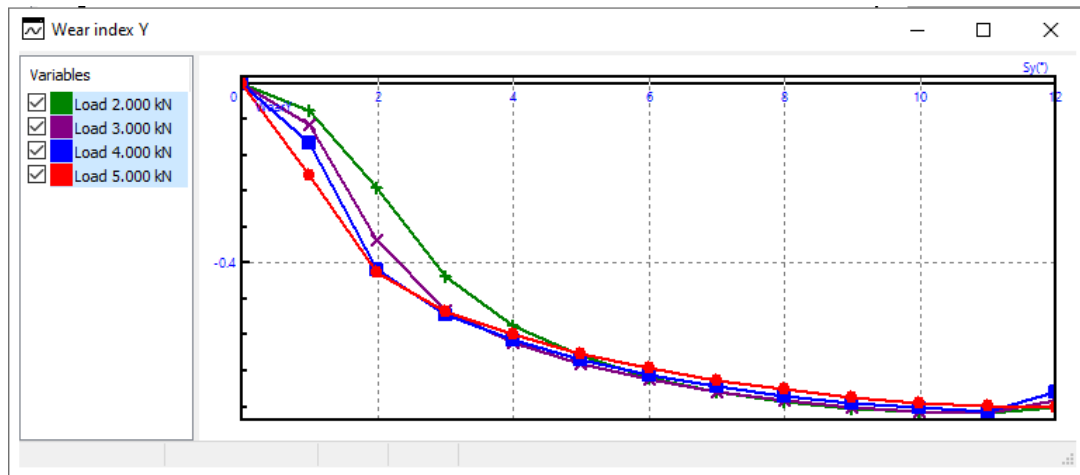


Figure 33.123. Wear index w_y vs. slip s_y

33.3.15. Creation of simplified massless tire models based on test results

33.3.15.1. General information

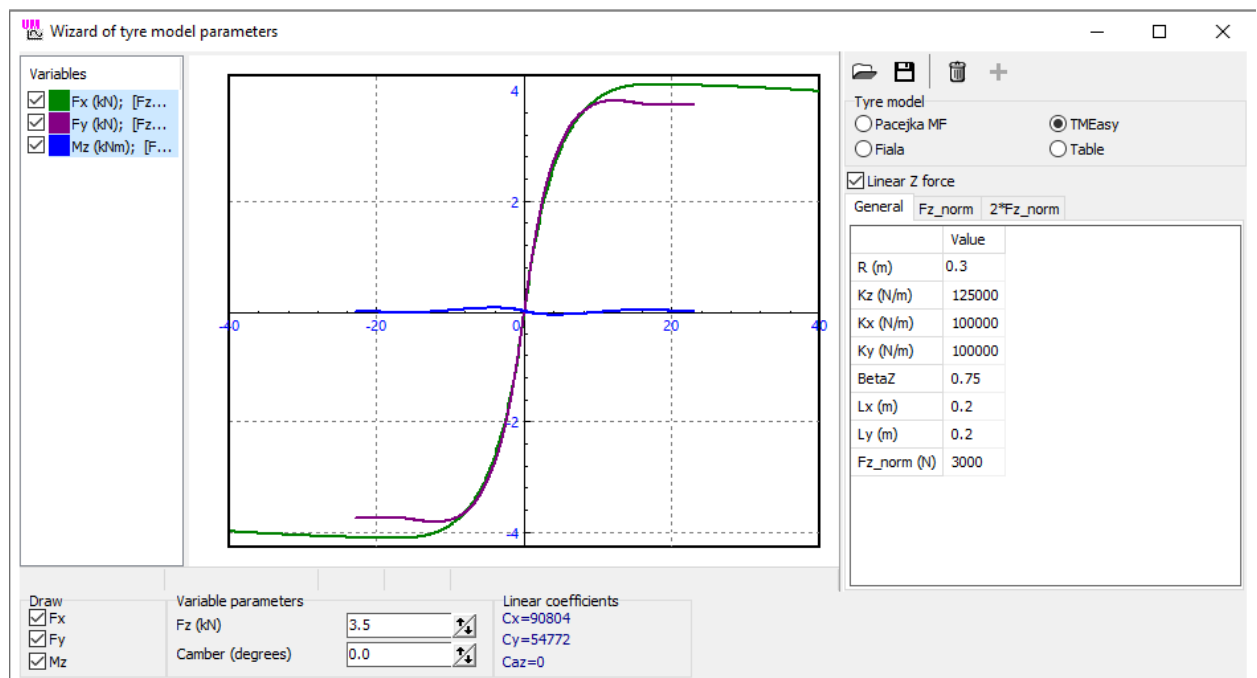


Figure 33.124. Tool for setting parameters of massless tire models

The creation of simplified tire models is one of the important results of performing the series of tests described in Sect. 33.3.14. *Test rig mode: automatic study of tire properties*. Several models of this type are implemented in UM: Pacejka MF, FIALA, TMEasy, Tabular, Figure 33.124, [Chapter 12](#). The simplified tire model is very fast, so it is almost always used in the analysis of vehicle dynamics, which requires performing a large number of computer experiments. The disadvantage of the simplified model is the difficulty of obtaining their parameters, for which test rigs are mainly used. In practice, for example, it is often impossible to quickly obtain such parameters when changing the tire pressure.

The simplified model constructed in accordance with the test results is *consistent* with the corresponding FE tire and it is recommended for use in coupling with the finite element model when studying the dynamics of a wheeled vehicle, Sect. 33.4.4.5 *Consistent models of massless and FE tires*.

The tool considered here allows one to calculate the parameters of simplified models of two types: tabular and TMEasy, Figure 33.125. The basis is the tabular model, and the TMEasy model [7] is of an auxiliary nature and allows, in particular, to use the developed model in other programs for modeling vehicle dynamics, such as Simpack.

33.3.15.2. Process of massless tire model development

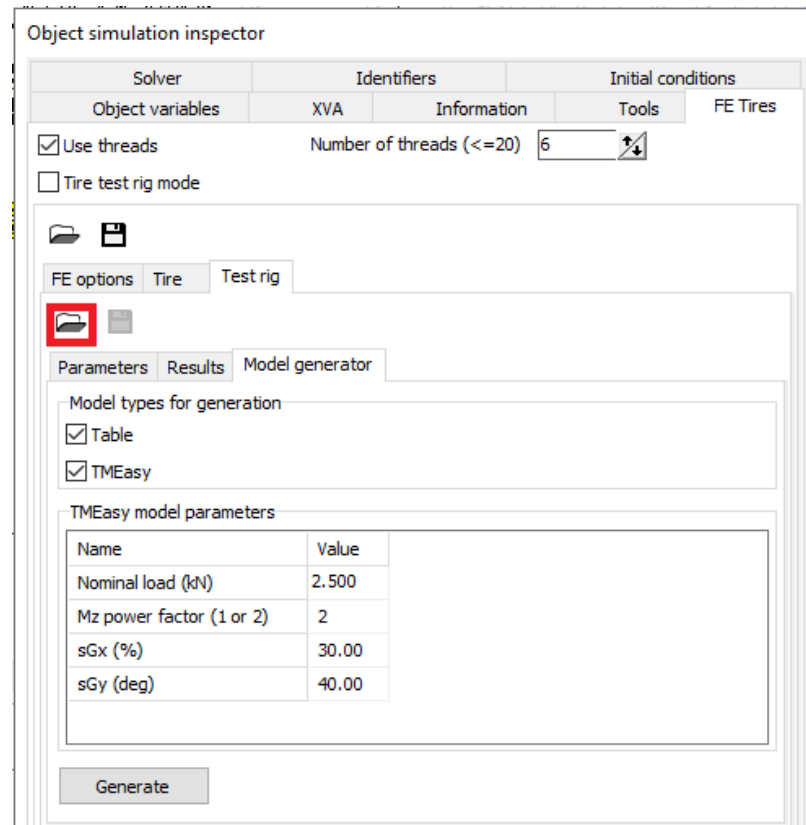



Figure 33.125. Tab of the generator of the massless model of tire. The button for reading the results of tests

The tire model generator in Figure 33.125 is available after completing the calculations in the test rig mode or after reading the results of such a calculation using the button , Sect. 33.3.14 *Test rig mode: automatic study of tire properties.*

To create a model the user should

- specify the type of models to be generated;
- set parameters of TMEasy model generation:
 - rated load, which is also used when generating the tire tabular model;
 - the power factor nM_z in the model of the aligning moment M_z (1 or 2); the original formulation of the model corresponds to $nM_z=1$, and the modification $nM_z=2$;
 - parameters sG_x , sG_y : slip values at which the longitudinal and lateral force models correspond to pure slip;
- click the **Generate** button, select or specify the model file name for *.tr files; for a tabular model, a '[Name].tr' file will be created, and for a TMEasy model, a 'TMEasy [Name].tr' file will be created, where [Name] is the user-specified name; for example, specifying the name Test will create the files Test.tr and TMEasy Test.tr.

33.3.15.3. Example of tabular model

Let us consider a tabular model created on the basis of test results presented in section 33.3.14.4 *Test rig simulation results.* The tabular model uses interpolation polynomials constructed on the basis of pointwise assignment of curves, so a good agreement between the test results and the

tabular model should be expected, Figure 33.126. The smooth curves in the figure were constructed according to the tabular model using the tool presented in Figure 33.124. The markers correspond to the test results with the tire, Sect. 33.3.14.4.

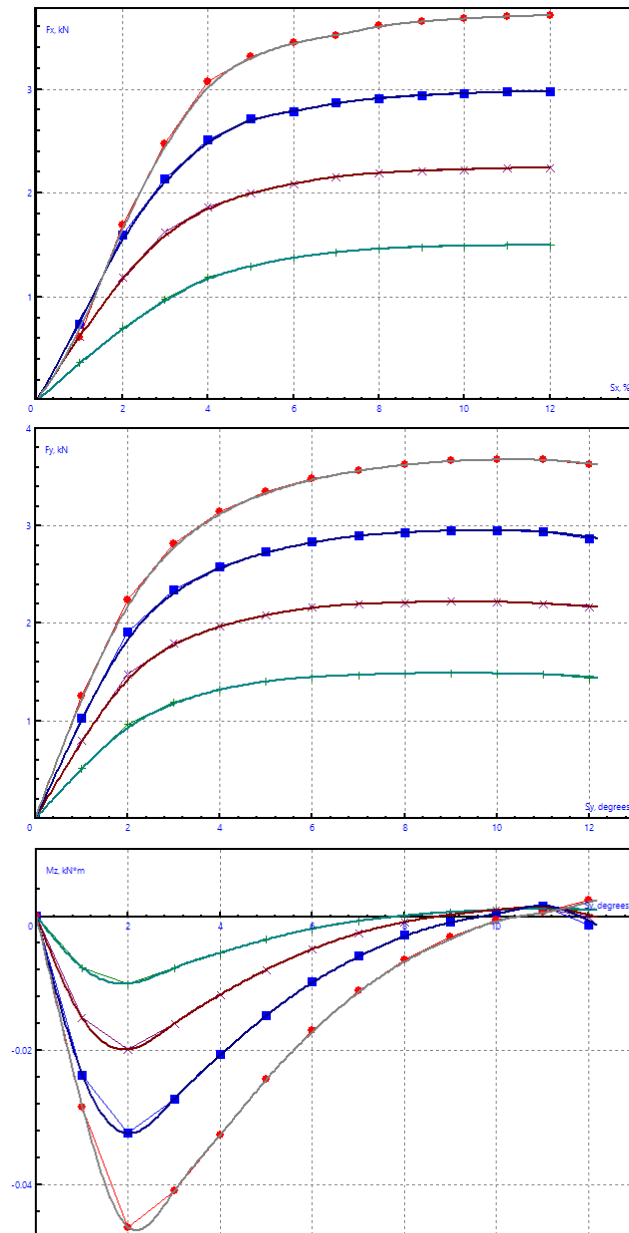


Figure 33.126. Example of comparison of the tabular model with test results

33.3.15.4. Example of TMEasy model

To construct the TMEasy model from the test results, the least squares method is used to determine the model parameters. Since the TMEasy model is very simple and depends on a small number of parameters, the discrepancy with the test results is much greater than in Figure 33.126. Particularly significant discrepancies are observed in the aligning moment, so we propose a simple modification of TMEasy, which we describe below.

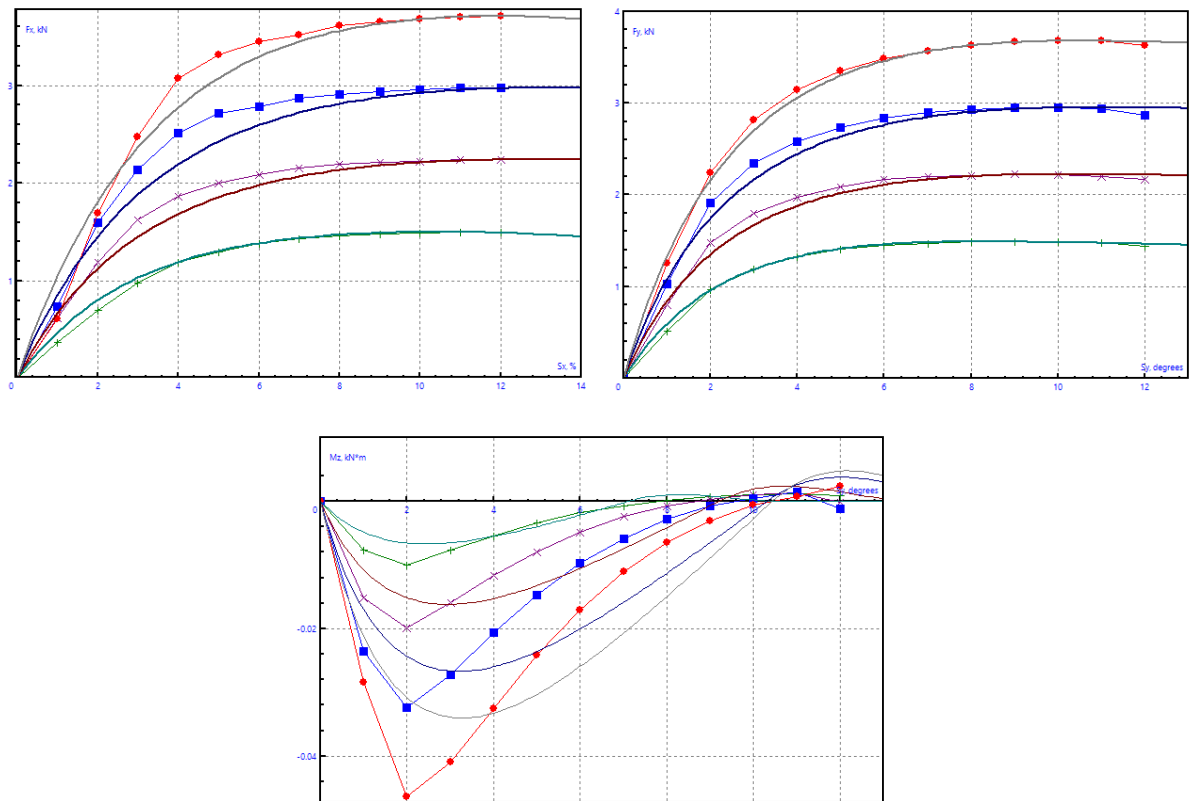


Figure 33.127. Comparison TMEasy with test results for $nMz=1$

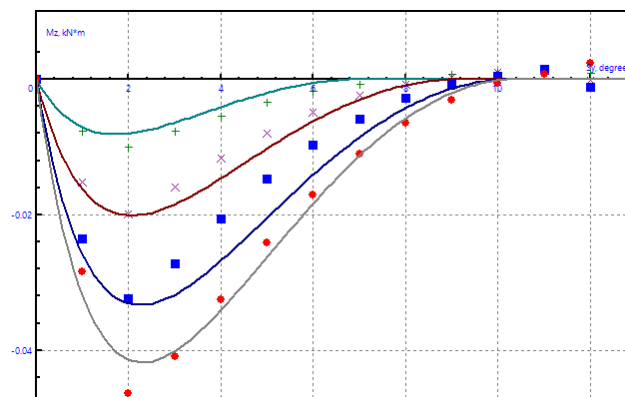


Figure 33.128. Comparison TMEasy with test results for $nMz=2$

Comparison of the unmodified TMEasy model ($nMz=1$) with the test results is shown in Figure 33.127. The best approximation is the lateral force model, which is the most important in modeling the vehicle dynamics. The longitudinal force can also be used, although it has less accuracy. When using the modified model, the results for the moment are much better, Figure 33.128.

As a modification, we introduce a change in the parameter n/L [7]. When $nMz=2$, it is calculated as follows:

$$\frac{n}{L} = \begin{cases} (n/L)_0 (1 - s_y/s_Q^0)^2, & s_y \leq s_Q^0, \\ 0, & s_y > s_Q^0. \end{cases}$$

33.3.16. Verification of FE tire model

Data published by other authors as well as the results of bench rigs are used to verify the developed model of the FE tire.

33.3.16.1. Comparison of static tests with experiment

The author in theses [8] presents the results of static tests of the Dunlop P195-65R15 tire: the dependence of deflection on load. We developed a tire model taking into account the layer-by-layer description of the material presented in the dissertation. A comparison of the numerical calculation of tire deflection from load with the experiment is shown in Figure 33.129. The marker corresponds to the experiment data, and solid lines draw the calculation results for different variants of modeling a multilayer material.

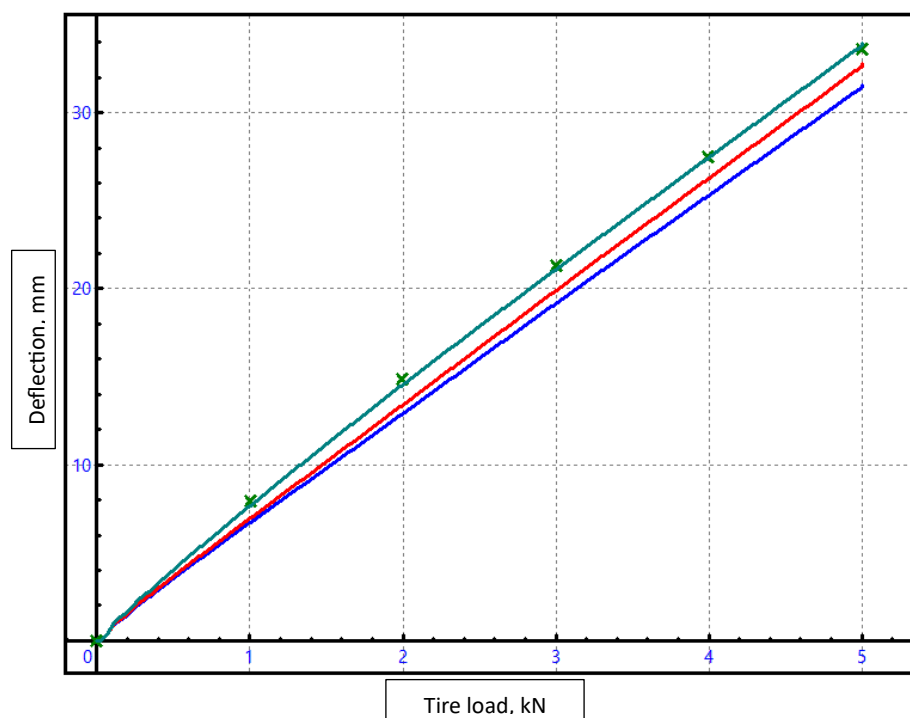


Figure 33.129. Comparison of static experiment for Dunlop P195-65R15 tire with calculation results

33.3.16.2. Comparison of calculations with bench tests of the tire

The dependence of the lateral force on the lateral slip angle for the Dunlop P195-70R15 tire under various loads and speeds are obtained on the test rig [9], Figure 33.130. The test results were kindly provided by the Department of Automobile Transport, Irkutsk National Research Technical University, Professor A.I. Fedotov, Figure 33.131. We express our sincere gratitude to Professor Fedotov and all the test rig developers.

A comparison of the calculation results using the FE tire model with the tests is shown in Figure 33.132 and shows a good agreement. The experimental data for different speeds are marked with a marker; the calculation for a speed of 3.6 m/s corresponds to the solid lines.



Figure 33.130. Tyre test rig at Irkutsk National Research Technical University

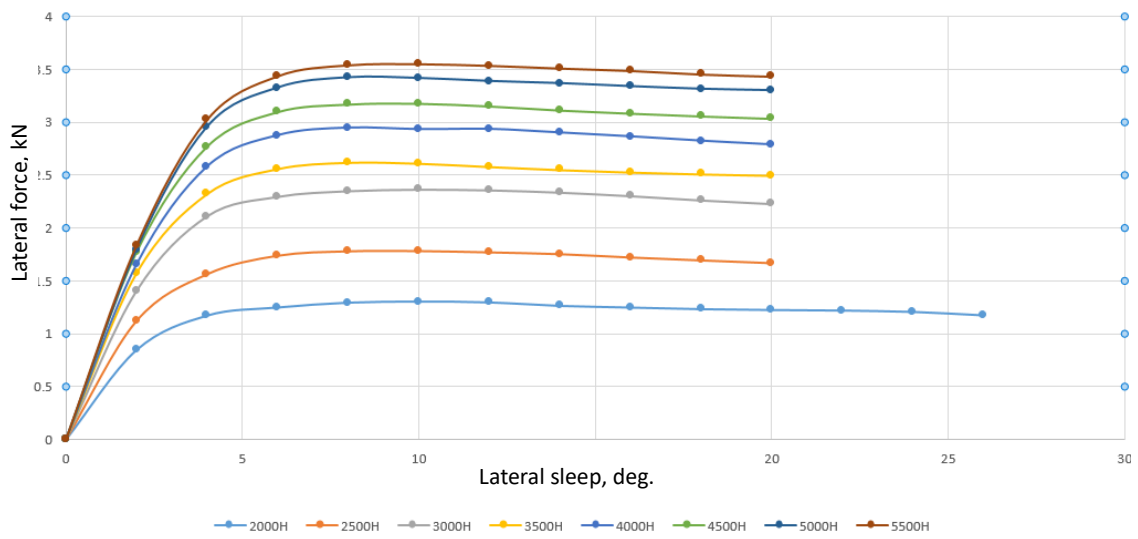


Figure 33.131. Test rig results for speed 3.6 m/c

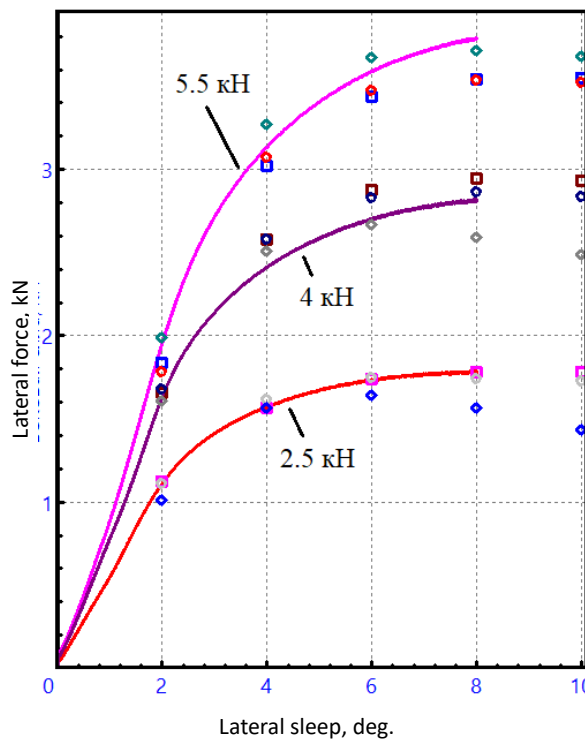


Figure 33.132. Comparison test results with computations for the Dunlop P195-70R15 tire

33.4. Creation and research wheeled vehicles with FE tires

33.4.1. General information on modeling wheeled vehicles with FE tires

Finite element models of tires (FE tires) are added in the UMinput program **to existing models of wheeled vehicles (WV)**, the dynamics of which can be modeled without using FE tires. For example, FE tires can be added to any existing model of a car, monorail vehicle or aircraft landing gear, Sect. 33.4.3 *Adding FE tire to wheeled vehicle models*, Figure 33.133, Figure 33.134, Figure 33.135.

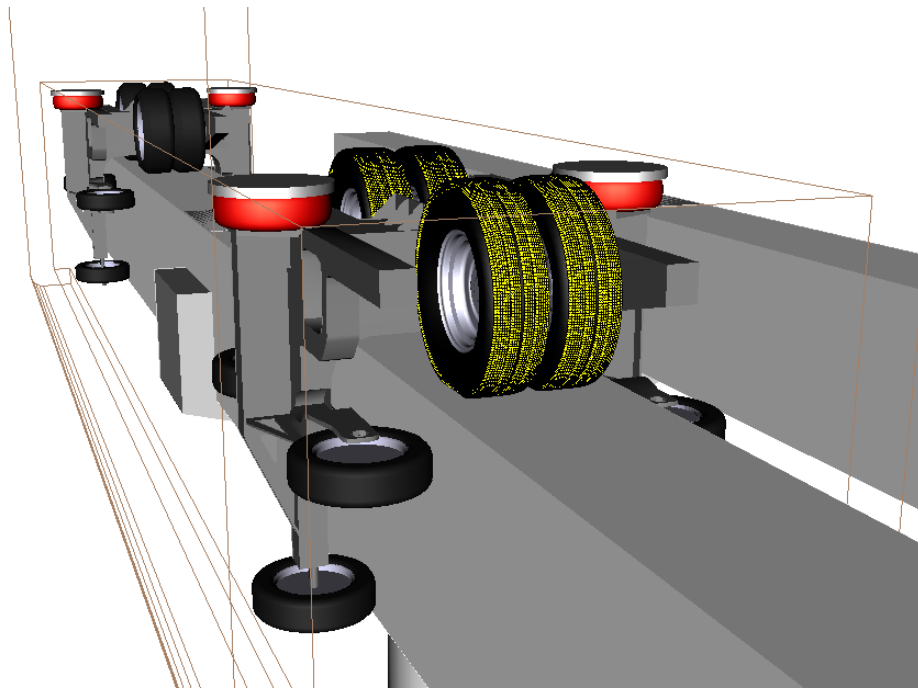


Figure 33.133. Monorail vehicle with FE tires



Figure 33.134. Passenger car with four FE tires

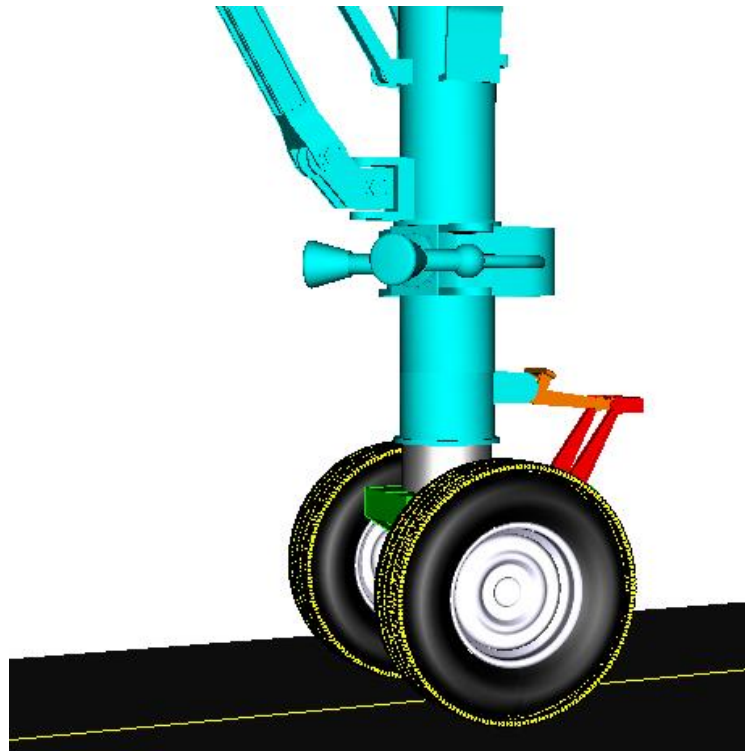


Figure 33.135. Aircraft landing gear

The WV model can include **any number of FE tires**: from one to the total number of wheels in the model. Since the finite element model of FE tires has a large number of degrees of freedom, then simulation of the WV dynamics using them is much slower than modeling the WV without FE tires. Therefore, **the number of FE tires included in the model should be minimized**. Special tools have been developed in UM for this purpose.

- The FE tire can be assigned to a wheel dynamically, i.e. **any FE tire in the simulation program can be assigned to any wheel**. For example, if a model of a passenger car includes two FE tires, then it is possible to switch between the following variants of modeling the dynamics of the car without closing and modifying the model:
 - without FE tires, that is, all wheels use a simplified massless tire model;
 - with one FE tire assigned to any of the four wheels;
 - with two FE tires assigned to any pair of wheels (front, rear, left or right side, diagonally).
- During the simulation, the **FE tires can be dynamically (“on-the-fly”) replaced by the massless tires**. For example, the FE tires can be activated in the section with an obstacle and disabled after passing the obstacle, and then activated again in near a new obstacle.

To add a FE tire to a WV model, the user must develop the desired FE tire and add to a **database** or use an existing FE tire model, Sect. 33.4.2 *Database of FE tires*.

Note. In the case of **monorail vehicles**, the FE tires can only be assigned to traction wheels and cannot be assigned to the guiding or stabilizing wheels, Figure 33.133.

33.4.2. Database of FE tires

To create WV models with FE tires, it is necessary to create a database of FE tire models developed in accordance with Sect. 33.3.12 *Creation and assignment of files for full and reduced FE tire models*. The database is located in the same directory where the models of massless tires *.tr supplied with the UM program are located: {Data UM}\Tire\. For example, it can be the directory


c:\Users\Public\Documents\UM Software Lab\Universal Mechanism\10\Tire\

The database is created as follows.

- In accordance with Sect. 33.3.12 *Creation and assignment of files for full and reduced FE tire models*, the necessary tire models are created for the tire template, including the files *.tire, *.tst, *.tur placed in the corresponding subdirectories of the template.
- The template directory together with all subdirectories is copied to the path {UM data}\Tire\.
- All files are deleted from the template directory copied to the database, except for the files *.tr, *.sbs, *.tire, *.tst, *.tur in the corresponding subdirectories, Sect. 33.3.1.7 *Development of wheel model for database*.

33.4.3. Adding FE tire to wheeled vehicle models

Consider the sequence of steps when adding FE tire to the wheeled vehicle model (WV).

- 1) Load the WV in the Input program.
- 2) Add a FE tire model to the VW by reading it from the *.sbs file using the button  or the **Edit | Read from file** menu command.

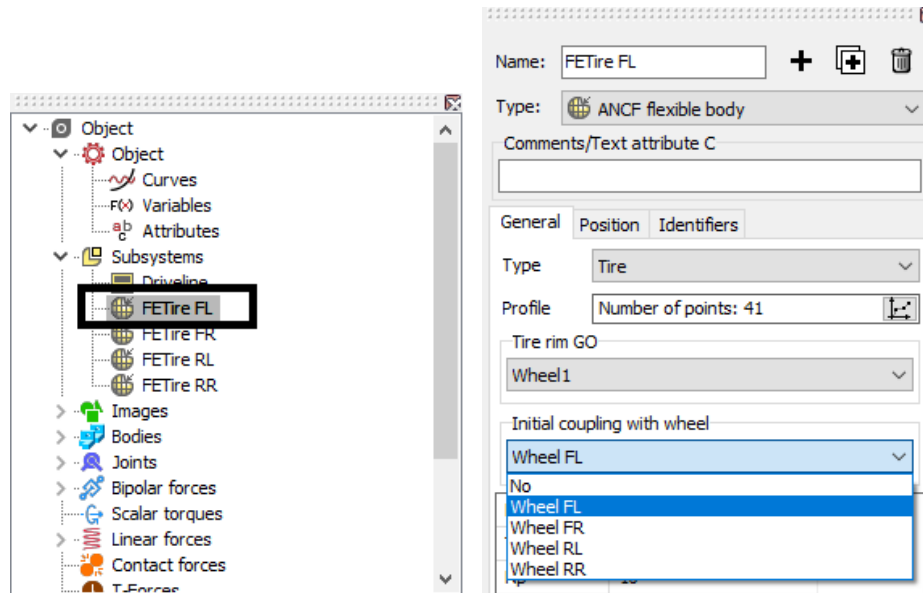


Figure 33.136. Coupling FE tire with wheel

Open the FE tire subsystem tab (Figure 33.136) and set the initial coupling of the FE tire to the rigid body modeling the wheel. To do this, use the drop-down list of wheels in Figure 33.136, right. After setting the coupling, the added wheel is automatically positioned in the desired position, the corresponding shifts and rotations are assigned to the subsystem position,

- 3) Figure 33.137. Editing the coupling coordinates (subsystem position) is blocked.

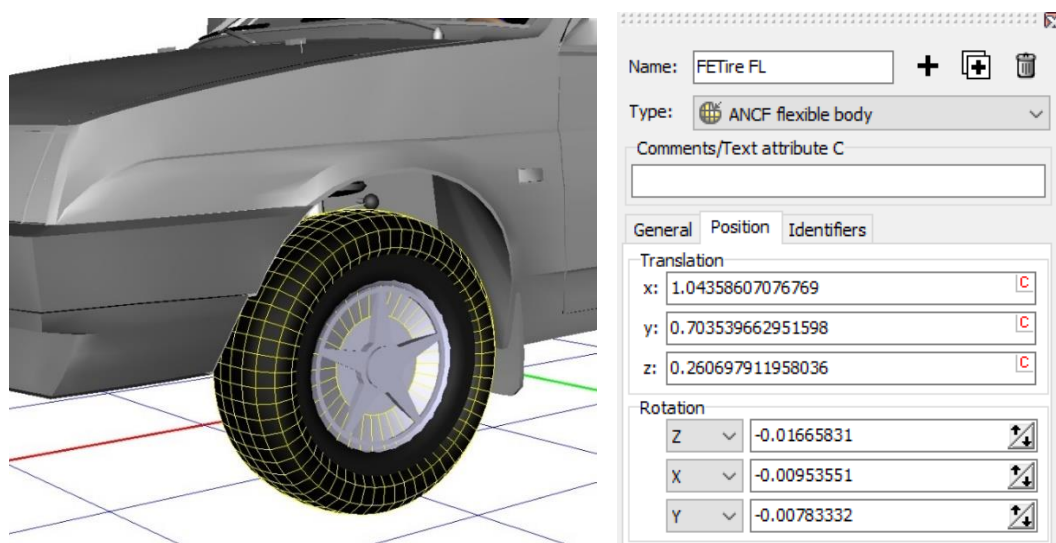


Figure 33.137. FE tire coupled with the front left wheel of car

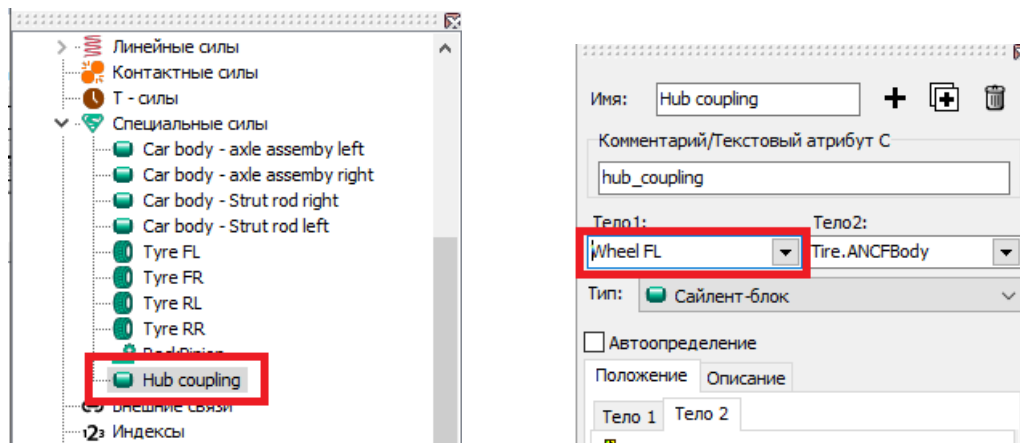


Figure 33.138. Connection of an elastic tire to a wheel using a bushing

- 4) Go to the model of the added bushing **Hub coupling** and assign the coupled wheel as the first body, Figure 33.138.
- 5) If other FE Tires are the same as the first one, copy the FE Tire subsystem and the bushing the required number of times, changing each time the coupling of the FE tire to the wheels as well as the first body in the bushing.

As it is written above (33.3.15.1 General information), the FE tire coupling can be changed arbitrarily in the simulation program.

- 6) If it is necessary to add a different type of FE tire (for example, in the case of a tractor), then repeat steps 2) - 4). When adding different types of FE tire, use different identifiers to describe the parameters in their original description. Identifiers can also be renamed in the WV model, for example, the mass and moments of inertia of the wheel disk *mrim*, *irimx*, *irimy*.
- 7) Remove identifiers from the list that parameterize the FE mesh, for example, *nalpha*, *np_tread*. Changing the FE tire mesh in the WV model is not provided.
- 8) Save the model, if necessary, under a different name.

33.4.4. Preparing a model of a wheeled vehicle with FE tires for modeling

33.4.4.1. Assignment of FE tire models from database

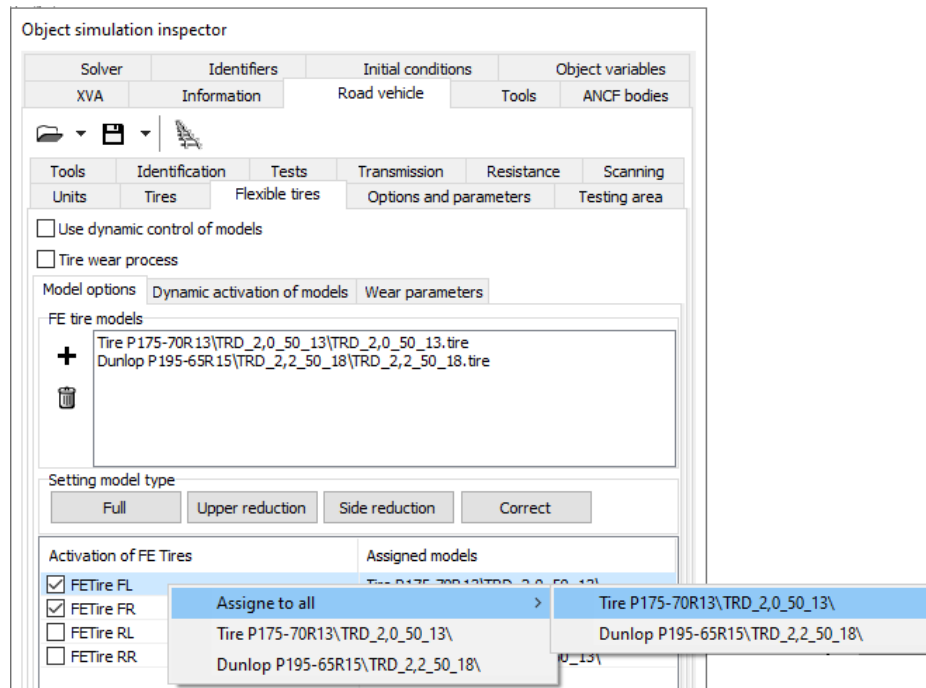


Figure 33.139. List of models and assignment to tires

To work with the FE tire in the simulation program, the user should assign them models from the database. To do this, use the **Road vehicle | Flexible tires** tab of the simulation inspector, Figure 33.139. In the case of a monorail vehicle, the **Monorail vehicle | Flexible tires** tab is used.

Creating a list of FE tire models

The list must contain one or more *.tire models. To add, use the button **+** in Figure 33.139, and in the window that appears select the *.tire file from the database, Sect. 33.4.2 *Database of FE tires*. The list should include tires that will be used in modeling the dynamics of the WV.

Assignment FE tire models

To assign models from the list to the FE tire subsystem, right-click on the FE tire and assign a model using the pop-up menu that appears, Figure 33.139. The user can assign models to all FE tire subsystems in one step using the **Assign to all** command.

33.4.4.2. Activation of FE tires

The FE tires with assigned models can be activated. In this case the simplified massless tire model is disabled and the FE model is enabled. Active models in the list are marked with check marks, Figure 33.140-Figure 33.142.

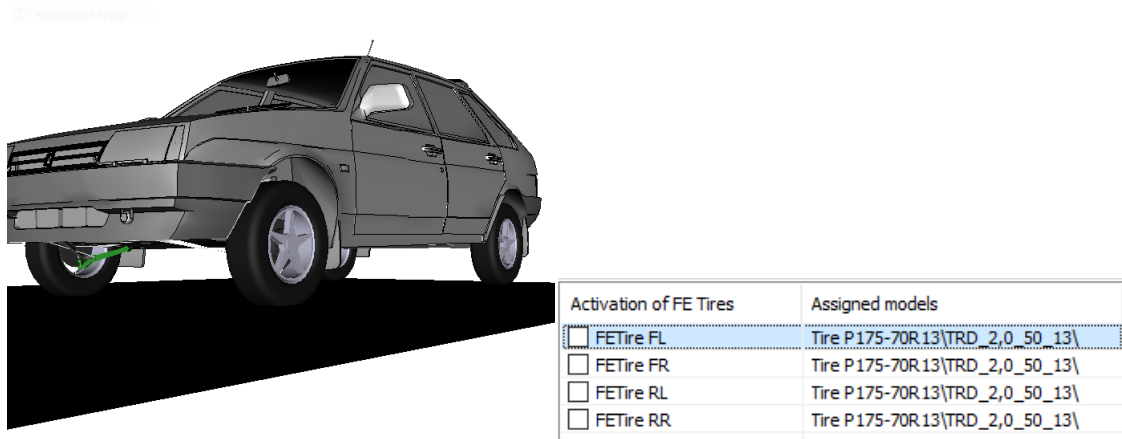


Figure 33.140. All FE tires disabled

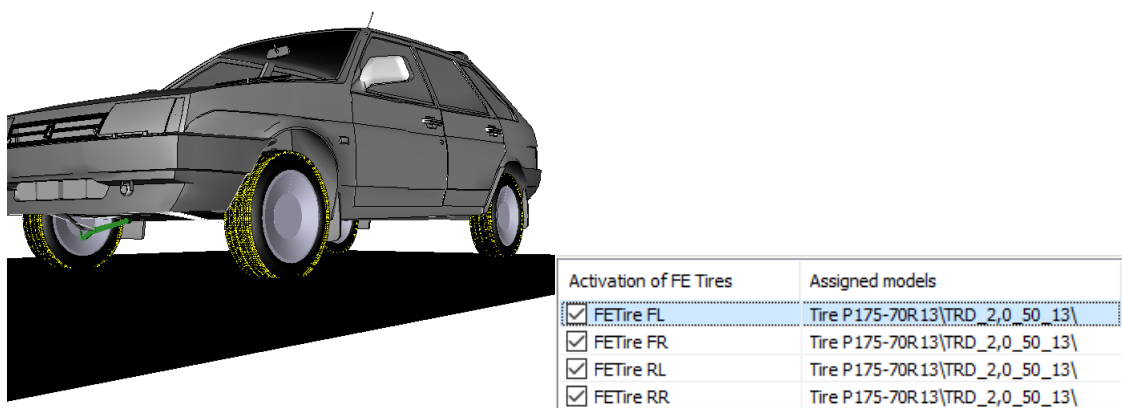


Figure 33.141. All FE tires enabled

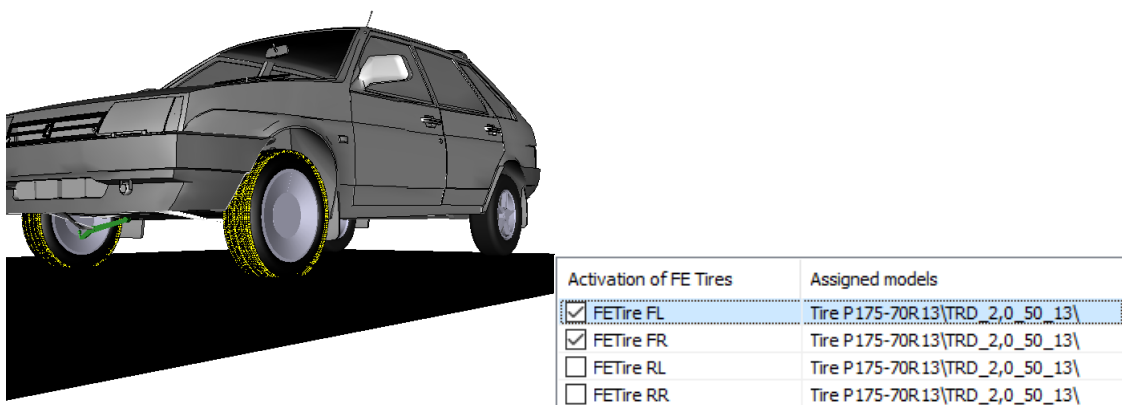


Figure 33.142. Front FE tires enabled

Disabling the FE tires speeds up the simulation since the corresponding degrees of freedom are excluded from the WV model. This property is used for dynamic switching on/off of the FE tires. For example, FE tire is activated near an obstacle, and disabled after passing it, Sect. 0

Dynamic activation of FE tires.

33.4.4.3. Setting FE tire type of reduction

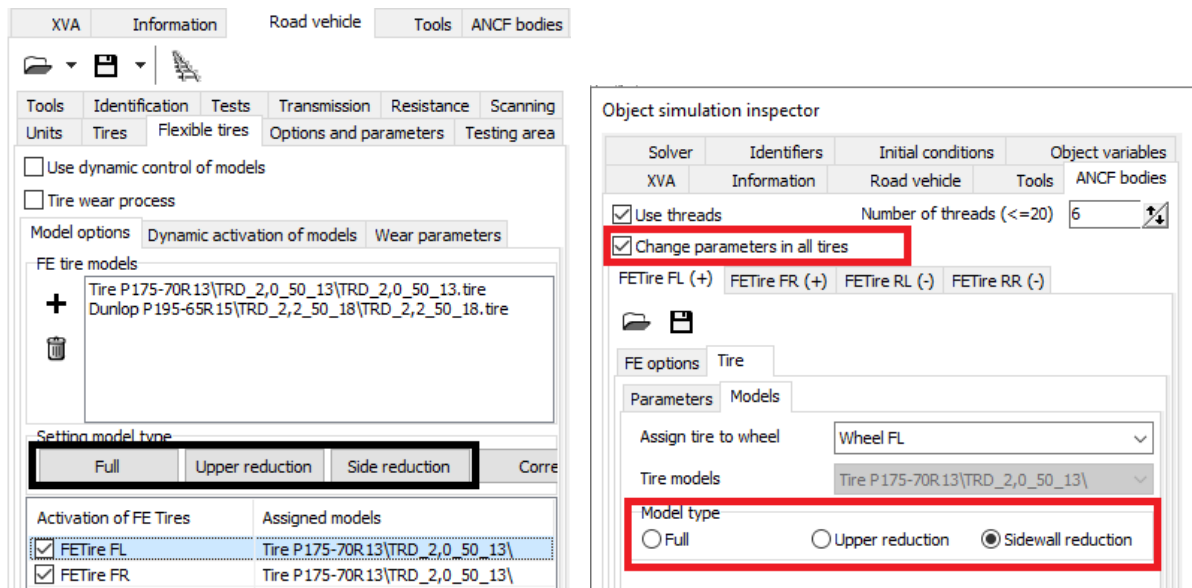


Figure 33.143. Setting FE tire type to all tires

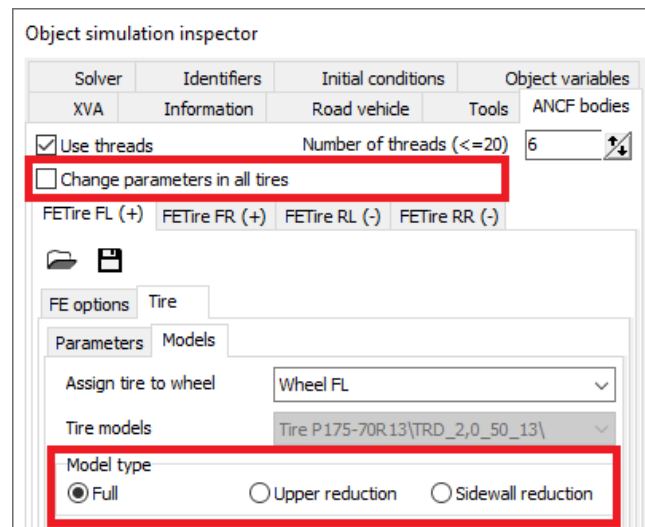
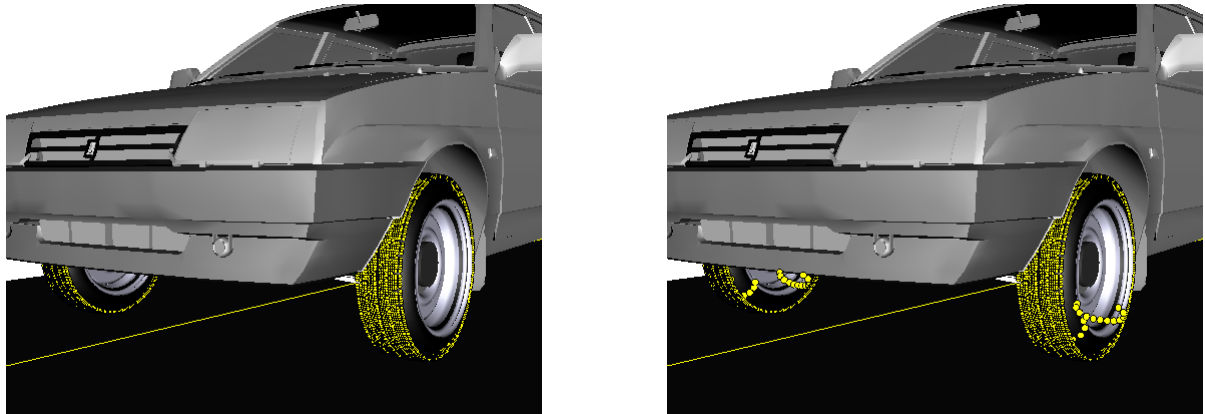


Figure 33.144. Setting FE tire type to one of the tires

If the assigned FE tire model includes one or two reduction options (see Sect. 33.2.7 *Reduced tire models*, 33.3.12 *Creation and assignment of files for full and reduced FE tire models*), the selected option can be assigned to all or some wheels:

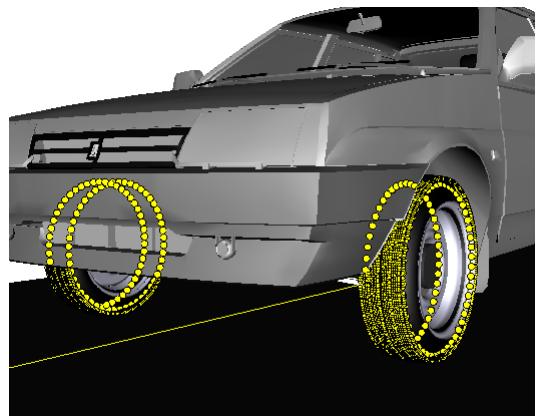
- to set the FE tire type for all wheels, use either the buttons on the **Road vehicle | Flexible Tires | Model options** tab (Figure 33.143, left), or the **Model type** group on the **ANCF Bodies | Tire | Models** tab, provided that the **Change parameters in all tires** is enabled (Figure 33.143, right);
- to set the FE tire type for an individual wheel, use the **Model type** group on the **ANCF Bodies | Tire | Models** tab, provided that the **Change parameters in all tires** is disabled, Figure 33.144.

Outside the process of numerical simulation of the vehicle's motion, the type of assigned models can be seen in the animation window: interface nodes are marked with thick yellow dots, Figure 33.145.



Full

Upper reduction



Sidewall reduction

Figure 33.145. Various options for FE tire models

Note. When a previously inactive FE tire is activated, its type is set to *full*. If necessary, the user should change the type to the desired one.

33.4.4.4. Modification of FE tire parameters

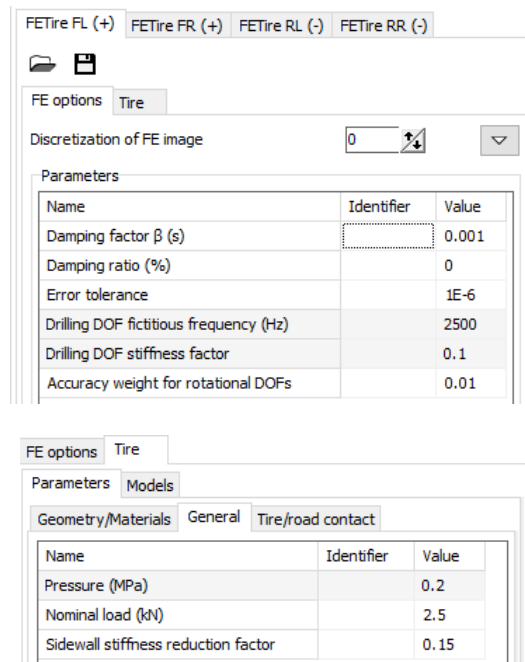


Figure 33.146. Parameters that cannot be changed are highlighted in gray in the tables.

The parameters of the FE tire models are modified in the **ANCF Bodies** tab, Figure 33.147. In comparison with the tire template, some parameters are not available for modification. In the tables, such parameters are highlighted in light gray, Figure 33.146. The description of the tire profile and material is also not available, Sect. 33.3.3.1 *Tire profile and material*. Therefore, the user should generate a set of FE tire models in advance with the required set of parameter values, for example, pressure, Sect. 33.3.12 *Creation and assignment of files for full and reduced FE tire models*.

The values of the tire-road contact parameters (Sect. 33.3.5 *Setting tire/road contact parameters*) and the tread geometry (Sect. 33.3.3.2 *Setting tread geometry*) are available for modification.

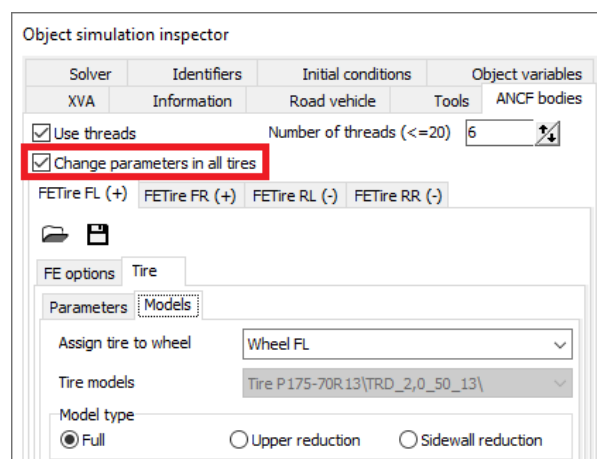


Figure 33.147. Setting the tire model and assigning the tire to the wheel

The parameters can be assigned either to each individual tire or to all tires at once. In the first case, the **Change parameters in all tires** key is disabled, in the second case it is enabled, Figure 33.147.

In the **Tire | Models** tab, the user can change the assignment of tires to wheels as well as set the type of reduced model for both an individual tire and all tires at once, Figure 33.147. See also 33.4.4.3 *Setting FE tire type of reduction*.

33.4.4.5. Consistent models of massless and FE tires

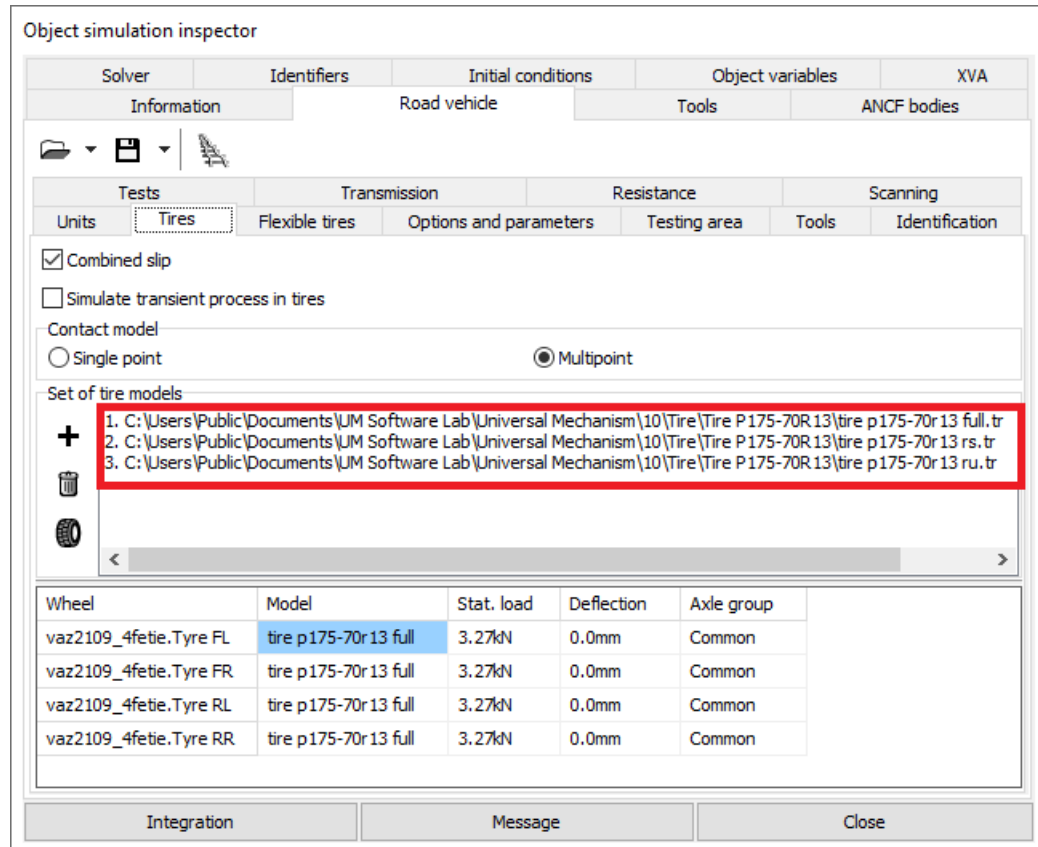


Figure 33.148. Consistent models for full and reduced models of FE tire

It is recommended to use simplified massless tire models consistent with the FE tires (Figure 33.148), that is, models constructed based on the results of virtual tests of an FE tire, Sect. 33.3.15.2 *Process of massless tire model development*. The static and dynamic behavior of the consistent tire is close to the FE tire and allows for correct modeling of the WV dynamics in cases where FE tires are assigned to some wheels, and simplified ones to the rest. It is especially important to use consistent models for dynamic control of the FE tire enabling and disabling, when the FE tire is active only in small sections of the WV motion, Sect. 0

Dynamic activation of FE tires. This approach allows for avoiding significant transient processes when enabling and disabling the FE tire.

Note. Since the simplified model depends on the friction parameters in the contact (e.g. friction coefficients), the massless models should be created for those values that are supposed to be used in the simulation of WV.

33.4.4.6. Initial coordinates in presence of FE tires

Calculation of initial coordinates is an important and necessary part of working with WV in the presence of FE tires. It is recommended to prepare files of initial conditions in advance for each possible configuration of the FE tires in the WV and the tire model (full/reduced), which will allow avoiding transient processes at the initial stage of simulation. Below we consider the main tools for working with initial coordinate values.

33.4.4.6.1. Corrections of FE tire positions when changing WV position

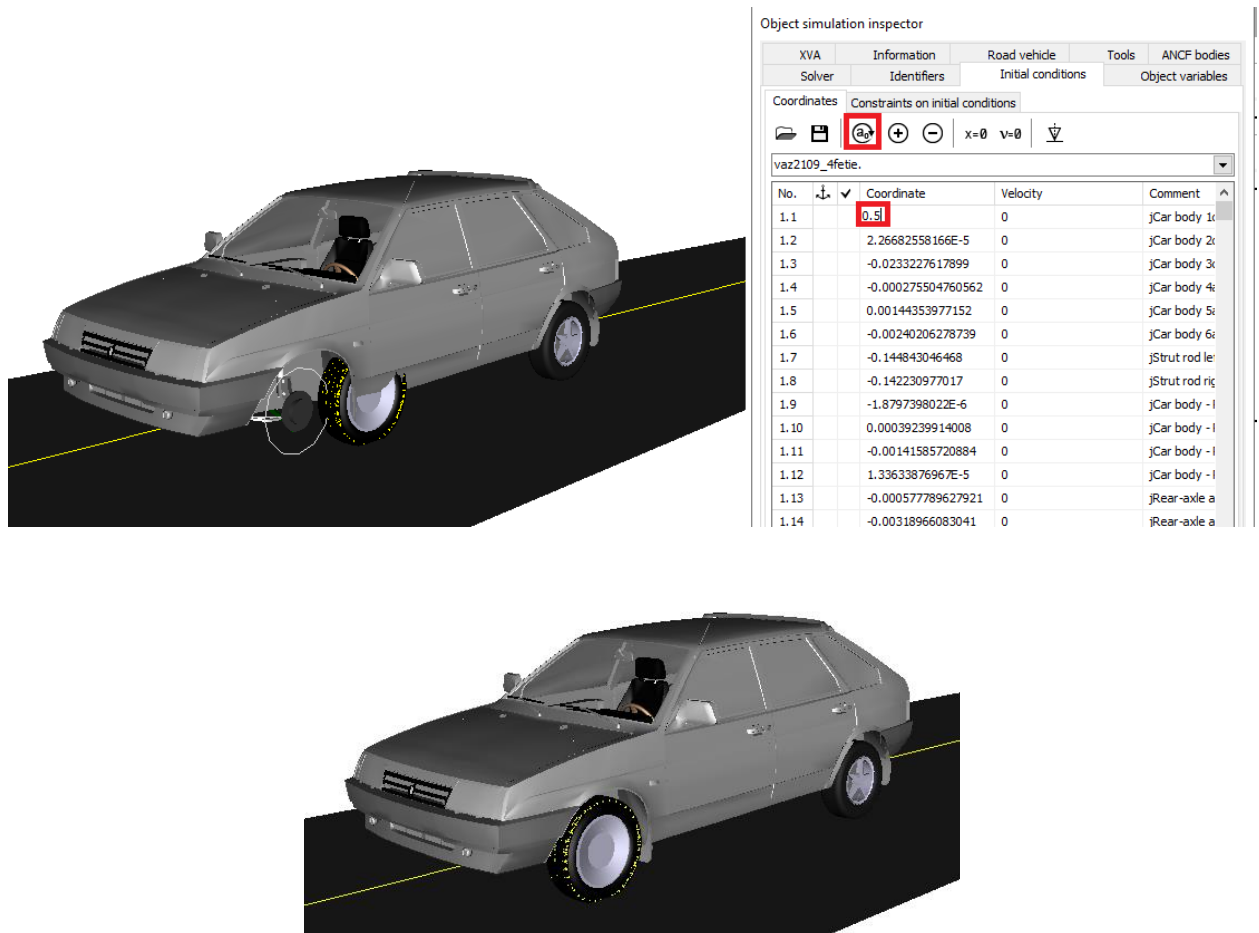



Figure 33.149. Change of WV position and correction of FE tire positions

If a coordinate defining the initial position of the WV on the **Initial conditions** tab is changed, the positions of the FE tires will not change automatically, Figure 33.153, top left. Use the button  to adjust the positions of the FE tires. It is important to note that the adjustment is automatically performed at the start of the simulation, so starting the simulation with shifted WV will not lead to an error.

33.4.4.6.2. Calculation of initial conditions for FE tires as part of the equilibrium test: road vehicles

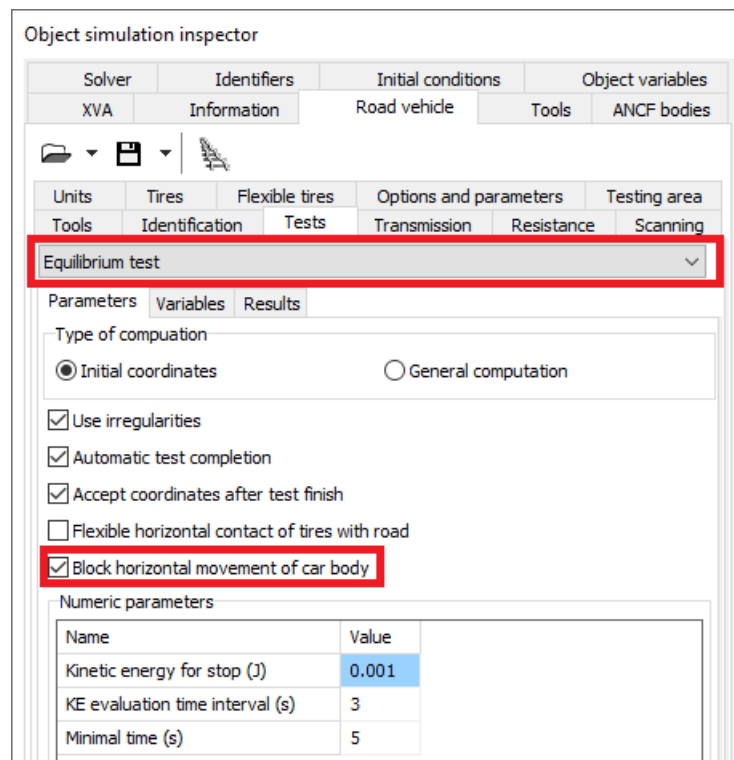


Figure 33.150. Parameters of equilibrium test for a road vehicle

To calculate the coordinates of the FE tires, use the equilibrium test for road vehicles. First, make sure that the contact surface of the tire does not penetrate deeply into the road surface and, if necessary, lift the car in the vertical direction on the **Initial conditions** tab, Sect. 33.4.4.6.1 *Corrections of FE tire positions*.

The following steps should be done for a road vehicle:

- set the test type to **Equilibrium** on the **Road vehicle | Tests** tab, Figure 33.150;
- set the calculation type to **Initial coordinates**;
- enable the **Block horizontal movement of car body**;
- run the simulation;
- wait for the test to complete automatically or, after reaching a state close to equilibrium, switch to pause mode and save the current coordinate values to a file, and also accept them as initial conditions at zero speed, Figure 33.151; the file name must identify the FE tire configuration at a given time.

The calculated initial conditions will be compatible only with the current configuration of the FE tires in the model. When changing the number of active FE tires, as well as when changing the type of the FE tire model (full/reduced), the number of coordinates changes and the file cannot be read as initial conditions.

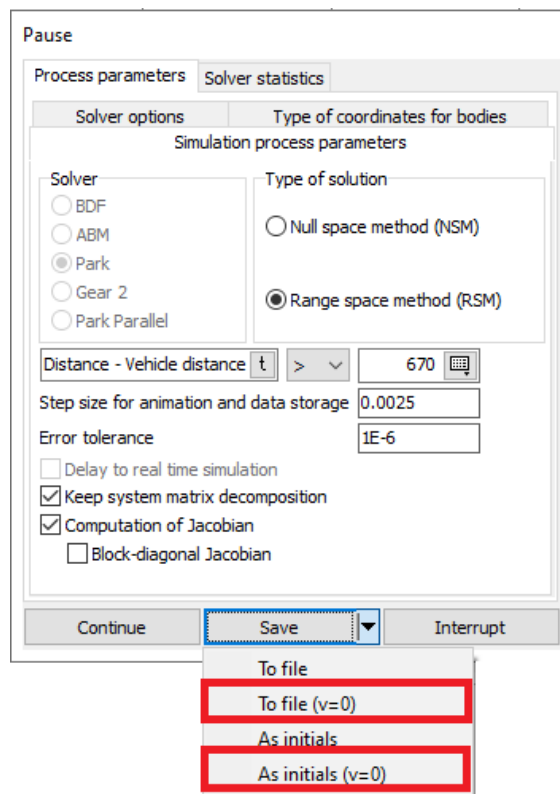


Figure 33.151. Saving current coordinates to file and as initial conditions

- Note 1.** When changing the type of the FE tire model full/reduced, as well as when changing the type of reduction, the number of FE tire coordinates changes, but the program automatically recomputes the coordinates, so that the equilibrium state of the tire is preserved. This means that if the equilibrium is calculated, for example, for the full FE tire model, then by switching the model to a reduced one the state close to the equilibrium is preserved, and the coordinates can be saved in the file for the new state of the model on the **Initial conditions** tab and can be used it in the future.
- Note 2.** When performing an equilibrium test with enabled option **Block horizontal movement of car body**, the friction between the tire tread and the road is automatically disabled.

33.4.4.6.3. Calculation of initial conditions for FE tires as part of the equilibrium test: monorail vehicles

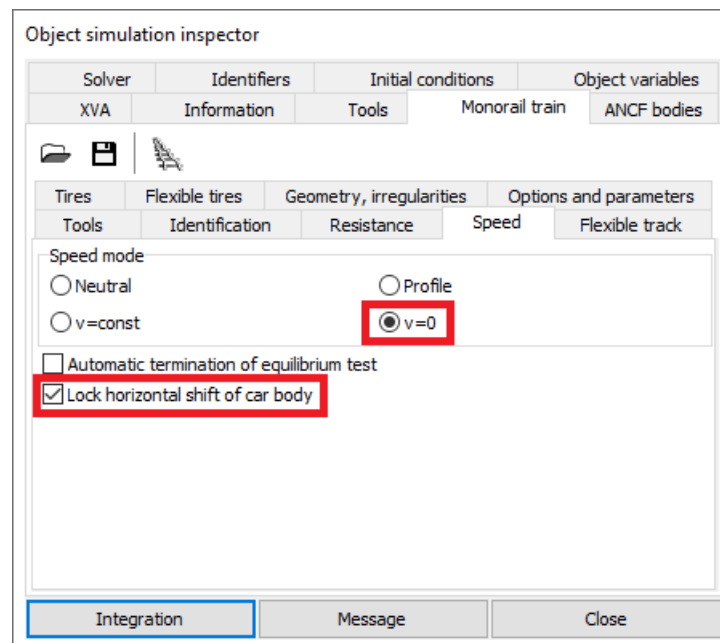


Figure 33.152. Equilibrium computation for monorail vehicle

To calculate the initial conditions of the monorail vehicle with FE tires, the user should set the speed mode $v=0$ and enable the **Lock horizontal shift of car body** option, Figure 33.152. Then the user should follow the method described in the previous section for road vehicles.

33.4.5. Dynamic activation of FE tires

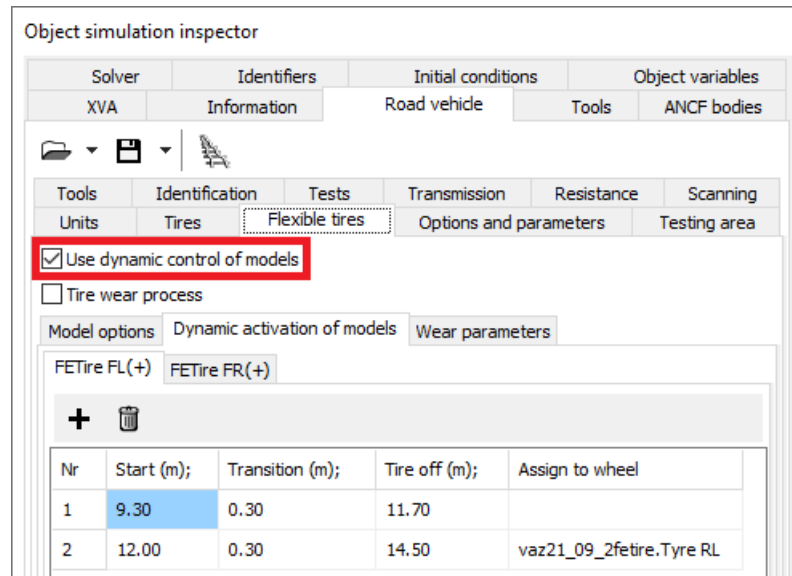


Figure 33.153. Dynamic control mode for activation of FE tires

The dynamic control of FE tire models allows the user to include FE models of tires only in certain sections of the vehicle's movement (FE tire activation sections), which significantly speeds up the vehicle simulation process. If the WV model contains less FE tires than wheels, one FE tire can be dynamically reassigned to another tire during the simulation process. For example, the first FE tire is assigned to the front wheel, and then to the rear wheel (the **Assign to wheel** column in the activation section table, Figure 33.153).

Thus, one or more active sections can be assigned to each of the FE tires, Figure 33.153. **Each section begins with a transition section**, in which the movement of the wheel disk of the FE tire model is determined by the motion of the wheel to which it is attached, and the forces of interaction of the FE tire with the road are calculated. However, the forces acting on the FE tire are not transmitted to the wheel in this section, i.e. the FE tire does not affect the motion of the vehicle. The transition section is used to suppress dynamic transient processes that occur immediately after the activation of the FE tire.

Since the modeling process switches from an massless to a FE tire model at the end of the transition section, it is important to use consistent models., Sect. 33.4.4.5 *Consistent models of massless and FE tires*.

Let us look at the main stages of using this tool.

1) Macrogeometry

Road vehicle

The use of the FE tire in a test with a driver is possible both for the movement of a car along a macrogeometry specified by a 2D curve and along a triangulated surface.

In case of specifying the macrogeometry of a 2D curve, special irregularities (curb, speed bump, road surface damage, etc.) should be specified using the *.trp file, which defines the profile of the test section, [Chapter 12](#), Sect. *Test section profile*, Figure 33.154, Figure 33.155. When driving, road roughness can be taken into account specified by the *.irr files by enabling the corresponding option.

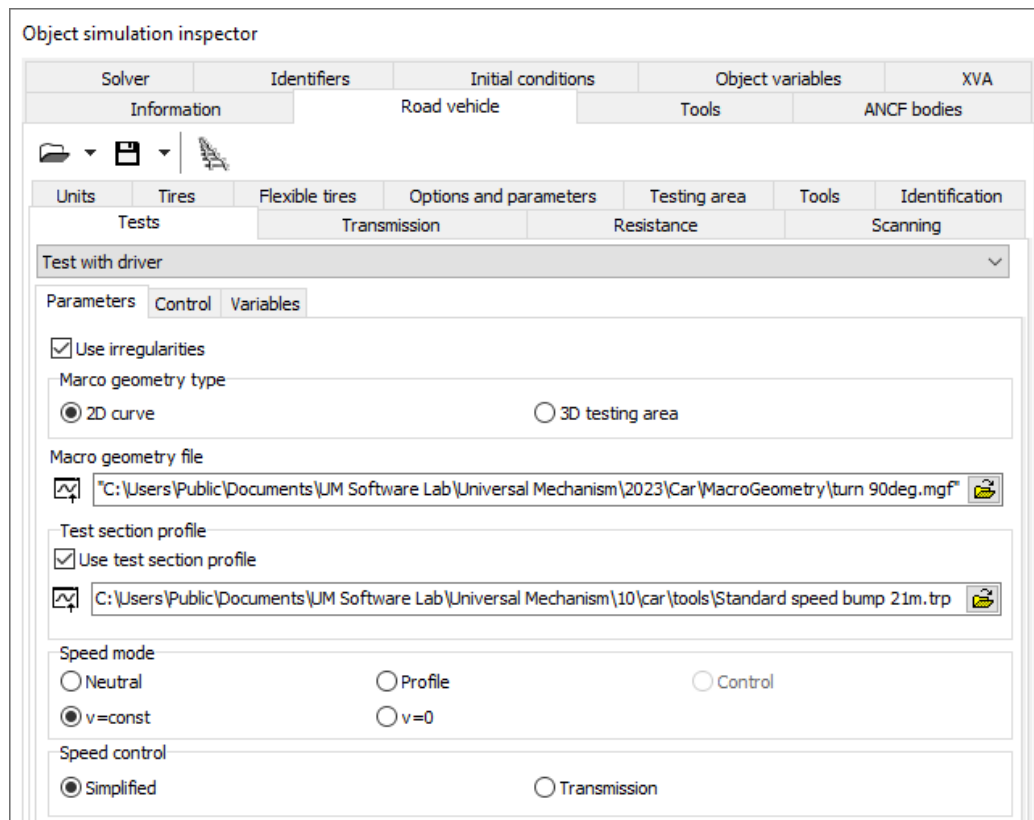


Figure 33.154. Macrogeometry settings when specifying a 2D curve

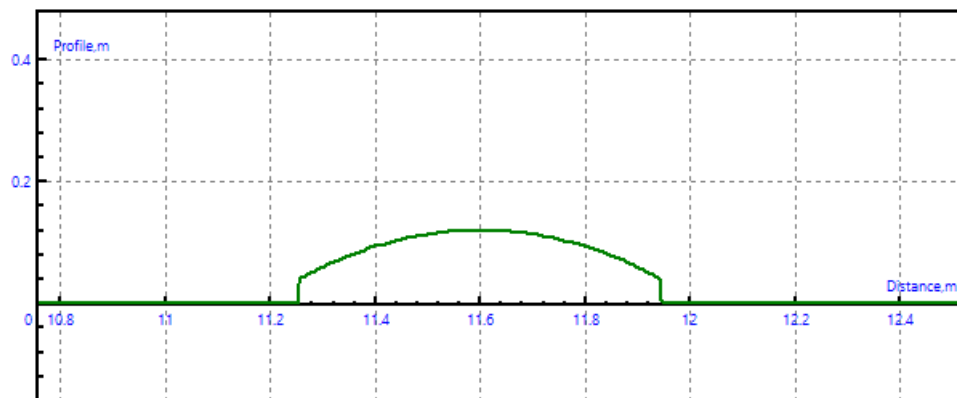


Figure 33.155. Example of speed bump

When modeling the dynamics of a road vehicle, the surface on which it moves can be specified by a **triangulated surface (testing area)**. To do this, load the required surface on the **Testing area** tab, set the corresponding method for specifying the macrogeometry and select a route, Figure 33.156, [Chapter 12](#), *Vehicle movement on a triangulated surface (testing area)*.

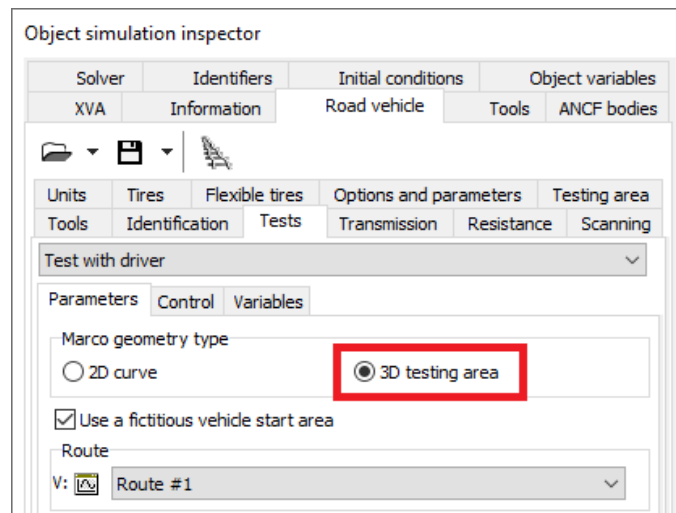


Figure 33.156. Macrogeometry options when specifying a triangulated surface

Monorail vehicle

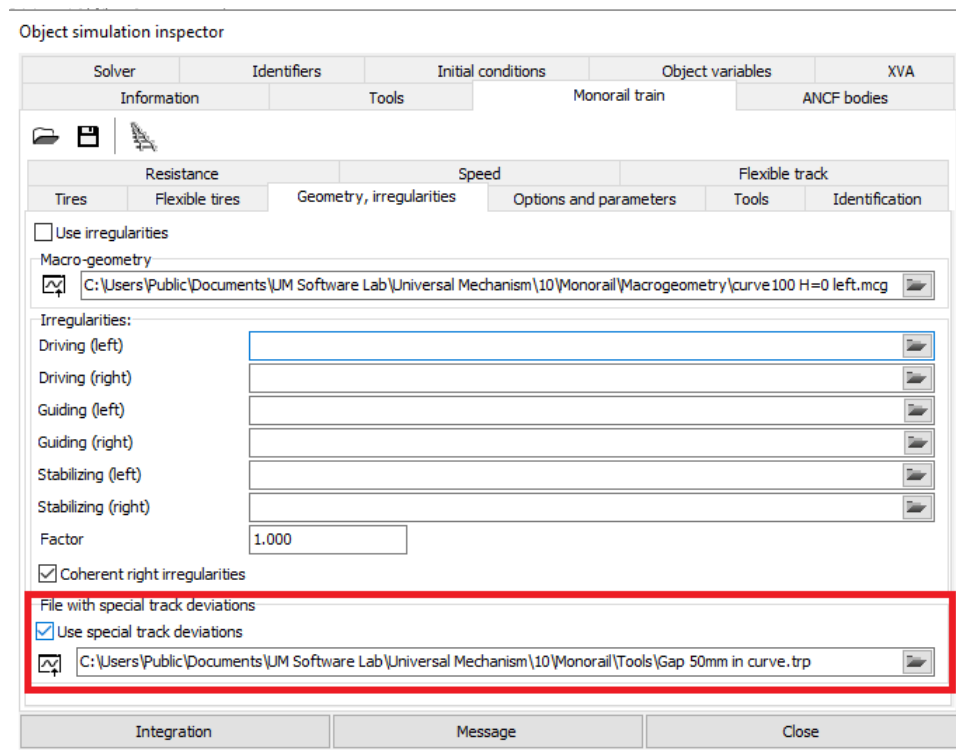


Figure 33.157. Track geometry settings for monorail train

Special track deviations such as step or gap should be specified using the *.trp file, [Chapter 12](#), Sect. *Test section profile*, Figure 33.157. Track roughness can be taken into account specified by the *.irr files by enabling the corresponding option.

2) Test simulation with simplified wheels to select the positions of the FE tire activation sections

In order to define sections where the FE tires are activated, it is recommended to perform a test simulation of the WV along the given macrogeometry. As an example, let us consider the drive of

a passenger car over a speed bump. Let us plot graphs of the vertical force for the right wheels of the car depending on the distance traveled. The run over the speed bump is clearly visible in the plots. First, the front wheel runs over (the distance traveled is 10.2 m), and then the rear wheel (the distance is 12.6 m), Figure 33.158.

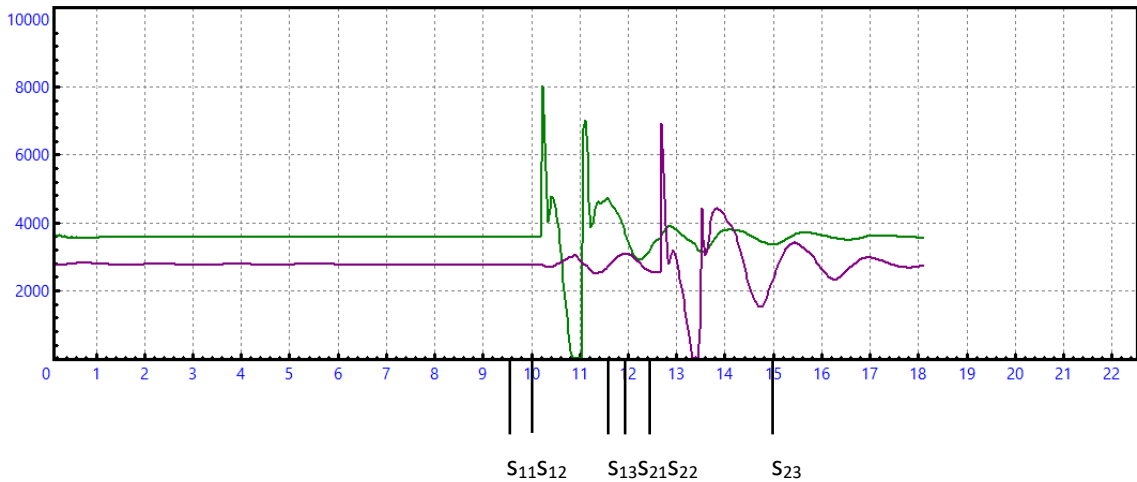


Figure 33.158. Selecting positions for setting the FE tire activation sections

Based on this graph, the following positions for the FE tire activation sections were selected.

Section 1 (front right wheel): $s_{11} = 9.5$ m (enabling), $s_{12} = 10.5$ m (end of transition section), $s_{13} = 11.5$ m (disabling).

Section 2 (rear right wheel): $s_{21} = 11.9$ m (enabling), $s_{22} = 12.4$ m (end of transition section), $s_{23} = 15$ m (disabling).

3) Assignment of FE tire activation sections

After selecting the positions for the FE tire activation sections, the section table for each tire should be filled in.

If a FE tire should not be activated in a given simulation, then one section with an activation position outside the simulation interval is specified for it, Figure 33.159.

An example of specifying two sections for FE tires moving over a speed bump is shown in Figure 33.153, Figure 33.160. At the start of the movement, the speed hump is assigned to the front right wheel, so it passes the first activation section as this wheel. However, in the second section, the FE tire is reassigned to the rear right wheel, Figure 33.160): to assign, right-click on the rightmost cell of the corresponding row and select the wheel from the pop-up menu.

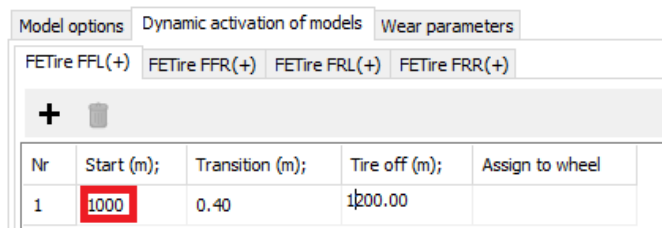


Figure 33.159. Example of setting a section to disable the FE tire throughout the entire simulation interval

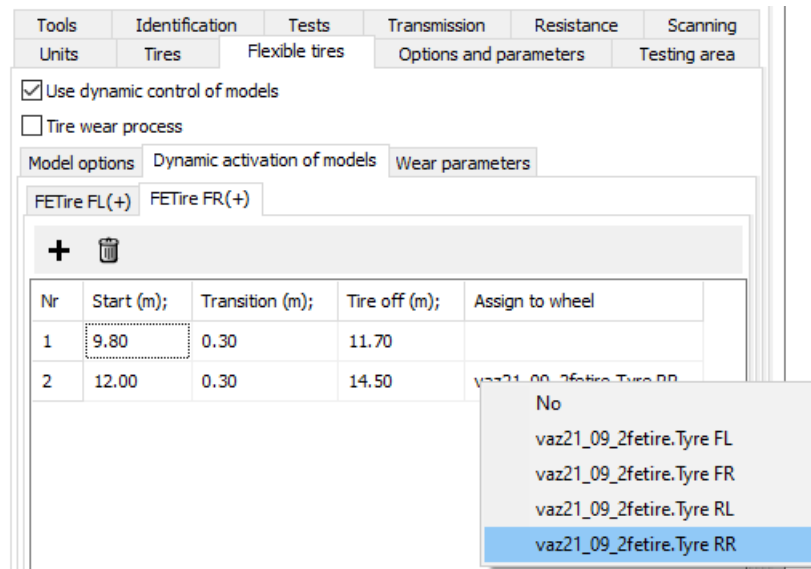


Figure 33.160. Dynamic reassignment of the FE tire to another wheel on the second activation section

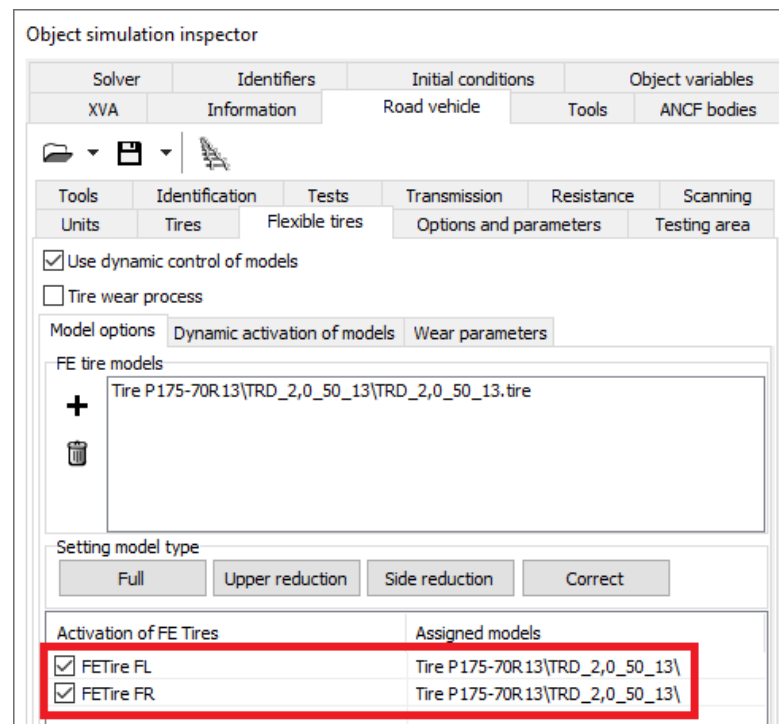


Figure 33.161. Both FE tires are active

Note. If the FE tire is activated during the simulation, it should be active at the start of the simulation, Figure 33.161, and initial conditions corresponding to the equilibrium state of all tires should be set, Sect. 33.4.4.6.2 *Calculation of initial conditions for FE tires as part of the equilibrium test*. This will reduce dynamic transient processes when the tire is activated at the beginning of the transition section.

33.4.6. FE tire variables

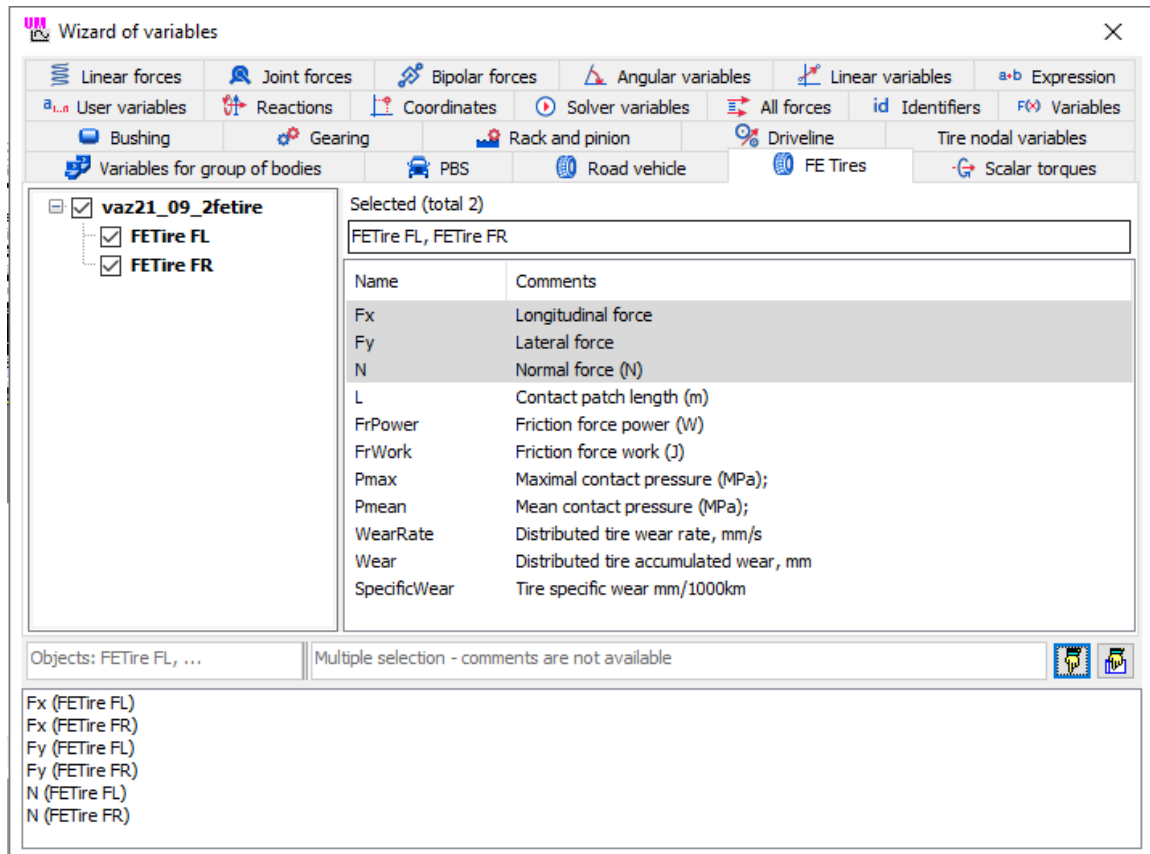



Figure 33.162. Tab with FE tire/Road interaction variables

To create variables, use the Wizard of variables (Figure 33.162), which is accessible when the model is open via the **Tools | Wizard of variables menu** command or via the button  on the toolbar. General recommendations for working with the Wizard can be found in [Chapter 3](#), Sect. *Wizard of variables*.

33.4.6.1. FE tire/Road interaction variables

To create a list of variables characterizing the interaction of the FE tire with the road during the simulation, the FE Tires tab of the Wizard of variables is used, Figure 33.162. The tab allows creating the following variables for each of the FE tires:

Fx, Fy, N – total forces of interaction of the tire with the road in directions;

L – size of the contact patch in the longitudinal direction;

FrPower, FrWork – total power and work of friction forces, only sliding friction is taken into account, adhesion friction does not do any work;

Pmax, Pmin – maximum and average pressure of the tire on the supporting surface;

WearRate, Wear, SpecificWear – tire wear factors calculated by the areas of tire slippage in contact with the road; the second variable characterizes the depth of wear and is the integral over time of the first variable, which determines the wear rate.

33.4.6.2. Kinematics of tire nodes relative to wheel rim

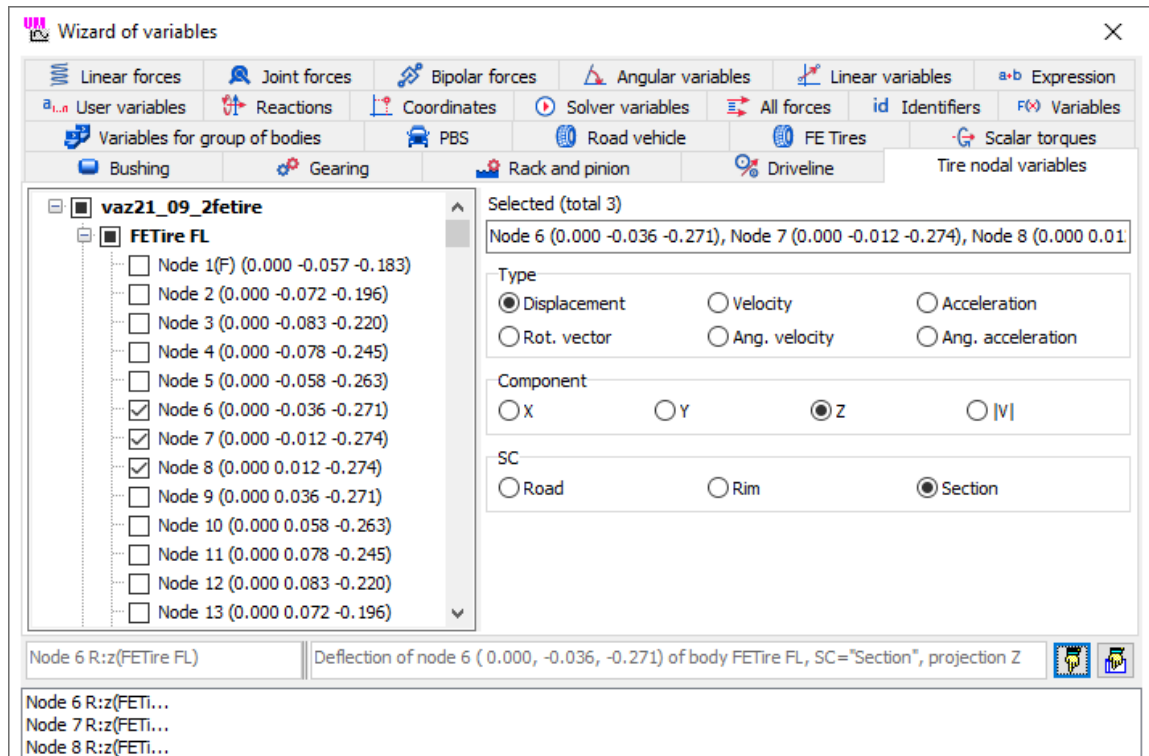


Figure 33.163. Tab with tire node kinematic variables

The **Nodal variables of the tire** tab generates kinematic scalar variables that characterize the motion of nodes relative to the wheel rim due to elastic deformations of the tire, Figure 33.163. The left part of the tab contains a list of nodes of the finite element mesh of the FE tire with the coordinates of the nodes in the undeformed state in the SC of the wheel rim. The letter F (Fixed) marks nodes that are rigidly connected to the rim.

Each scalar variable for a node is a projection of a vector of a certain type onto one of three systems of coordinates (SC), Figure 33.164:

- **Road** – inertial fixed SC of the model (SC0);
- **Rim** – movable SC rigidly connected to the wheel rim with the origin at the wheel center;
- **Section** – movable SC passing through the given node in its undeformed state and the wheel axis; like the previous SC, it is rigidly connected to the wheel rim.

Note that the orientations of the last two SC coincide for the first nodes of the mesh located in the first vertical section of the tire, therefore the variables in these SC coincide for the corresponding nodes, Figure 33.165.

Note. The nodes rigidly connected to the wheel rim are marked with the letter F in the list in Figure 33.163. These nodes are fixed relative to the wheel rim, and all variables for them are equal to zero.

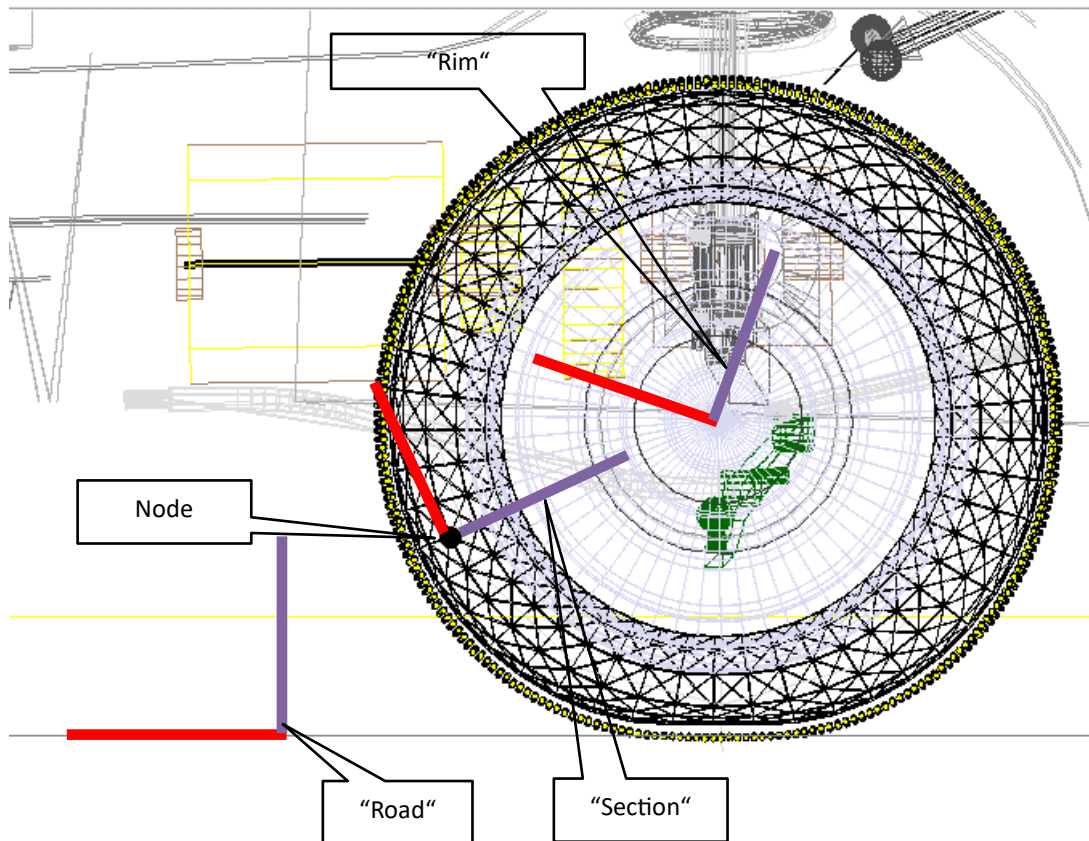


Figure 33.164. Systems of coordinates used to calculate nodal variables

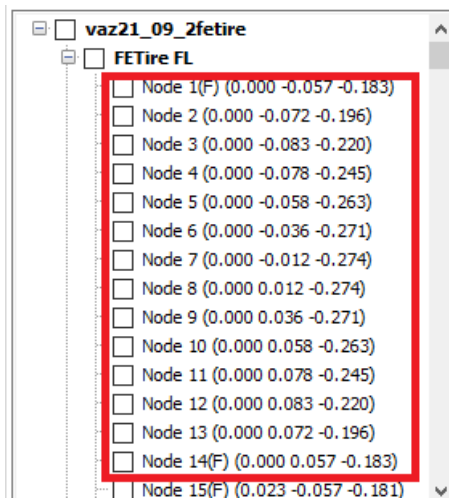


Figure 33.165. Example of nodes for which the wheel rim SC and the Section SC coincide

Consider the types of variables.

- **Displacement** is the vector of elastic displacement of the node relative to the position with an undeformed tire.
- **Rotation vector** is the vector of rotation of the node's SC relative to the undeformed position. For small deviations in the orientation of the node's SC, the projection of the rotation vector is equal to the angle of rotation around the corresponding axis.
- **Velocity, angular velocity, acceleration, angular acceleration** are standard kinematic vectors that determine the motion of the node relative to the wheel disk due to elastic deformations of the tire.

Components:

X, Y, Z are projections of vectors onto the corresponding axis of a given SC;

$|V|$ is the vector modulus; does not depend on the SC.

Figure 33.166 shows an example of the graphs of the time dependence of the projections on the Z axis of all three SCs of the displacement vector of node number 246, located approximately in the middle of the tread. The variables have the same maximum values in the “Road” and “Section” SC when the node passes through the contact with the road.

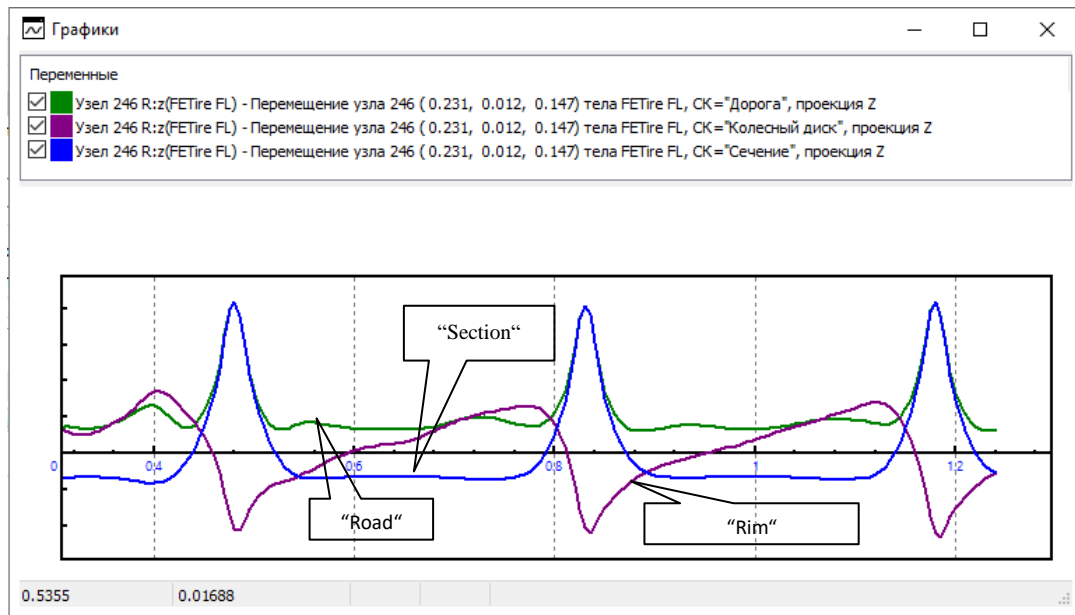


Figure 33.166. Comparison of the graphs of Z component of the node displacement vector for different coordinate systems

- Note 1.** Usually, the most informative are the projections of vectors onto the Section SC, which is offered by default.
- Note 2.** In case of using reduced tire models, variables are also calculated for reduced nodes using formulas (33.8), (33.9) except for linear and angular accelerations, Sect. 33.2.7 *Reduced tire models*.

33.4.7. Simulation of dynamics of vehicles with FE tires

Consider the main stages of modeling a WV with FE tires.

- 1) Activate the required FE tires, Sect. 33.4.4.2 Activation of FE tires.
- 2) If necessary, calculate the initial conditions or load them from a file, 33.4.4.6 Initial coordinates in presence of FE tires.
- 3) When simulation of road vehicle, set up a test with a driver and form the required macrogeometry.
- 4) When using dynamic control of FE tires, check the consistency of tire models, Sect. 33.4.4.5 Consistent models of massless and FE tires.
- 5) Create a list of variables, if necessary, open graphic windows, place variables in them for visual control.
- 6) Run the simulation process.

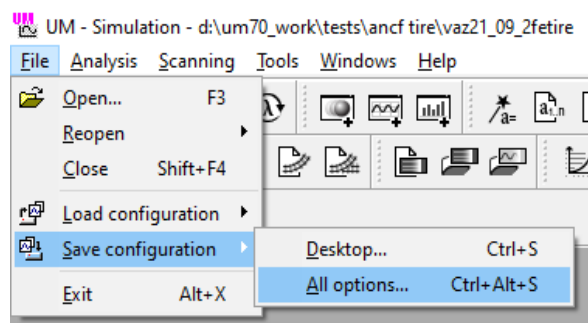


Figure 33.167. Saving full configuration of model

- 7) After finish of the simulation process, if necessary, save the complete model configuration for further use, Figure 33.167.

33.4.8. Modeling tire tread wear

Tyre wear prediction is one of the tasks that can be solved using the FE tire models.

33.4.8.1. Mathematical model of wear

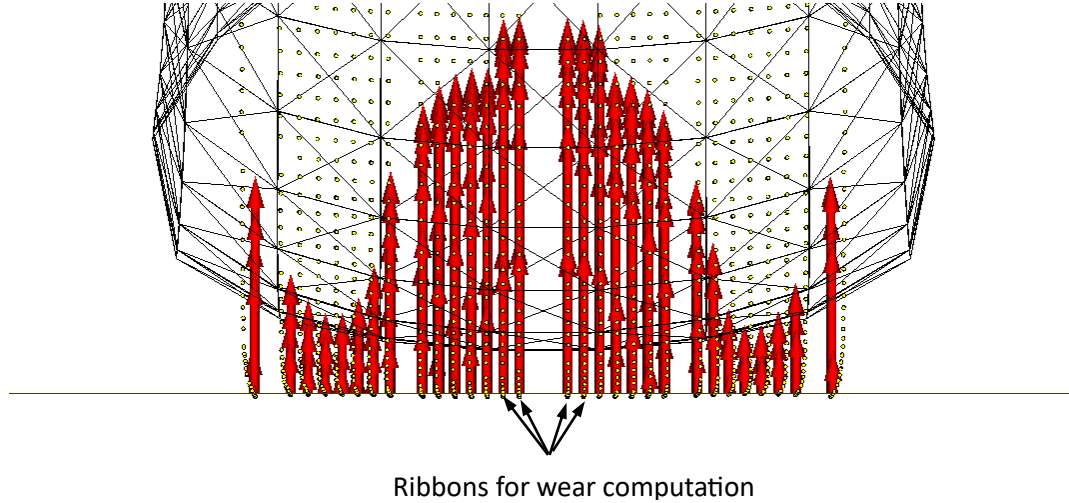


Figure 33.168. Example of ribbons for wear computation and accumulation

The Archard model of tire tread wear is used [8], according to which the wear rate is proportional to the power of sliding friction forces. The contact surface of the tread is divided into ribbons. For an undeformed tire, each ribbon is a narrow cylinder with a diameter equal to the diameter of the wheel, and the cylinder axis is the axis of rotation of the wheel. The contact points of the tread with the road lie on the median circle of an individual ribbon, Figure 33.168. In the example in Figure 33.168, we have 36 stripes, six of which do not contain contact points due to channels in the tread. The number of ribbons and contact points in a ribbon depends on the finite element mesh, the parameters n_x, n_y , which determine the number of contact bristles in the finite element, and the tread pattern, Sect. 33.3.5 *Setting tire/road contact parameters*.

Wear within one ribbon is averaged, i.e. it is considered the same across the circumference and width of the ribbon.

The distributed wear rate factor for ribbon number i is calculated in mm/s using the formula

$$\dot{h}_i = 10^3 \frac{k}{HA_{ci}} \sum_{j=1}^{n_i} F_{ij}^T v_{\tau,ij}, \tag{33.15}$$

where the summation is performed over all bristles in the ribbon, j is the bristle number;

$F_{\tau,ij}$ in the friction force for bristle j (N), and $F_{\tau,ij} = 0$ if the bristle is not in contact;

$v_{\tau,ij}$ is the sliding velocity for bristle j (m/s), and $v_{\tau,ij} = 0$ if the bristle is in the adhesion mode;

the product $F_{ij}^T v_{\tau,ij}$ is equal to the power of the sliding friction force at a separate contact point;

k is the dimensionless wear coefficient; the default value is 10^{-3} , Figure 33.169;

H is the material hardness (Pa), the default value is $2 \times 10^9 \text{Pa} = 2 \text{GPa}$, Figure 33.169;;

A_{ci} is the contact area of the ribbon in the tread minus the grooves and channels;

10^3 is the multiplier that converts velocities into mm/s.

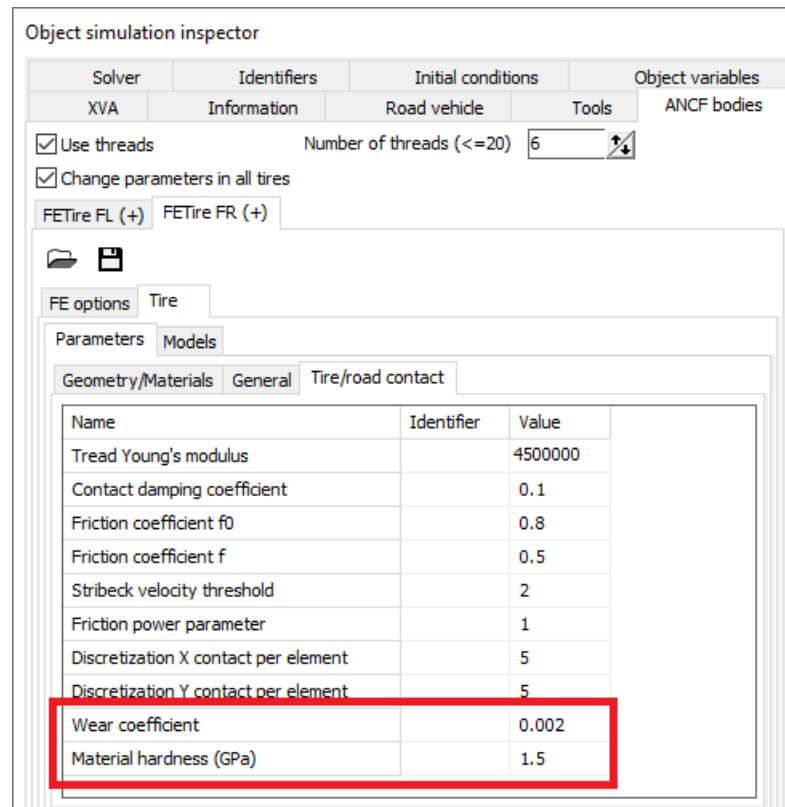


Figure 33.169. Tire wear parameters

The distributed accumulated wear factor determines the wear depth in mm within a single strip

$$h_i = \int \dot{h}_i dt \tag{33.16}$$

Tire specific wear mm/1000km

$$I_h = \frac{10^6 h_i}{L} \tag{33.17}$$

where L is the distance in m.

33.4.8.2. Variables characterizing tire wear

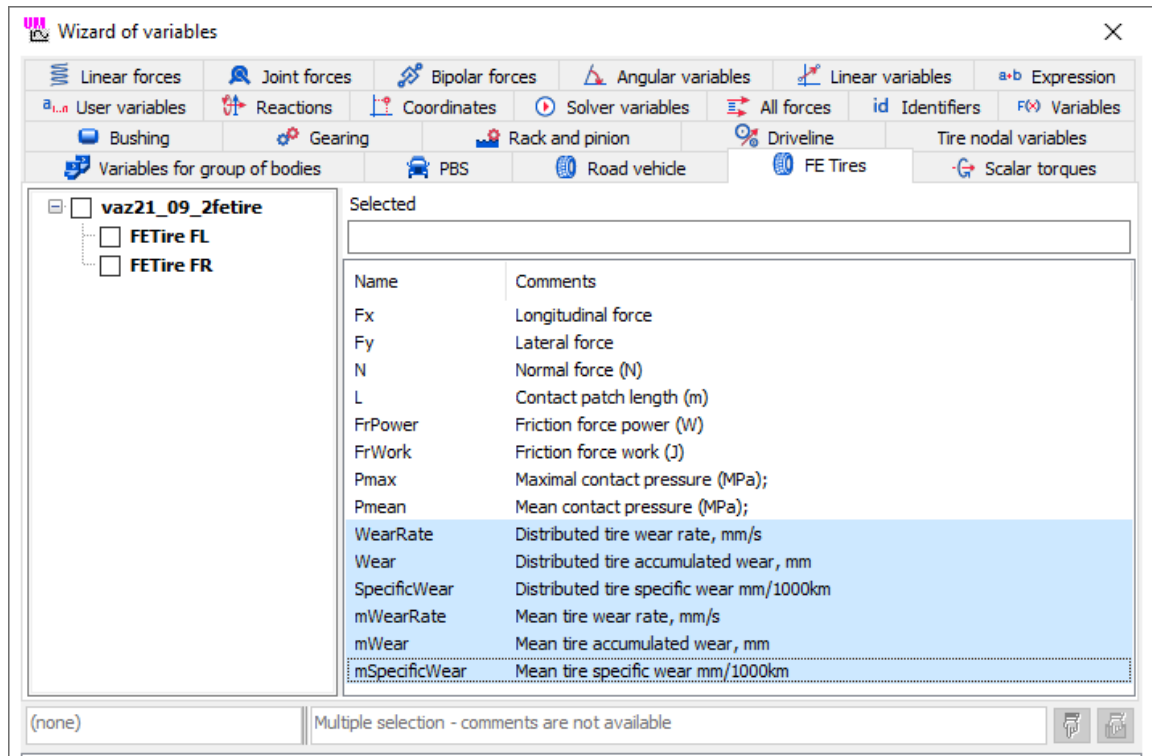


Figure 33.170. Distributed tread wear factors in the Wizard of variables

Variables calculating distributed wear factors (33.15), (33.16), (33.17) are formed in the Wizard of variables, Figure 33.170, Sect. 33.4.6.1 *FE tire/Road interaction variables*. A distinctive feature of these variables is that at each moment in time they represent the dependence of factors on the transverse coordinate of the tread in the tire SC, i.e. the diagrams, Figure 33.171.

Mean wear variables correspond to averaging over the entire tread surface, for example, the mean accumulated wear factor is determined by the formula

$$h = \frac{\sum_{i=1}^n h_i}{n}$$

The wear rate factor is auxiliary and allows visually assessing the wear intensity on different sections of simulation. The distributed accumulated wear and specific wear are the main variables characterizing the degree of tire wear. Specific tire wear is also a base indicator that allows one to assess the contribution of a particular driving mode for wear.



Figure 33.171. Examples of graphs of the distributed wear rate factor when driving on a straight section of the road

33.4.8.3. Methodology for assessing the intensity of tire wear in various driving modes

The driving modes that affect the wear rate include:

- speed mode: speed value, acceleration and deceleration sections;
- uphill and downhill sections;
- turns, overtaking and lane changes;
- the state of the tire contact with the road, expressed in different values of the friction coefficient.

The listed factors are usually combined, for example, a WV turn can occur at different speeds, on slopes and in different road conditions, so there are a large number of options for evaluation. Let us briefly consider how the modes are set; for more details for a car, see [Chapter 12](#).

Setting the speed mode

The speed mode is set for a road vehicle on the driver test parameters tab, Figure 33.172, and on the **Speed** tab for a monorail vehicle. Usually, when analyzing tire wear, the driving modes with a constant speed (**v=const**) and the speed graph set depending on the distance (**Profile**) are used. The constant speed is set on the identifiers tab (identifier v_0) in m/s or km/h, Figure 33.172.

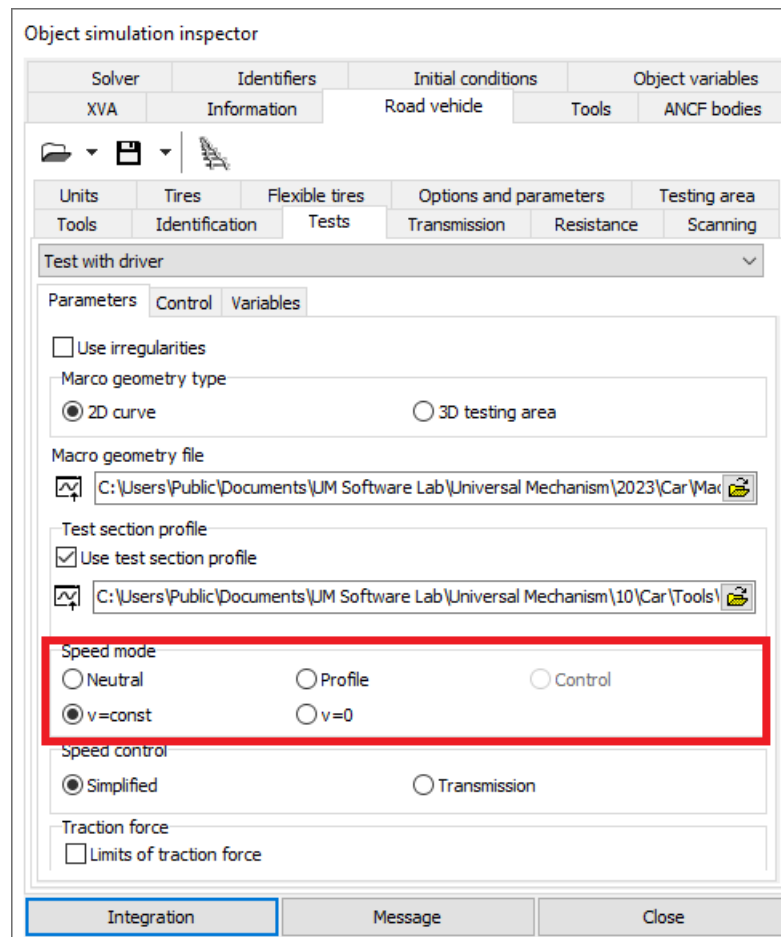


Figure 33.172. Setting speed mode for road vehicle

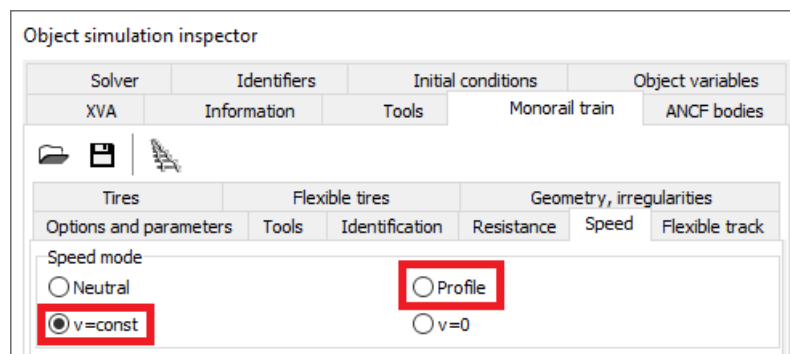
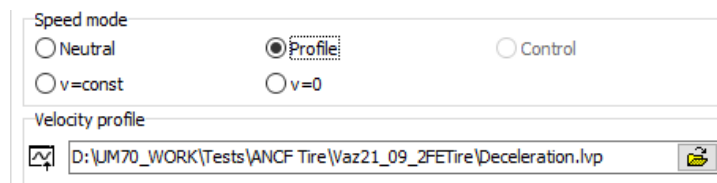


Figure 33.173. Setting speed mode for monorail vehicle

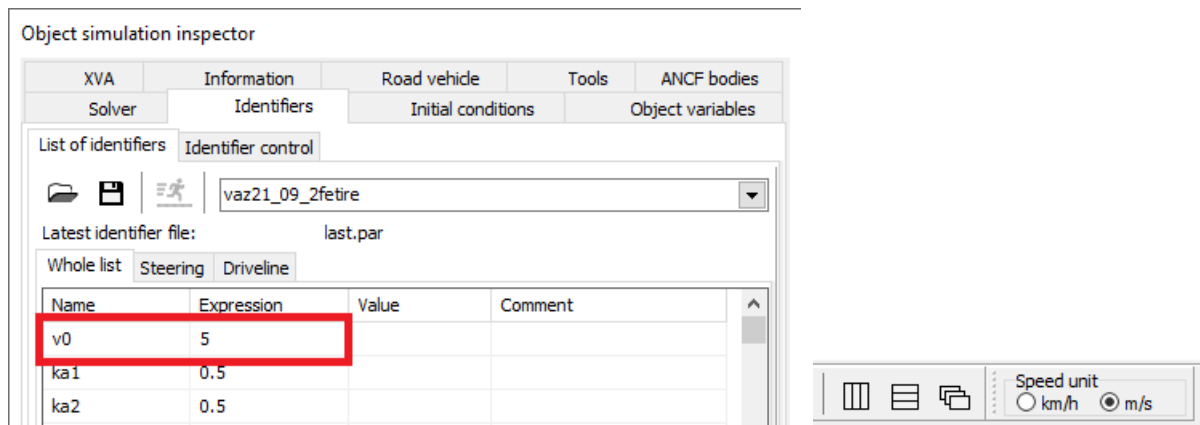


Figure 33.174. Setting identifier and unit of measurement of vehicle speed

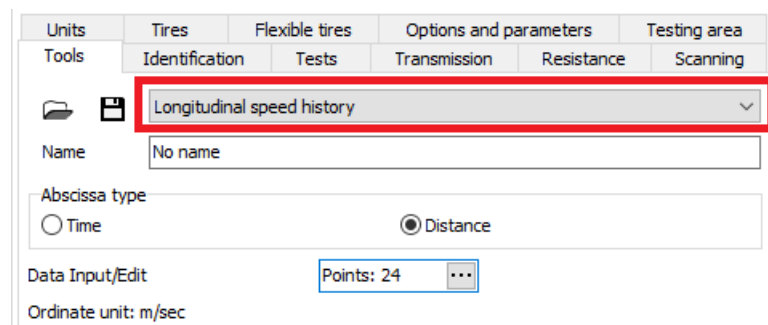


Figure 33.175. Tool for development of speed history file

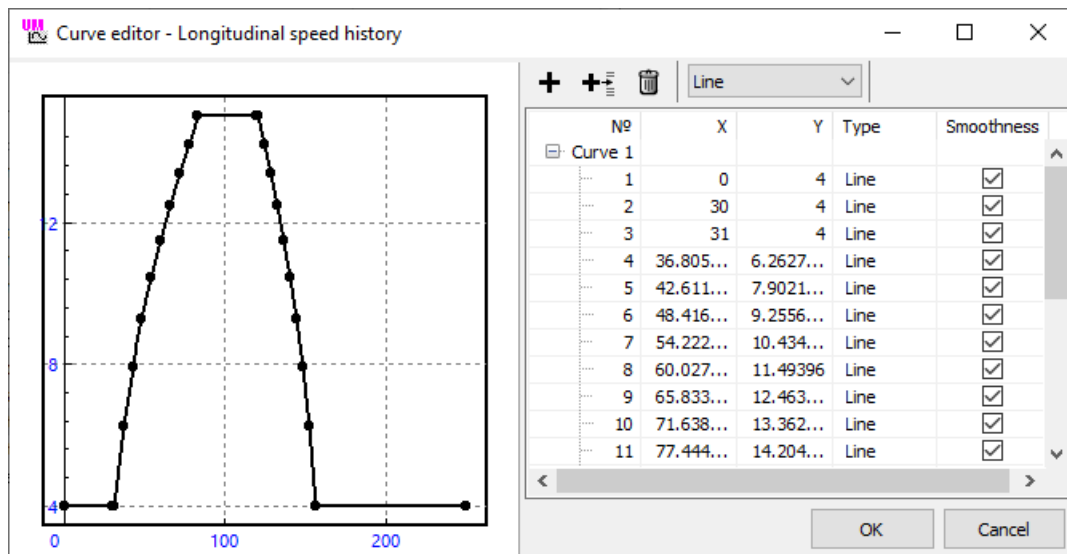


Figure 33.176. Speed vs distance in curve editor

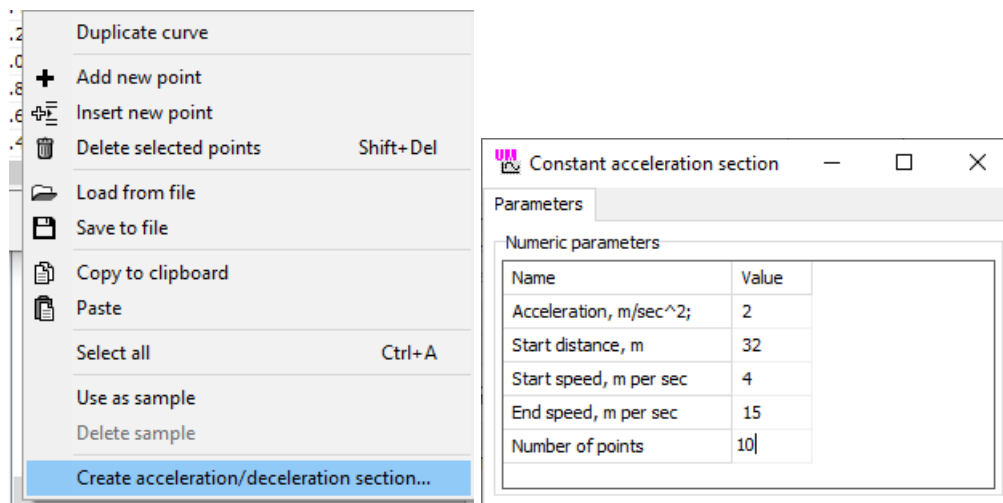



Figure 33.177. Creating a speed vs. distance plot with constant acceleration

To specify a file with a speed-depending-on-distance data, use the **Tools** tab, **Longitudinal speed history** section. The button  opens the curve editor, in which the speed versus distance dependence is specified point by point, Figure 33.176.

In the example in Figure 33.176, the speed graph has acceleration section from 4 to 15 m/s and a deceleration section from 15 to 4 m/s. In both sections, the acceleration value is constant and equal to 2 m/s². To automatically insert such sections in the table, right-click on the table with coordinates, select the **Create acceleration/deceleration section** command, fill in the section parameters in the window that appears, and accept the input results after closing it, Figure 33.177. It is recommended to smooth the sections of variable speed with splines.

Note. To obtain smooth acceleration graphs, it is recommended to smooth the sections of variable speed with splines, Figure 33.178.

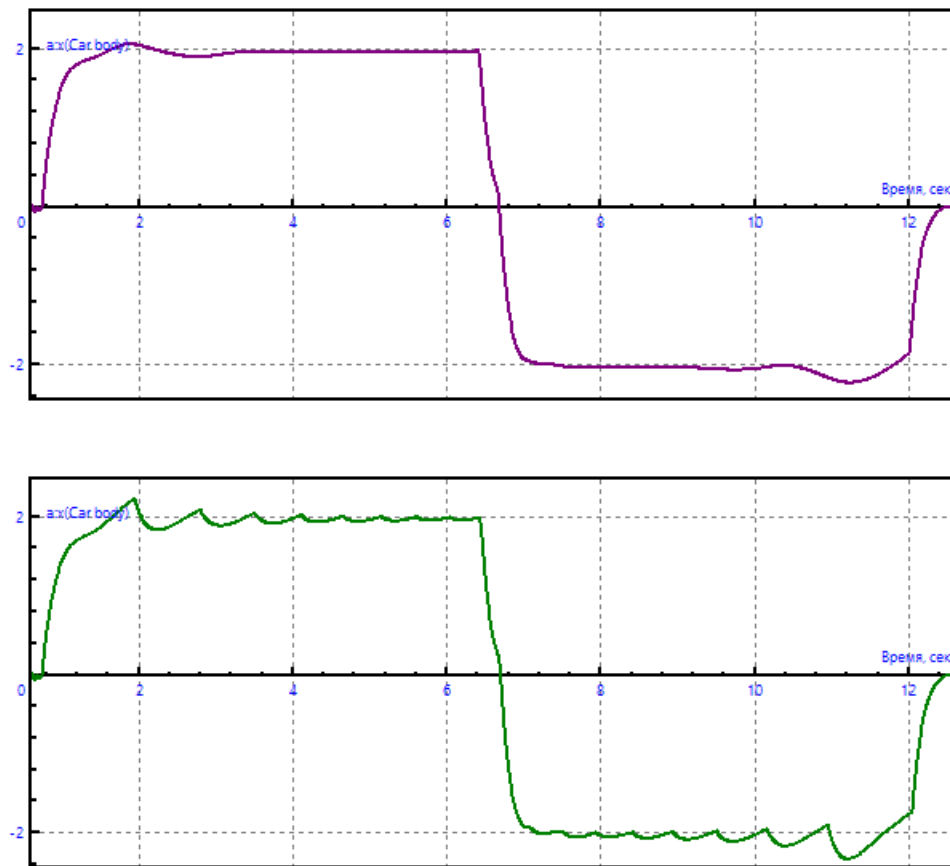


Figure 33.178. Graph of acceleration of a car when the dependence of speed on the distance is specified by splines (above) and straight-line segments

33.4.8.4. Methodology for assessing accumulated tire wear during operation

The prediction of tire wear during operation using the FE tire model consists of performing a series of simulations with calculation of the accumulated wear factor when the WV is moving under various conditions. We propose using an approach based on creating a representative set of tests that reflect the main conditions of WV movement. In this case, the wear factor accumulates from experiment to experiment. During the simulations, the length of the contact model bristles decreases in accordance with the accumulated wear, which leads to the **influence of wear on the contact interaction of the tire with the road**.

33.4.8.4.1. Parameters for modeling wear accumulation in a separate numerical experiment

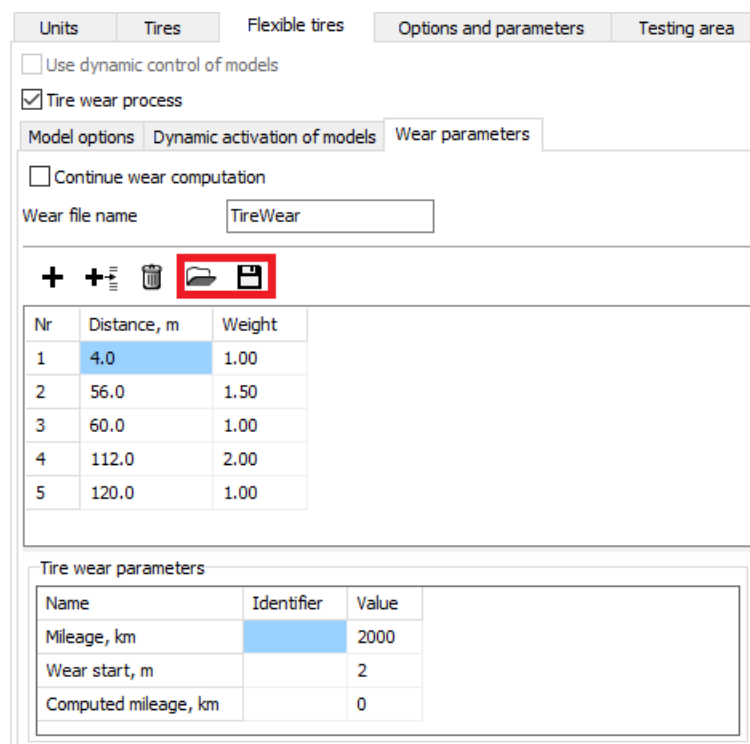


Figure 33.179. Parameters for modeling the tire wear accumulation process

Tire wear computation – the option enables the tire wear accumulation process mode as a result of a series of calculations.

The settings for a separate calculation include:

- macro geometry file;
- speed setting mode, 33.4.8.3 *Methodology for assessing the intensity of tire wear in various driving modes*;
- options and parameters of the **Tire wear parameters** tab, Figure 33.179

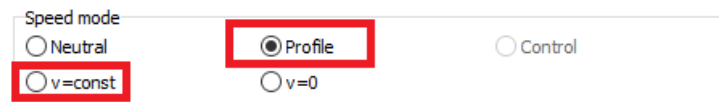




Figure 33.180. Permissible speed modes when modeling the tire wear accumulation process

Two speed modes are allowed: movement at a constant speed (**v=const**) and at a speed specified by the dependence on the distance (**Profile**), Figure 33.180.

The specified settings are saved in a text file *.twc (tire wear configuration). The buttons   are used to write and read files, Fig. 33.175. Examples of files:

```
wear_nodes=4.0,1.00,56.0,1.50,60.0,1.00,112.0,2.00,120.0,1.00;
wear_mileage=,2000;
wear_start=,2;
mgfile="C:\Users\Public\Documents\UM Software Lab\Universal Mechanism\10\Car\MacroGeometry\Straight line 3km.mgf";
lvmode=profile;
lvprofile=D:\UM70_WORK\Tests\ANCF Tire\Vaz21_09_2FETire\Acceleration&deceleration.lvp;
```

```
wear_nodes=2.0,0.00,5.0,2.00,22.7,1.00,25.7,2.00;
wear_mileage=,500;
wear_start=,2;
mgfile="C:\Users\Public\Documents\UM Software Lab\Universal Mechanism\2023\Car\MacroGeometry\turn 90deg.mgf";
lvmode=constant;
v0=5;
```

The calculation result is the variable **Distributed tire accumulated wear**. The variable is saved at the end of each calculation in the text file *.twr (tire wear results) for each wheel with FE tire.

During the first calculation, a *.twr file is created with the name specified in the **Wear file name** box. If the **Continue wear calculation** mode is enabled at start of the the next calculation, the program analyzes the *.twr file with the specified name and sets the distributed wear factor values corresponding to the last record in this file. After the end of the current calculation, it appends the final values of the variable in the same file. Before each new calculation of the series, you can change the macrogeometry file and the speed mode. This way, wear accumulates when moving along different sections of the road and at different speed modes.

The table **Tire wear parameters** includes the values of some auxiliary parameters

Mileage: the mileage value that is assigned to the current calculation when the **Tire wear process** option is enabled;

Wear start: an auxiliary parameter that excludes the initial simulation section, so that possible transient processes at the start of the simulation do not distort the result;

Computed mileage: an auxiliary parameter indicating the previously calculated mileage before the start of the current simulation. This parameter is automatically set by the program when the option **Continue wear calculation** is enabled. This value is set after reading the *.twr file. When the **Continue wear calculation** option is disabled, the parameter value is set to zero.

The table of weighting coefficients of macrogeometry sections (Figure 33.181) is used only when the **Tire wear process** option is enabled (Figure 33.179). The table should be generated for each specified macrogeometry. The trajectory specified by the macrogeometry file for simulation with number j is divided into sections by specifying the distance from the start s_{ij} (the first column of the table). For each section, its weight w_{ij} is specified (the second column). The accumulated wear H_j at the end of the current simulation for a separate tire tread ribbon is determined by the formula

$$H_j = H_{j-1} + \sum_{i=1}^{n_j} h_{ij} \frac{w_{ij}}{w_j} \frac{S_j}{s_{ij} - s_{ij-1}}, \quad w_j = \sum_{i=1}^{n_j} w_{ij} \quad h_{ij} = \int_{t_{i-1}}^{t_i} \dot{h} dt$$

where S_j is the milage in meters assigned to the current section (Figure 33.179), n_j is the number of sections in the computation j , s_{0j} is the wear start parameter.

Usually, the boundaries of the calculation sections coincide with the curve description points in the macro geometry file, therefore, to create a table, it is convenient to use the **Assign from current macrogeometry** pop-up menu command, Figure 33.181.

Nr	Distance, m	Weight
1	4.0	1.00
2	56.0	1.50
3	60.0	1.00
4	112.0	2.00
5	120.0	1.00

Figure 33.181. Table of weight coefficients of macrogeometry sections

In the wear computation mode, a successful completion of a separate simulation occurs at the moment when the path traveled by the vehicle coincides with the value of the end of the last section (120 m in Figure 33.181). For this reason, the macrogeometry file must contain a trajectory which length is greater than the end position of the last section. The program appends a straight section to the curve if necessary.

33.4.8.4.2. Example of accumulated wear computation

Let us look at an example of calculating accumulated wear to explain the methodology.

Consider the vaz21_09_2FETire model and perform a test calculation of the front left tire wear. The model directory contains three wear configuration files.

Acc_Dec.twc: the car moves along a horizontal straight line with variable speed. The speed versus distance graph is shown in Figure 33.182, i.e. the process of accelerated and decelerated car motion is simulated.

Up_Down.twc: the car moves at a constant speed of 20 m/s along a section with an ascent and descent, Figure 33.183.

R10left_right.twc: the car moves at a constant speed of 5 m/s along a horizontal section with two 90-degree turns with a radius of 10 m, Figure 33.184.

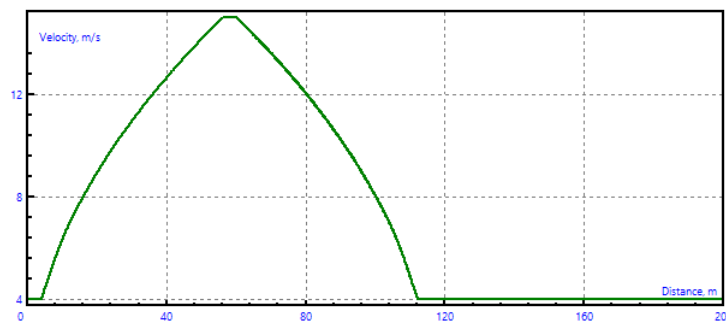


Figure 33.182. Speed vs. distance

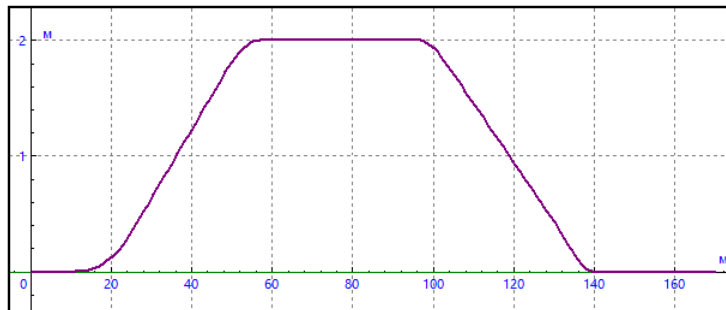


Figure 33.183. Vertical road profile

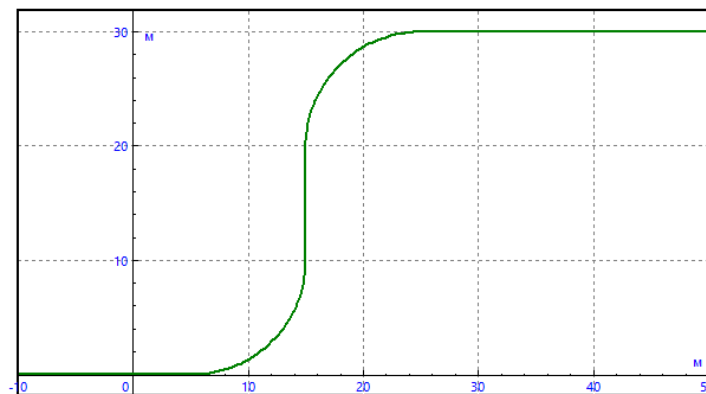
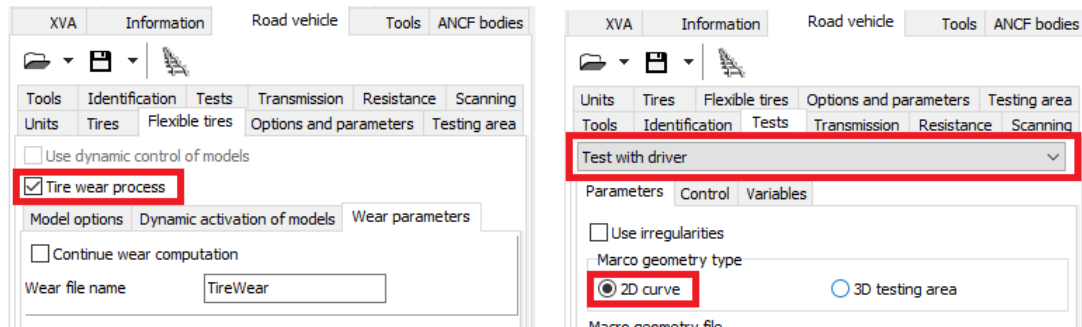
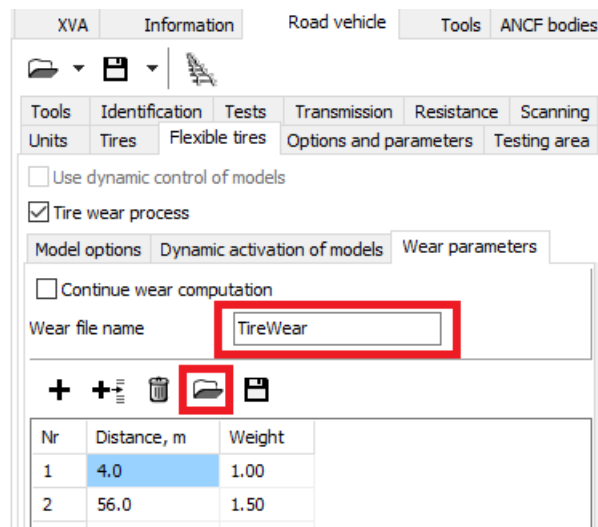


Figure 33.184. Horizontal road profile

Consider the process of wear calculations using data from three configurations.



- Enable the **Tire wear process** option. In case of road vehicle modeling, the test type will be set to **Test with driver** and the macrogeometry type to **2D curve**.



- Assign a name to the wear result file and read the *.twc file corresponding to the first section of the series. In our case, this will be the Acc_Dec.twc file.
- Run the simulation process.
- After the calculation is automatically completed, enable the **Continue wear calculation** option and read the next *.twc file, in our case Up_Down.twc. Run the simulation. After the calculation is completed, run the third file R10left_right.twc. We will call ‘an iteration’ the calculation for all files in the series.
- Continue iterations to obtain the required mileage.
- The results of each calculation are saved in a *.twr file with the specified name. This file can be displayed in the graphics window at any time. The sequence of steps for reading the result file is as follows:
 - open a new graphic window;
 - right-click in the Variables window area to open the pop-up menu and select **Load from file** command, Fig. 33.4;
 - in the standard file opening window, specify or select the *.twr type and open the required file, Fig. 33.4.
- Assign a name to the calculation file and read the *.twc file corresponding to the first section of the series. In our case, this will be the file Acc_Dec.twc.

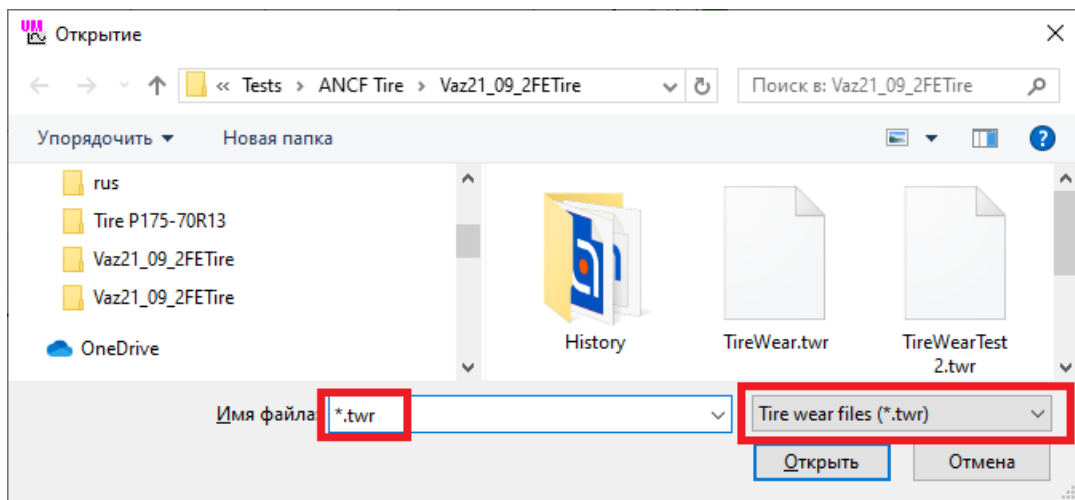
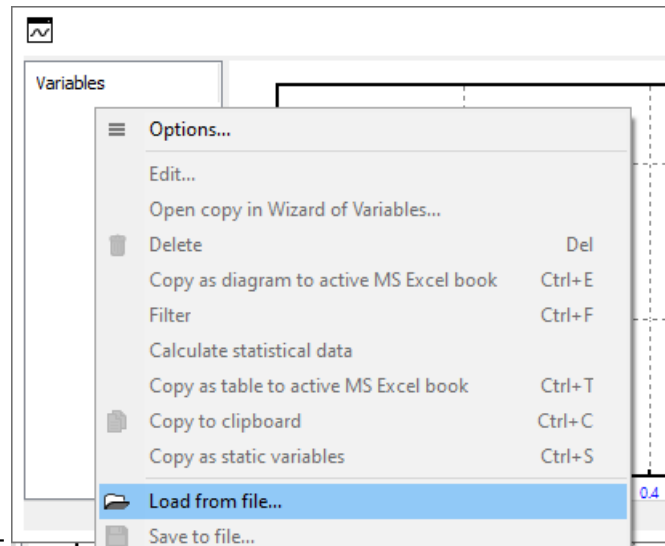


Figure 33.185. Open file with tire wear results

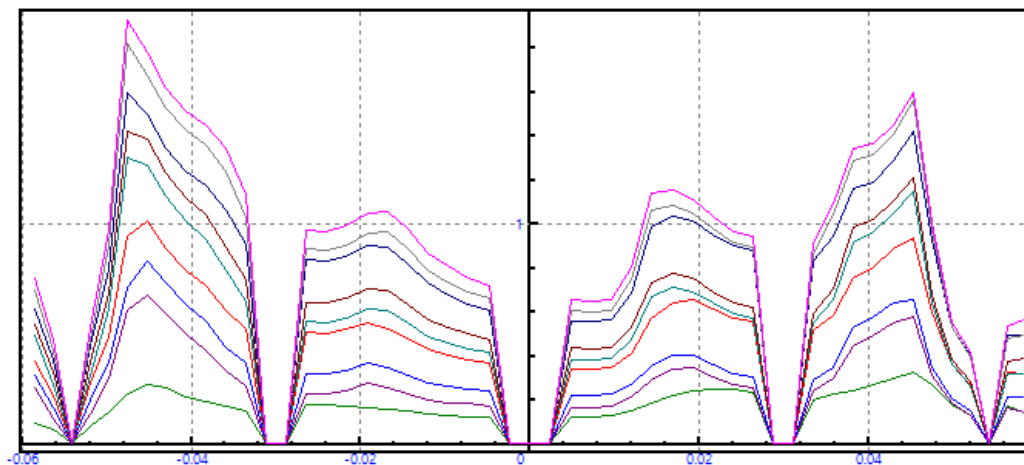


Figure 33.186. Example of wear accumulation with increasing mileage

Figure 33.186 shows an example of performing three iterations with three wear calculations in each.

33.4.9. Examples of simulation of dynamics of wheeled vehicles with elastic tires

33.4.9.1. Road vehicle simulation examples

Consider examples of modeling the dynamics of a passenger car with FE tires using the `vaz21_09_2fetire` model. Since the number of FE tires is less than the number of wheels, the modeling uses dynamic activation of FE tires and reassigning FE tires to wheels, Sect. 33.4.5 *Dynamic activation of FE tires*. Before modeling, load the car model in the UMSimul program. The car model is located in the directory

[{Data UM}\Samples\FETire\vaz21_09_2fetire](#)

33.4.9.1.1. Driving over speed bump

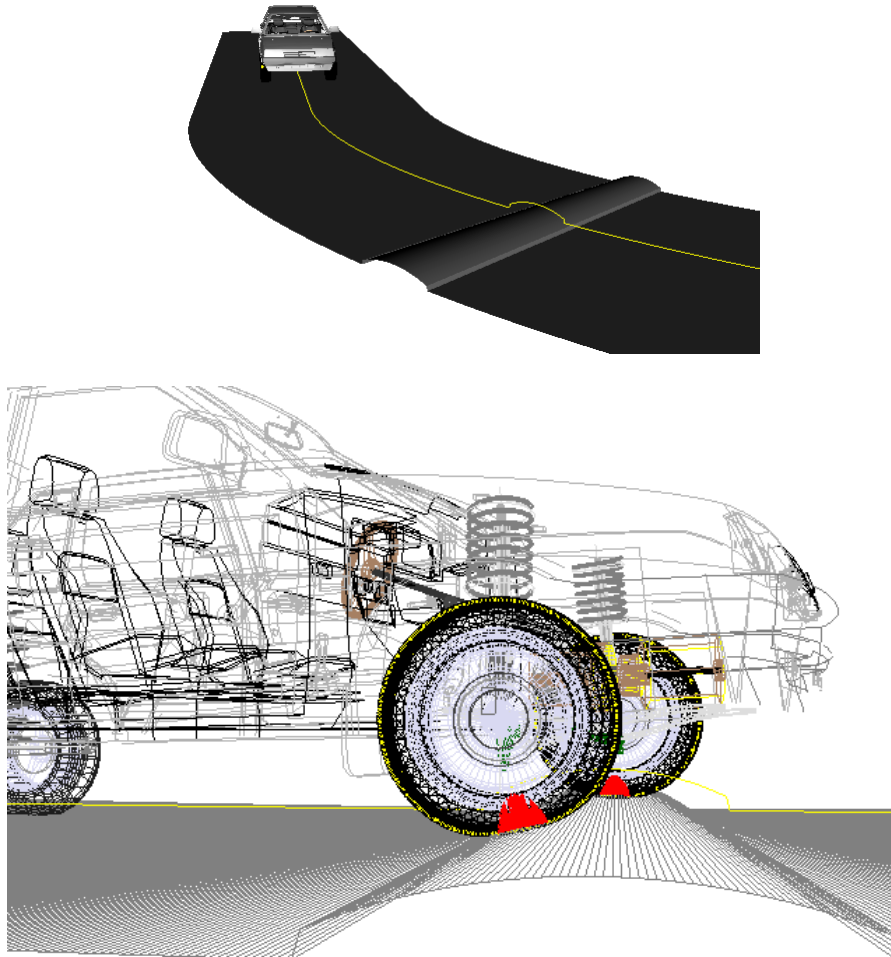


Figure 33.187. Drive over speed bump

Load the full configuration **Speed bump in curve**. In this test, the car moves at a constant speed of 1.5 m/s in a curve with a radius of 11 m. The speed bump is located at a distance of about

11 m from the start of the movement. The control of the FE tire activation is close to that described in Sect. 33.4.5 *Dynamic activation of FE tires*.

33.4.9.1.2. Driving over damaged road

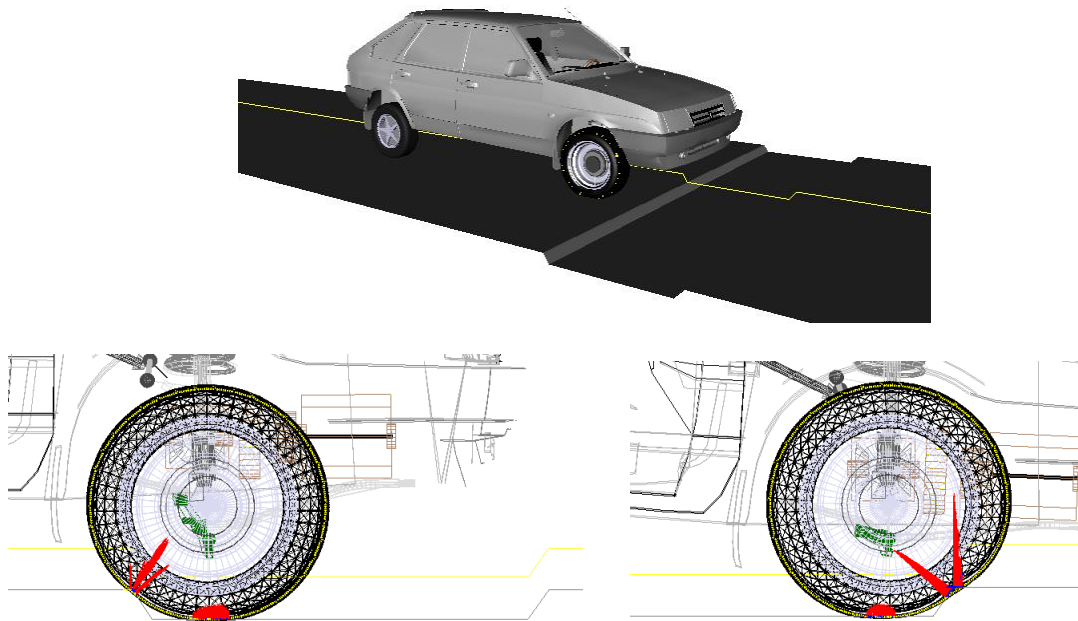


Figure 33.188. Drive over road damage

Download the full **Road hole** configuration. In this test, the car moves at a constant speed of 1.5 m/s through a 7 cm deep hole in the road.

33.4.9.1.3. Driving over road curb

Load the full **Road curb** configuration. In this test, the car moves at a constant speed of 1.5 m/s and drives its front and rear wheels over a curb 11 cm high.

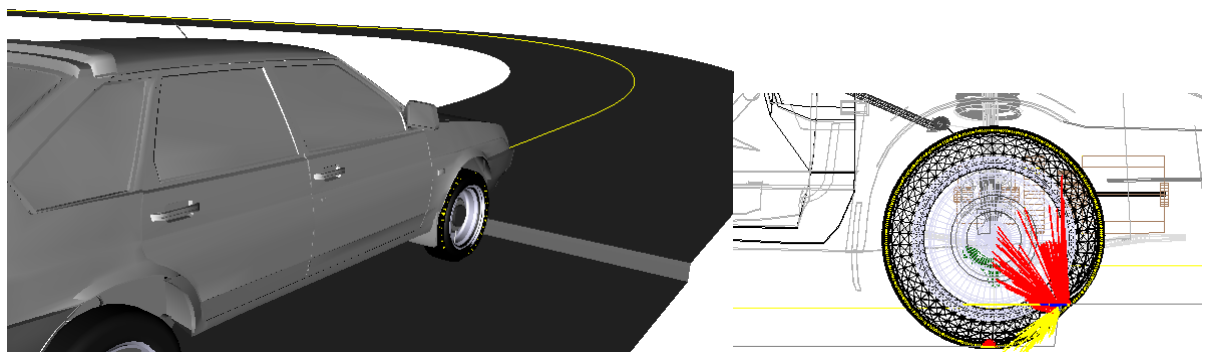


Figure 33.189. Drive over curb

33.4.9.2. Monorail vehicle simulation examples

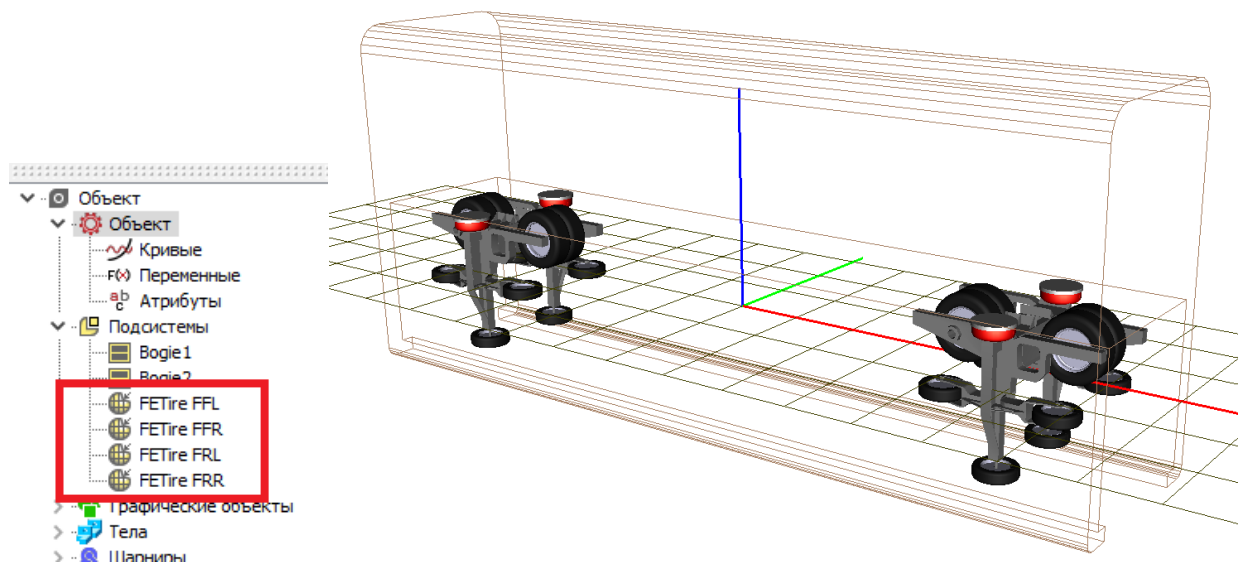


Figure 33.190. Model of monorail vehicle with four FE tires in UMinput program

An example of a monorail vehicle with FE tires is located in the directory

[{UM Data}\Samples\Monorail\Monorail vehicle 4FETire](#)

In the monorail vehicle model, the FE tires can only be assigned to the traction wheels. In the model under consideration, the FE tires are pre-assigned to all four traction wheels of the first bogie.

33.4.9.2.1. Driving over steps

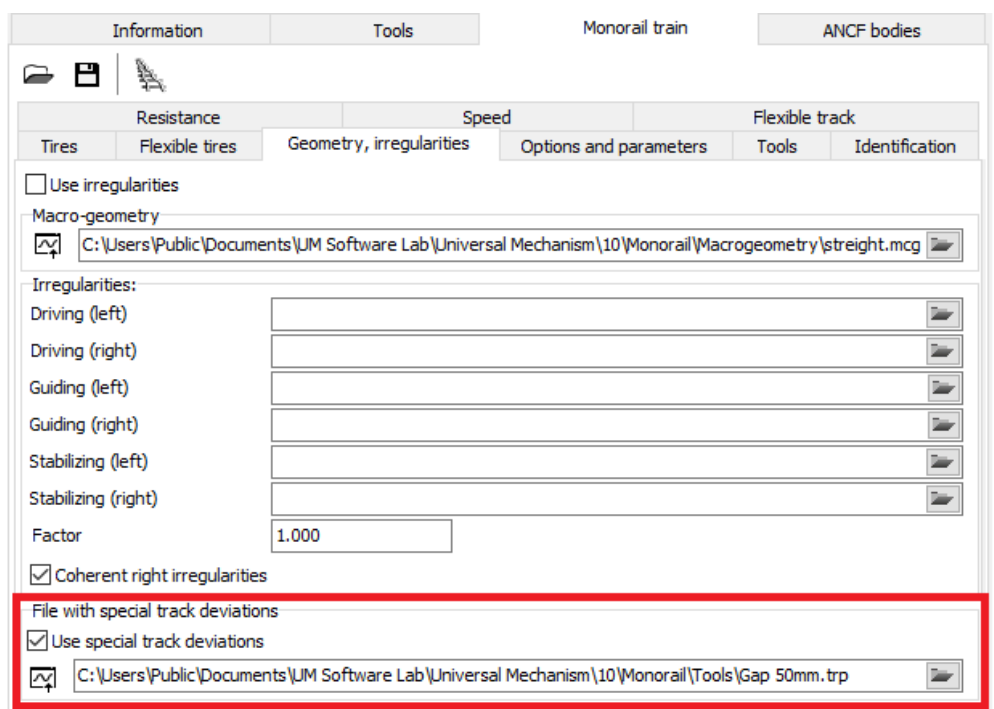


Figure 33.191. Assignment of file with special track deviations

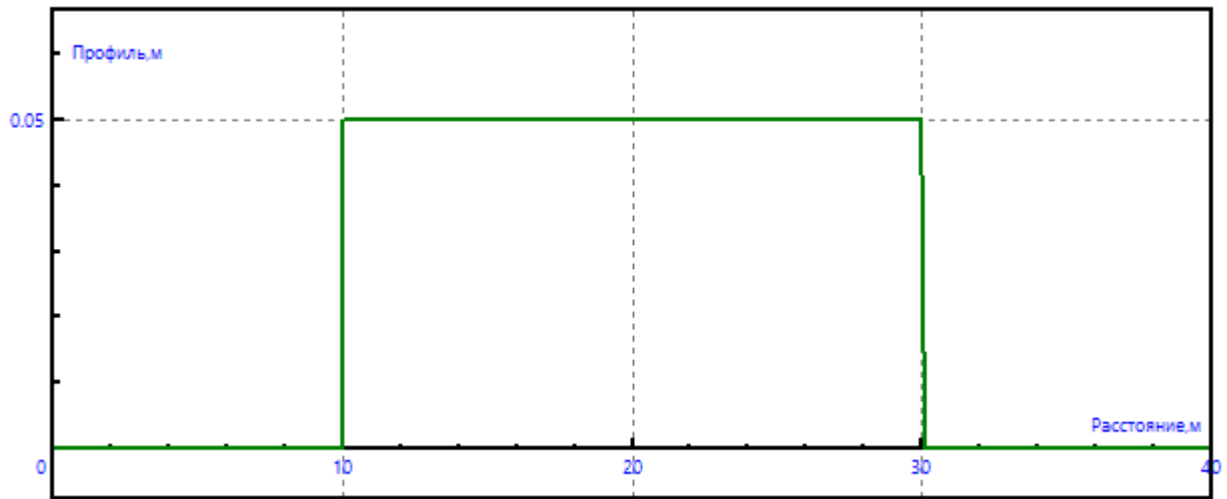


Figure 33.192. Graph of deviation in file ‘Step 50mm.trp’

Open the model in UMSimul program and load the full **Steps** configuration. In this configuration, the car travels along a straight section, enters a 50mm high step and leaves it. The step geometry is specified in the special deviation file ‘Step 50mm.trp’, Figure 33.191. Track roughness is not taken into account. The car enters the step at a distance of 10 m from the origin of the track coordinate system, and leaves at a distance of 30m, Figure 33.192. The **Tools** tab is used to create and modify special deviation files, Figure 33.193.

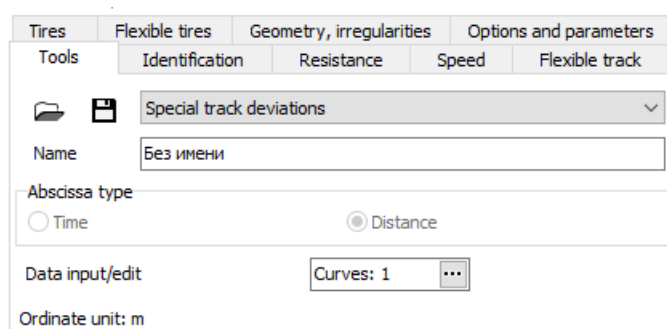


Figure 33.193. Tool for specifying the special track deviations

Any test with the FE tire models enabled must start with the assignment of the FE tire models (see Sect. 33.4.4.1 *Assignment of FE tire models from database*) and the corresponding simplified massless tire models (33.4.4.5 *Consistent models of massless and FE tires*). After that, the initial conditions should be calculated in the situation when all FE tires are active, Sect. 33.4.4.6.2 *Calculation of initial conditions for FE tires as part of the equilibrium test*. These actions have already been performed in the loaded configuration.

To simulate the run over the deviations by all eight traction wheels with FE tires, the test implements dynamic control of the FE tire enabling and disabling. Let us explain the creation of the activation schedule using the example of an FE tire assigned in the initial state to the front left traction wheel of the front bogie, Figure 33.194. The positions in which the tire should be enabled and disabled should be determined. The positions are determined by the tire distance travelled from the moment it started moving.

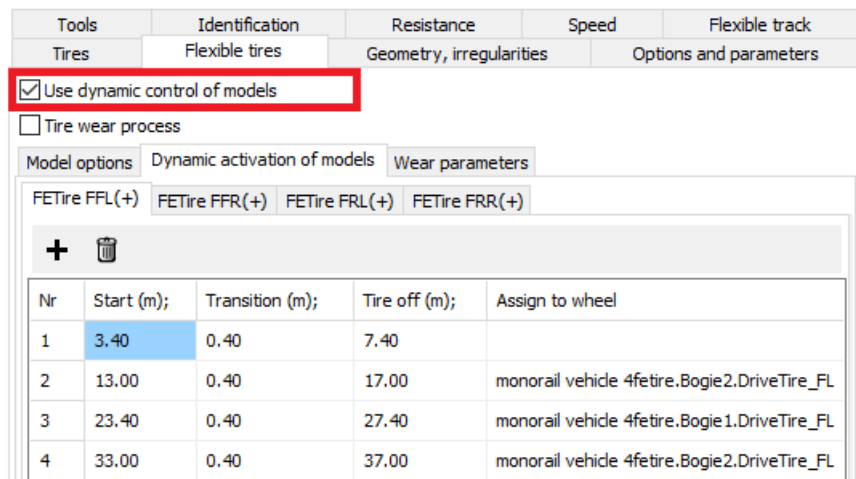


Figure 33.194. Activation section for tire FETire FFL

Let us plot the graphs of vertical tire/road interaction forces for all tires depending on the distance traveled using simplified massless tire models. For this purpose, we deactivate all traction wheels (Figure 33.195). Note that when deactivating a traction wheel, information about its coordinates is lost, and when re-activated, zero values are assigned to tire coordinates.

Run the simulation process. Graphs of vertical forces of all traction wheels depending on the distance traveled are plotted in the graphics window. After finishing the simulation, in the pause mode, we hide the graphs of all forces, except for the front left traction wheels of both bogies, Figure 33.196. Hitting and leaving a step for these wheels will be simulated by the same flexible tire FETire FFL.

The first section of the FETire FFL corresponds to enabling the tire in position 3.4m, the length of the transition section is 0.4m, i.e. in the 3.8m position the FE tire replaces the simplified one and is included in the dynamic process as the tire of the front left wheel of the first bogie, to which FE tire is assigned by default. In the 7.4m position the FE tire is disabled and replaced by the massless one.

In the second section the FETire FFL is assigned to the front left wheel of the second bogie, in the third - to the first bogie and in the last section - again to the second one.

For the tire used for assignment to the front right traction wheels, the section parameters are the same, so you can use the tool for copying a set of sections using the pop-up menu by right-clicking, Figure 33.197.

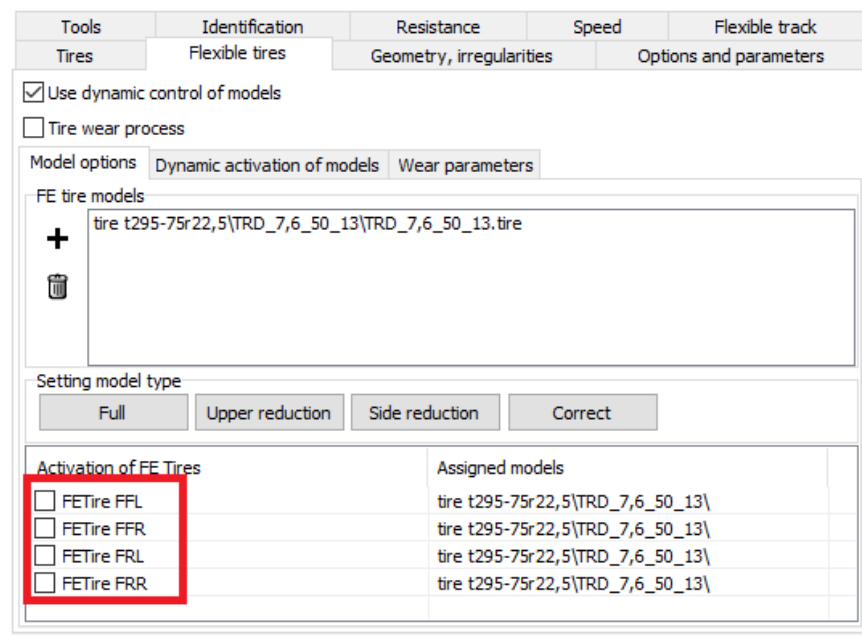


Figure 33.195. All FE tire disabled

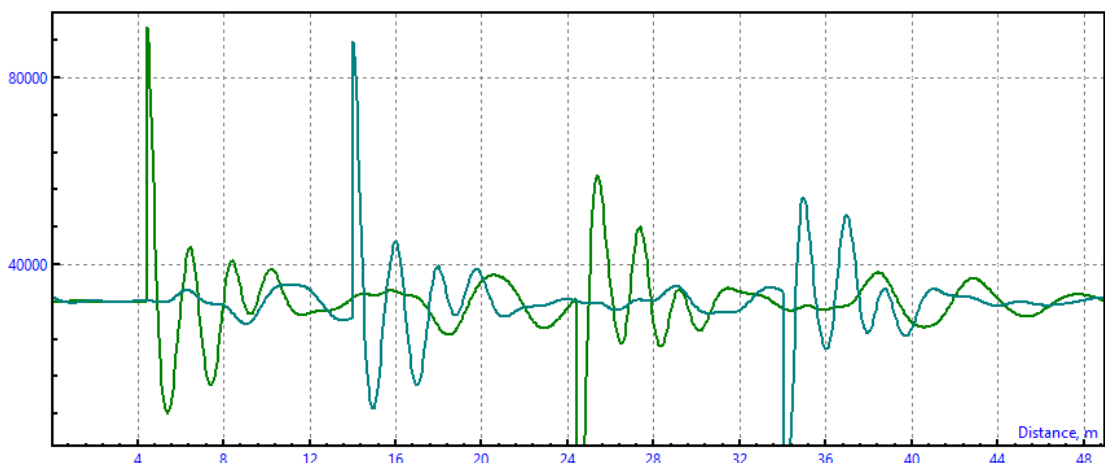


Figure 33.196. Graphs of vertical forces for the front left traction wheels depending on the distance traveled

Nr	Start (m);	Transition (m);	Tire off (m);	Assign to wheel
1	3.40	0.40	7.40	
2	13.00			ire.Bogie2.DriveTire_FL
3	23.40	0.40	27.40	ire.Bogie1.DriveTire_FL
4	33.00	0.40	37.00	ire.Bogie2.DriveTire_FL

Figure 33.197. Assigning sections to another FE tire

After entering all sections, the user should

- reload the full **Steps** configuration or activate all the FE Tires and read the initial conditions from the last.xv file, since when the FE tires are deactivated, information about their coordinates is lost, and when they are reactivated, zero values are assigned to them;

- run the simulation of the dynamics of driving through special irregularities using the FE tires. The result of simulation at a speed of 10 m/s is shown in Figure 33.198.

Comparison of the results for different FE tire models (full, reduced) shows a very good agreement in this test (Figure 33.199), and the integration of equations using reduced FE tire models occurs approximately twice as fast as using the full model.

In conclusion, we note that the use of an unmodified FE tire model with sidewall reduction leads to a significant difference in the results due to the increased rigidity of the reduced part under large elastic displacements, Figure 33.200, 33.3.13.4.4 *Evaluation of the parameters of a modified tire model with a reduced sidewall* .

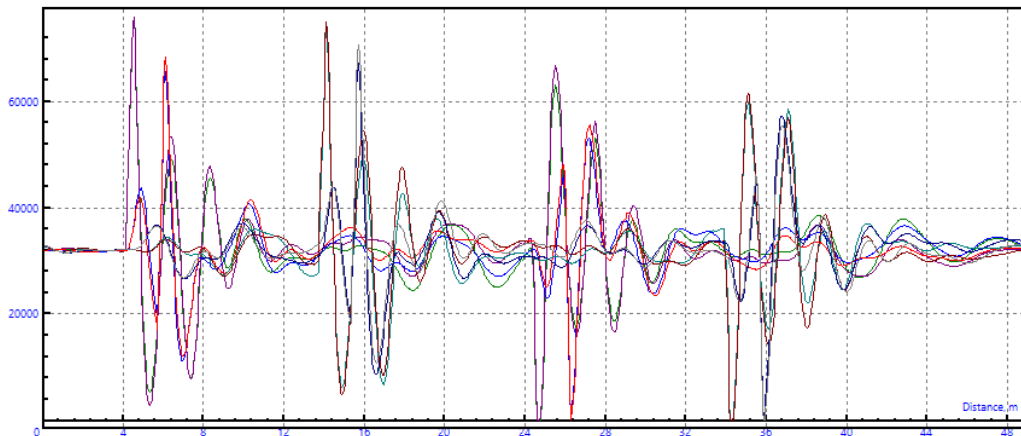


Figure 33.198. Run over steps with FE tires

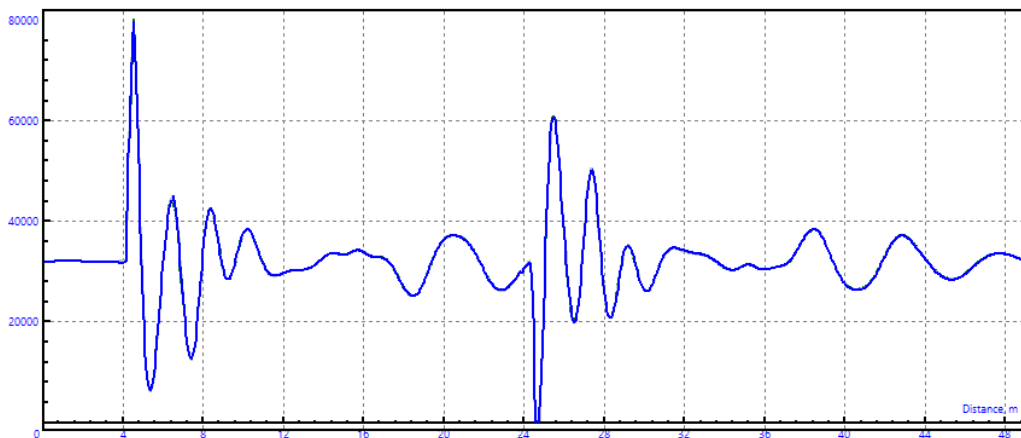


Figure 33.199. Comparison of the vertical force on the front left tire of the first bogie for the full and reduced models of the FE Tire. The results are almost identical.

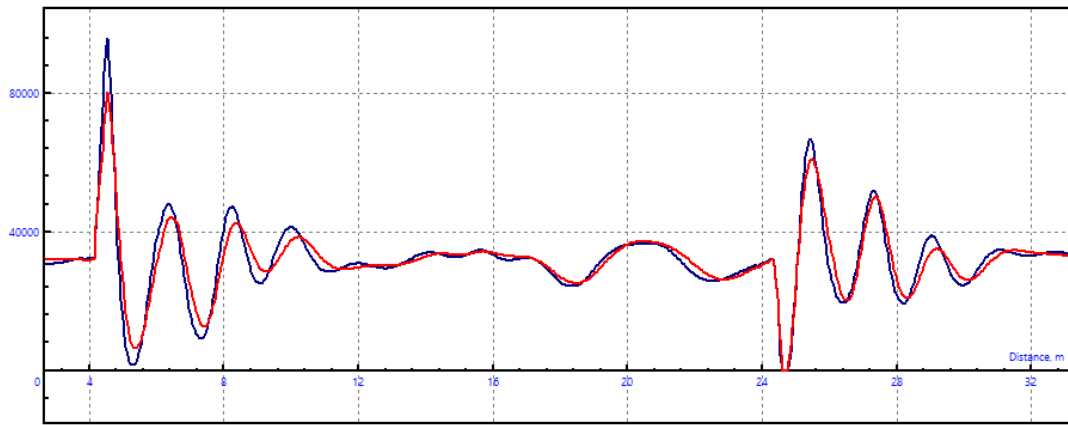


Figure 33.200. Comparison of vertical force on the front left tire of the first bogie for the full FE tire model and unmodified model with sidewall reduction

33.4.9.2.2. Driving through a narrow gap in a track structure

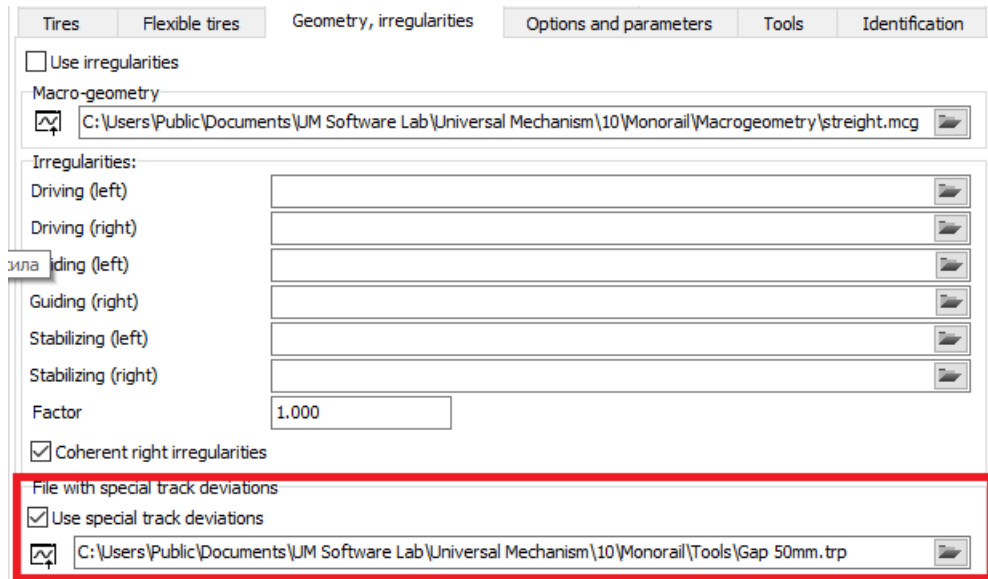


Figure 33.201. Assignment of file with track deviation ‘Gap 50mm.trp’

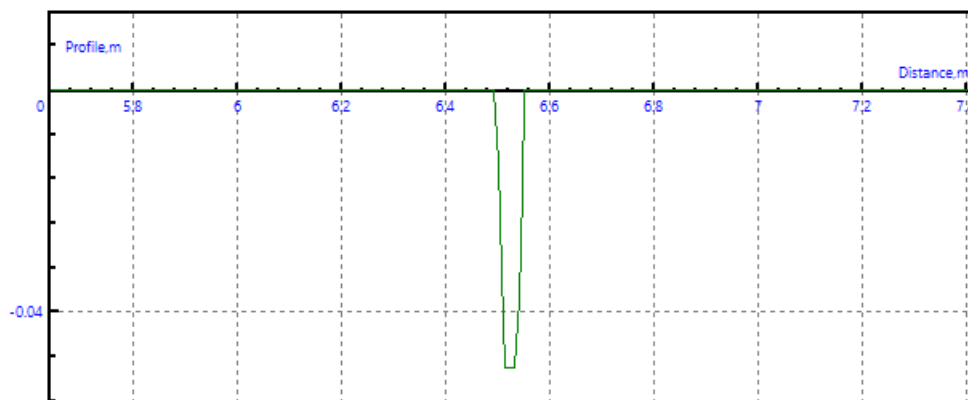


Figure 33.202. Special track deviation ‘Gap 50mm.trp’

Open the model in UMSimul program and load the full **Gap** configuration. In this configuration, the vehicle travels along a straight section and runs over a gap in the track structure with a width of 50 mm. The gap geometry is specified in the special track deviation file ‘Gap 50mm.trp’, Figure 33.201. Micro-roughness is not taken into account. The gap is located at a distance of 6.5 m from the track coordinate system origin, Figure 33.202, Figure 33.203.

To speed up the simulation of motion over the gab by four traction wheels of the front bogie with the FE tires, the test implements the dynamic control of activation the FE tires. The graphs of the vertical forces acting on the wheels when passing the gap are shown in Figure 33.204.

To obtain smoother graphs in this test, the number of bristles in the longitudinal direction was doubled compared to the previous one, Figure 33.205.

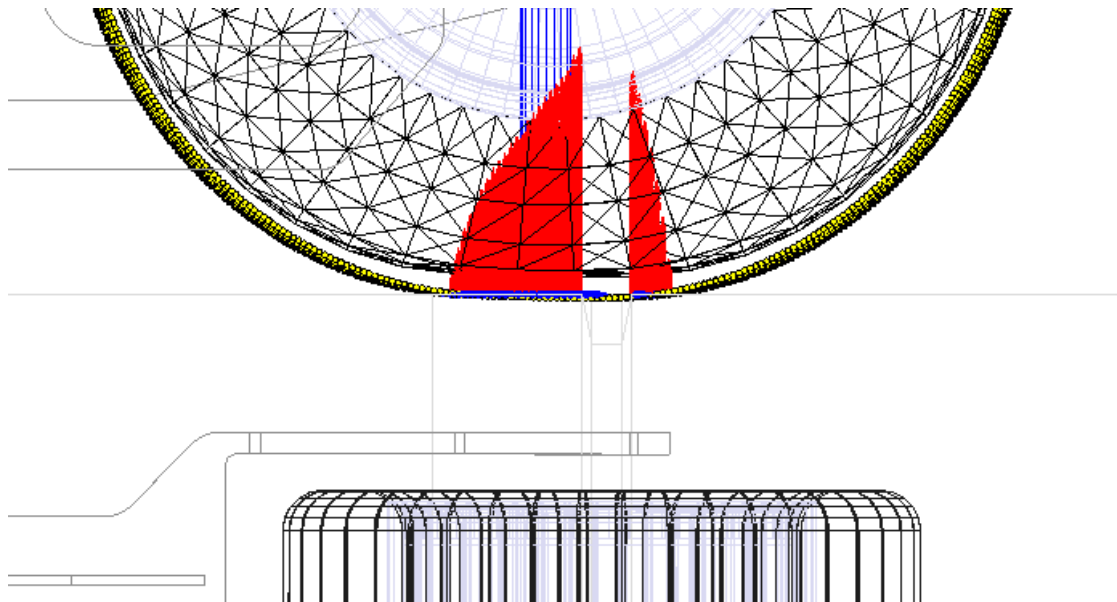


Figure 33.203. Vertical contact pressures

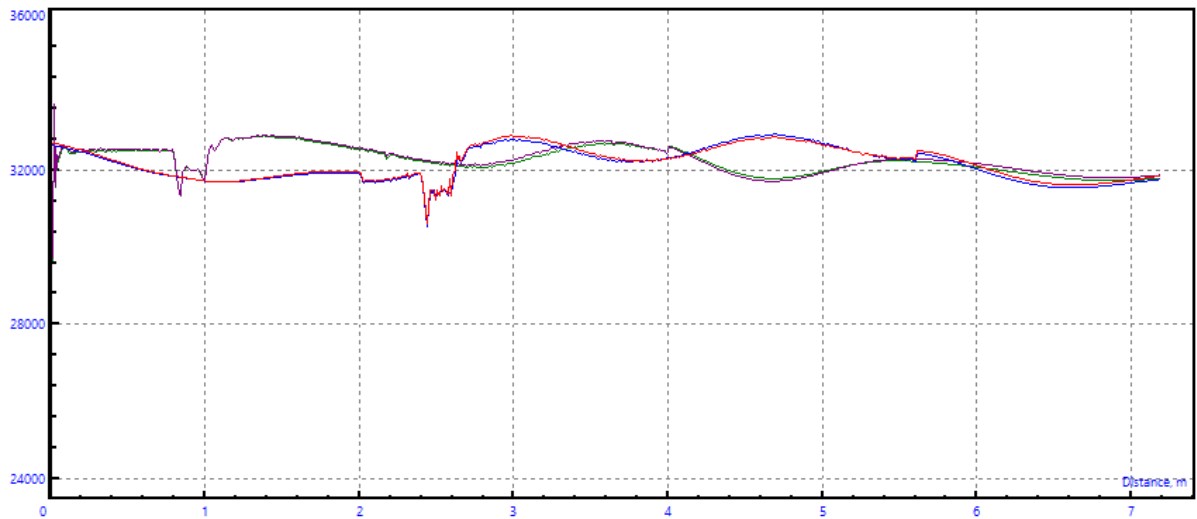


Figure 33.204. Vertical forces for the front bogie tires when passing a gap in the track structure beam

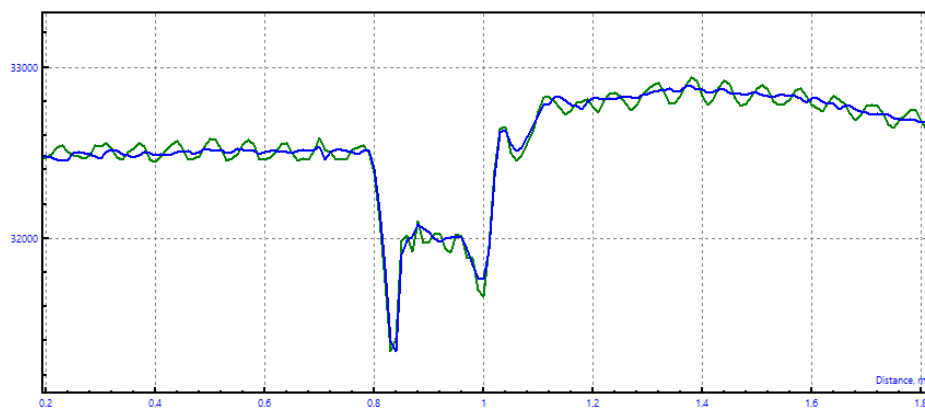


Figure 33.205. Comparison of vertical force for the front left wheel with 5 and 10 bristles in the longitudinal direction for one finite element

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