# Modeling of the Lgnss-D Freight Vehicle

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29th January 1999

# Contents

**Abstract**

This report describes the different steps in modeling and simulating a rail vehicle for freight transport. The vehicle axle suspension consists of a number of leafsprings connected to the vehicle chassis through chains. In the project, special emphasis has been placed on modeling this suspension concept in a calculation time efficient manner.

A parametrised model is defined of a freqight railway car, the model can be used in a wide range of applications with minimal model adjustments required.

# Chapter 1

# The suspension model

In this project, A model has been defined for a leafspring used for dynamic simulations of Railway vehicles. The model is defined in the modelling and simulation language ADAMS.

## 1.1 Overall description

The used model of the leafspring suspension is a combination of geometrical stiffness and stiffness due to a force model as defined by employees of the University of Michigan Transport Research Institute (UMTRI) see [1]. A side view of the model of one axle suspension is included in figure 1.1.

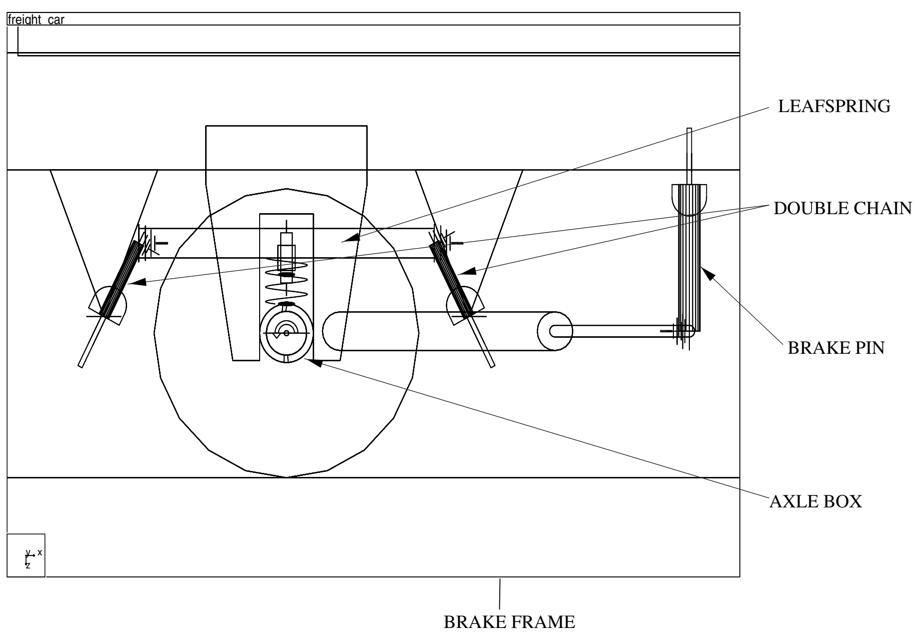


Figure 1.1: Side view of the axle suspension model

In figure 1.1 one can distinguish the different elements in the leafspring suspension model. Modelling method and assumptions will be further explained in the text.

- The leafspring is modelled as a rigid part, connected to the vehicle chassis using two *chain elements*. The axle box parts are connected to the rigid leafsprings using a translational joint, allowing only relative motion in vertical direction.
- The remaining degree of freedom of the translational joint represents the bending deflection of a real-life leafspring. In this direction, a special spring-damper model is defined between the leafspring part and the axle box part. The model defined for describing the spring-damper behaviour of the leafspring incorporates the specific dynamic force-deflection behaviour of leafsprings. This behaviour has been measured extensively by employees of UMTRI for truck-based leafsprings. Based on these measurements, a model has been proposed (see [1]) which has been adopted for use in the current model.
- The chain elements are represented as rigid parts. They are connected to the leafspring part using spherical joints. To prevent rotation about the longitudinal axis of the chain parts, the connection chain-to-chassis is done using hooke-joints. In this manner, the pendulum effect of this type of leafspring suspension is modelled geometrically correct. This ensures correct stiffness values of the suspension in longitudinal and lateral direction. The advantages of the method are:
  - Stiffness values automatically depend on the load on the suspension.
  - Non-linear effects at large suspension deformations are considered.
  - Compared to a *flexible parts* leafspring model, the described model introduces a minimal set of degrees of freedom for the motion of the leafspring wrt the vehicle chassis.
- With the above described elements, the implementation method for vertical stiffness and damping of the suspension as well as (a part of) the linearised stiffness in the horizontal plane of the suspension is described.
  The additional stiffnesses in horizontal direction as well as the Coulomb-like damping forces generated in the contact between different parts of the suspension is modelled using a *general force* element between the Axle Box parts and the vehicle chassis. In figure 1.1 this element is visualised by the coilspring graphic. To accurately describe the Coulomb behaviour of these elements, a UMTRI based description is used to model the longitudinal and lateral linear stiffness and damping as well as the rotational stiffness and damping about the vehicle vertical axis. The rotational torque-deflection properties are necessary because only one *general force* element is used at the centre point of the leafspring. Thus, the lateral contact properties of two chain elements at the ends of the leafspring are modelled as one equivalent element including lateral as well as torsional properties.
- The dynamic properties of this force element must be fitted with measurements results to accurately describe:
  - The double stiffness effect in case of two-link chains at the deflection where one of the chain links encounters its bounce stop.
  - The friction effects generated in the contact between the different parts of the suspension (i.e. chain-to-leafspring, axle-box-to-chassis). This effect is typically load dependent which is therefore considered in the method.

  - The bump stop effect at the in-plane deflection limits of the suspension

- In the current application, the parameters of the connection element axle box - chassis depend on the amount of vertical load in the suspension. This is implemented by a smooth interpolation between parameters for an unloaded and a loaded vehicle.

## 1.2 The UMTRI leafspring model

The force-versus-deflection properties of leaf springs has been measured and approximated by several investigators. A successful model is proposed and described in [1]. Based on an extensive series of measurements the following conclusions were drawn:

- The force producing properties of leaf-springs are independent of frequency for oscillations occurring in the range of 0 to approximately 15 Hz.

- The force-deflection properties of leaf-springs depend upon both the amplitude of motion and the nominal load in the spring.

The energy dissipated in one cycle can be described using two parameters:

- Average coulomb damping force $C_F$,

- Effective spring rate, $K_e$.

Where the parameters mentioned can be estimated using x-y plots of the characteristics.

The method described is applied to obtain an empirical description of spring properties using measurement data. Tendencies in derived parameters indicate that effective spring rate decreases and average Coulomb friction increases as deflection amplitude increases. These results indicate the stimulation of low damped high frequent oscillations at small amplitudes and higher damped low frequent oscillations at larger amplitudes.

For modelling purposes, equation 1.1 is proposed to represent the characteristics of leaf springs

$$F = F_{ENV} + (F_{i-1} - F_{ENV})e^{-|\delta - \delta_{i-1}|/\beta} \tag{1.1}$$

Where:

$F$ , $F_{i-1}$ : the suspension force at the current and last simulation time step

$\delta$ , $\delta_{i-1}$ : the suspension deflection at the current and last simulation time step

$F_{ENV}$ : the force corresponding to the upper and lower boundaries of the envelope of the measured spring characteristics at $\delta$. These boundaries are obtained by the spring force plus or or minus a Coulomb force contribution. The sign at the Coulomb force depends on the velocity of the spring deformation. Examples of the splines to create the envelopes are shown in figure 1.2.

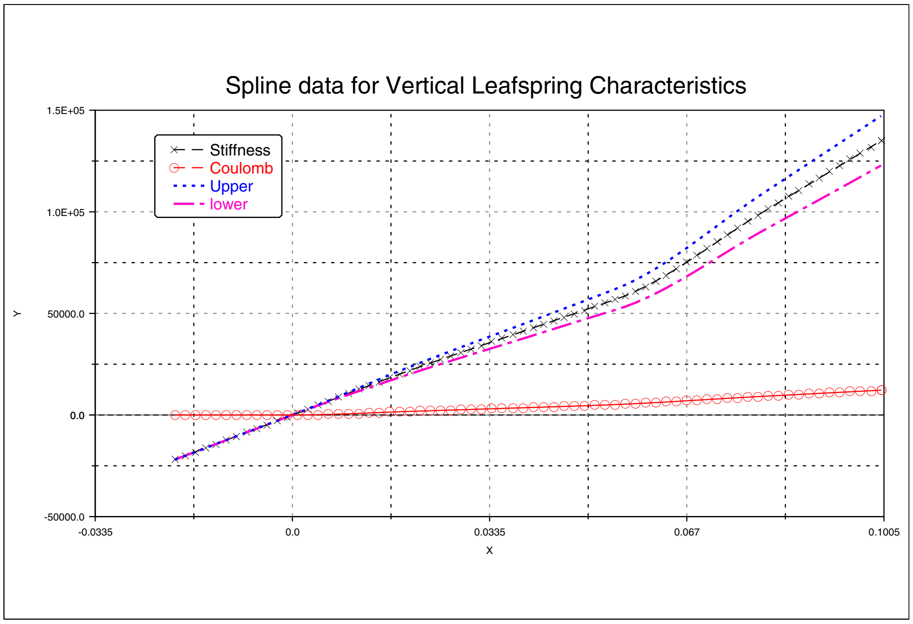


Figure 1.2: Spline data for vertical spring and Coulomb behaviour

$\beta$ : an input parameter used for describing the rate at which the suspension force within a hysteresis loop approaches the outer boundary of the envelope, $F_{ENV}$. Different values of $\beta$ for increasing versus decreasing deflection may be specified

The typical results of a UMTRI based spring-damper model compared to a viscous model are shown in figure 1.3

## 1.3 Implementation of vertical forces

The above mentioned algorithm is used in a macro defined in ADAMS. Where applicable, model variable names are shown in typewriter font in the text.

A general set of model parameters is applied to cover a wide range of measurements. For the z-direction, the friction force is modelled as follows:

- The force envelope $F_{ENV}$ is described using one spline function for the spring force (`STIF_SPL`) and one for the Coulomb force (`COUL_SPL`). Both characteristics are a function of the spring deflection. The total force envelope equals :
  $F_{ENV} = \texttt{STIF\_SPL} + \mathcal{A}\left(\dot{\delta}\right)\ \texttt{COUL\_SPL}$. Where $\mathcal{A}\left(\dot{\delta}\right)$ is a smooth function defined as -1 for negative $\dot{\delta}$ and +1 for positive $\dot{\delta}$. Linear extrapolation is applied in both splines for $\delta$ values not defined in the table.

- The approach coefficient $\beta$ is described as a linear function in $\delta$ : $\beta = \beta_c + \delta\beta_s$. Complete description of $\beta$ is obtained using parameters $\beta_c^+$ and $\beta_s^+$ for positive $\dot{\delta}$ and $\beta_c$ and $\beta_s$ for negative $\dot{\delta}$

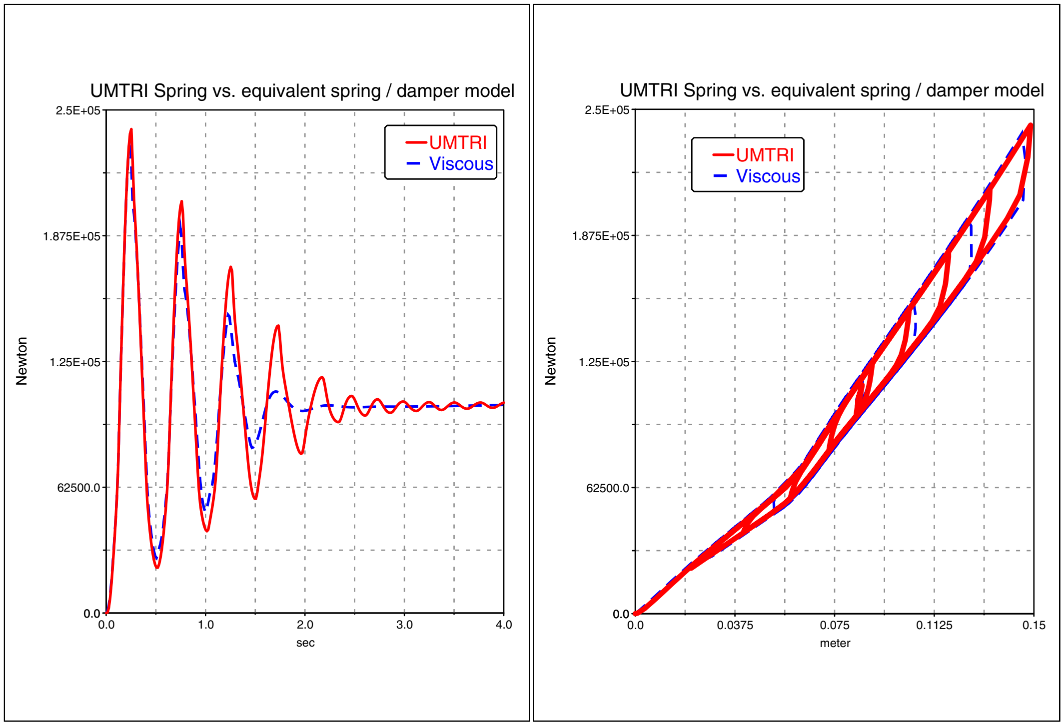


Figure 1.3: UMTRI based vs a viscous Spring-damper element

- The model is designed to run with a variable step numerical integrator. For this purpose, the delayed values of force and deflection $F_{i\ \ 1}$ and $\delta_{i\ \ 1}$ are defined using a first order differential equation. The time delay introduced in this differential equation (_TAU) must be tuned to a value small enough not to influence model behaviour.

A typical curve resulting from this description can be seen in figure 1.3.

Horizontal suspension forces are defined as a function of the vertical force. To reduce model complexity, $\beta$ is assumed a constant for the horizontal forces.

The vertical leafspring force is applied in ADAMS using a vector force with a non-zero z-component only. The implementation is separated in a *data definition* part (macro `umtri_dat_cre`) and an *equation definition* part (macro `umtri_vec_cre`).

The data definition macro defines design variables and splines describing the dynamic properties of a certain type of spring. The full name is generated by a user defined prefix (i.e. data_name = `LEAFDAT`) and a fixed suffix defined in the macros.

This method supports ease of use of models (i.e. in design studies or optimisation runs) with several identical suspension elements. By defining one single data definition block, the user can simultaneously change the properties of several suspension elements in the model.

The variables defined are:

| MacroName | Suffix | Default | Unit | Description |
|---|---|---|---|---|
| Lower_approach | _BCL | 1.0e-3 | - | Time constant $\beta_c^+$ to reach lower envelope |
| Lower_app_grad | _BSL | 1.0e-3 | - | Time constant $\beta_s^+$ F(Eps) to reach Low envelope |
| Upper_approach | _BCU | 1.0e-3 | - | Time constant $\beta_c$ to reach upper envelope |
| Upper_app_grad | _BSU | 1.0e-3 | - | Time constant $\beta_s$ F(Eps) to reach upper envelope |
| time_constant | _TAU | .001 | 1/time | Response time of force (tuning) |
| Eps_Init | _E_Init | 0.0 | length | Initial deformation of spring |
| Force_Init | _F_Init | 0.0 | force | Initial force in spring |
| V_Limit | _V_Limit | 1e-1 | velocity | velocity to define amount of viscous damping |
| V_Viscous | _V_Viscous | 1e-2 | velocity | Speed boundary for viscous behaviour |
| FStif_curve_(x,z) | _STIF_SPL | ... | force-length | Data storage for Stiffness force curve |
| FCoul_curve_(x,z) | _COUL_SPL | ... | force-length | Data storage for Coulomb force curve |

## 1.4 UMTRI model vs. viscous model

The variable with suffix `V_Viscous` is defined to introduce the possibility of a smooth transfer to a viscous model.

In the implementation, a *viscous* model is defined to represent a alternative force-deflection model using the same parameters for Stiffness and Coulomb force. In this viscous model, the variable `V_Limit` is used to define the amount of damping around the point of zero deformation velocity.

In the explanation, figure 1.4 is referred for the behaviour of a smooth step function and its first and second derivatives. The plot shows that the STEP5 function (fifth order polynomial smooth fit) must be scaled over 1.85 to obtain a unit slope around zero.

In the area $1.85$`V_limit`$< \dot{delta} < +1.85$`V_limit` the friction force is scaled from zero to full value (defined by the spline `COUL_SPL`) using a smooth differentiatable step function. As a result, at zero $\dot{\delta}$ the amount of linear damping equals $C_{lin} =$ `COUL_SPL/_V_Limit`.

This damping will be found when one performs a linear analysis on the model at a steady state situation (deformation velocity of suspension is zero). Thus, the user is enabled to vary the amount of damping in a linear analysis by the parameter for `V_Limit`.

Figure 1.5 is included to illustrate effect of non-linearity of the UMTRI based modelcompared to a viscous model in a very small amplitude. The results show the level of stiction introduced in the UMTRI model. Whereas the viscous model oscillates around the point of equilibrium, the UMTRI model approaches the final point from one side of the force-deflection curve.

In some models, it may occur that numerical problems are reduced at smaller amplitudes

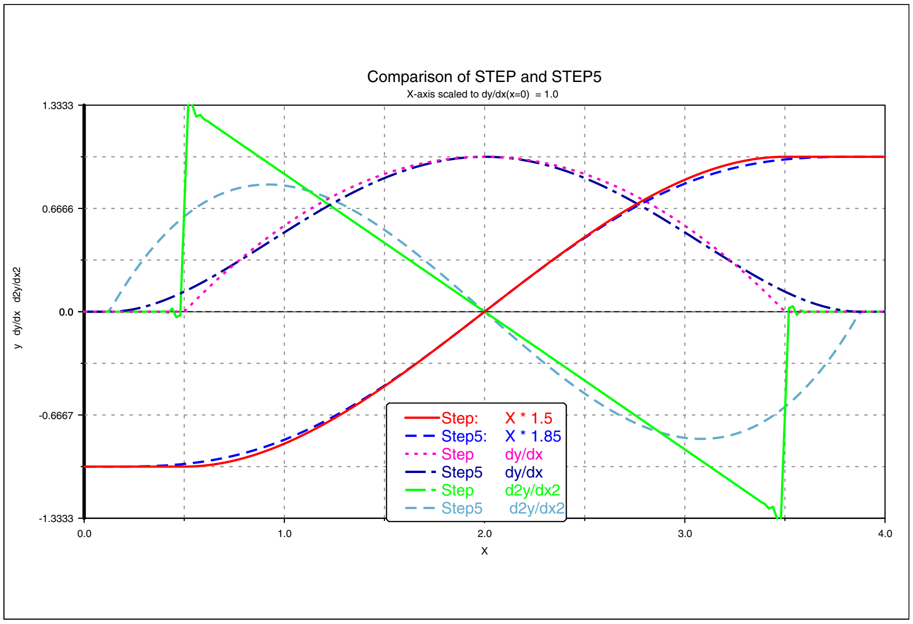


Figure 1.4: Curves of a fourth and fifth order polynomial smooth function

by allowing the force description to transfer from the UMTRI description to the viscous description. This use of the parameters is still under investigation and will result in a faster model in some specific excitation cases.

| | |
|---|---|
| $\|\dot{\delta}\| < \texttt{\_V\_Viscous} \quad 0.1\|\texttt{\_V\_Viscous}\|$ | Use a viscous model |
| $\texttt{\_V\_Viscous} \quad 0.1\|\texttt{\_V\_Viscous}\| < \|\dot{\delta}\| < \texttt{\_V\_Viscous}$ | Transient of both models |
| $\texttt{\_V\_Viscous} < \|\dot{\delta}\|$ | Use the UMTRI model |

In this method, the user ensures that:

- $\texttt{\_V\_Viscous} >> 0$ : the viscous model is always used
- $\texttt{\_V\_Viscous} \leq 0$ : the UMTRI model is always used

Finally, in the current implementation, the macros that define the model equations for the suspension also have parameters to specifically not define the UMTRI equations but only to use the viscous equations. This parameter (Version = Viscous) may be useful in cases where the exact behaviour of the leafspring dynamics is not (yet) relevant. In this cases, ignoring the UMTRI equations will increase the calculation speed of the simulation model with a factor 3 to 10. This is useful for instance when one is using the model on a testrig model to fit the parameters of the suspension.

## 1.5 Parameter determination with a testrig model

The vehicle model is placed on a modelled test rig. In this rig, both the front and rear axle are fully constrained to ground. At the location of the centre of gravity of the chassis two motions are defined moving the chassis relative to ground. A sinusoidal displacement of the motion is prescribed as a function of time with a period of 10 seconds. The amplitude of

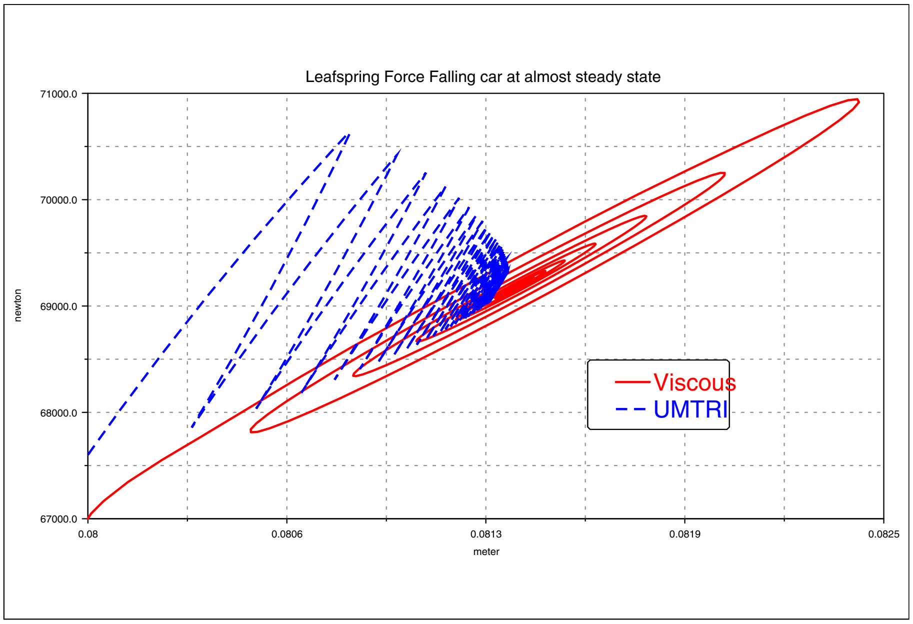


Figure 1.5: UMTRI vs. viscous model for small deflections

the motion is 0.025 m in both x- and y-direction. Using this method, the parameters for the suspension are fitted to available test results. In a first simulation, additional stiffness and damping in the friction element are set to zero. In this case, therefore, only the geometrical stiffnesses will be active. From the runs, half the force required to move the chassis is plotted against the deflection of the chassis-cg.

A simple approximation of the geometrical stiffness leads to:

$$C_{xy} = F_z / L_{chain} \tag{1.2}$$

Where $C_{xy}$ is the longitudinal and lateral stiffness of the suspension, $F_z$ the vertical load in each of the leafsprings and $L_{chain}$ the total length of the double chain parts connecting the leafspring part to the vehicle chassis.

This approximation is exact for small angles of the chain and for the case the chain parts are mounted vertically.

From drawings, the length of the double-chain parts is defined at 0.28 m. Mass and load of the vehicle are set to obtain vertical leafspring forces of 22.5 KN for an empty car (index $lo$) and 105 KN for a loaded car (index $hi$).

This leads to geometric stiffness values of $C_{lo,g} = 1.6e5$ [N/m] and $C_{hi,g} = 7.5e5$ [N/m] for deformation in both x- and y-direction.

## 1.6 Parameter investigations

Using double-chain elements of constant length, a number of geometry variations are performed to determine the effect on the longitudinal stiffness. Parameters varied are:

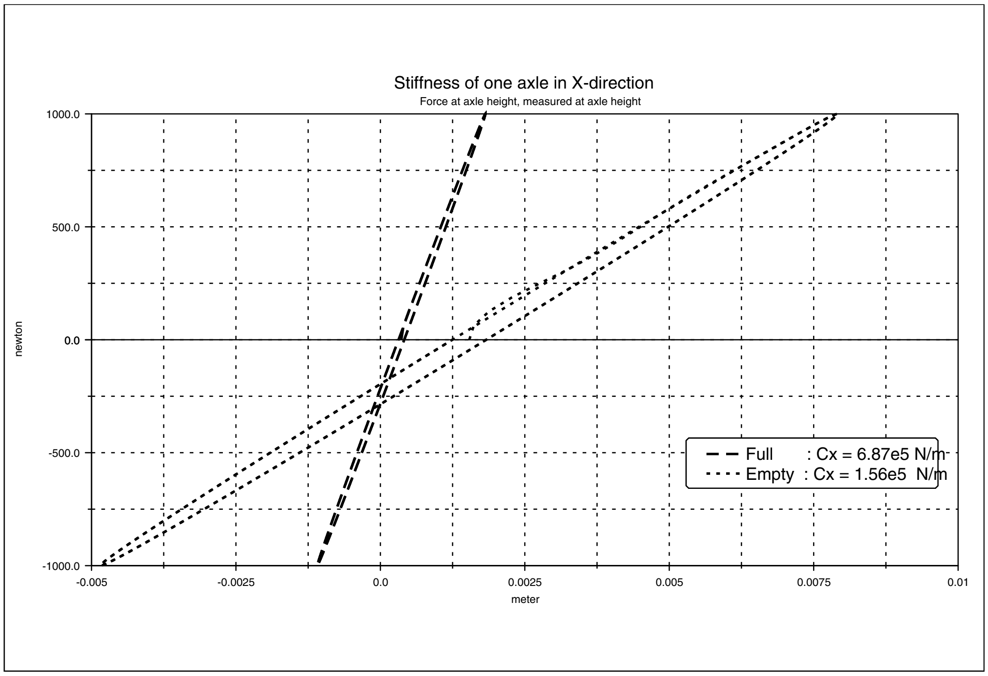


Figure 1.6: Geometric X-stiffness as measured in model

- $\beta_{ch}$: the angle of the double-chain parts about the lateral axis is varied by moving the lower point on the double-chains in x-direction,
- $L_{ls}$: the length of the leafspring is varied by moving top and bottom of the double-chain parts relative to the axis over the same distance.

From the results, the following conclusions can be drawn:

- For $L_{ls} = 1.1$ meter, an increase of $\beta_{ch}$ from 0.0 to to 38 $^\circ$ results in a 15 % decrease of the stiffness value. At larger angles, this effect is counter-acted by the geometrical stiffening effect. At 50 $^\circ$ the longitudinal stiffness shows a slight increase.
- At $L_{ls} = 2.1$ an increase of $\beta_{ch}$ results in an increase of the stiffness. At this value of $L_{ls}$, the angle of the leafspring about the lateral axis $\beta_{ls}$ has decreased compared to $L_{ls} = 1.1$. This $\beta_{ls}$ is due to the torque about the y-axis of the leafspring from the longitudinal force in the axle centre line. Due to the kinematics of the model, non-zero $\beta_{ls}$ can only occur at non-zero values of $\beta_{ch}$. At parallel chains ($\beta_{ch}$=0), the leafspring length doesnot influence $C_{x,g}$.
- This leads to the conclusion that, at $\beta_{ch}$ unequal to zero, the effect of rotation of the leafspring cannot be neglected. This effect is also taken into account in the method described by [2]. Similar effects can be expected from torsional bending of the leafspring.

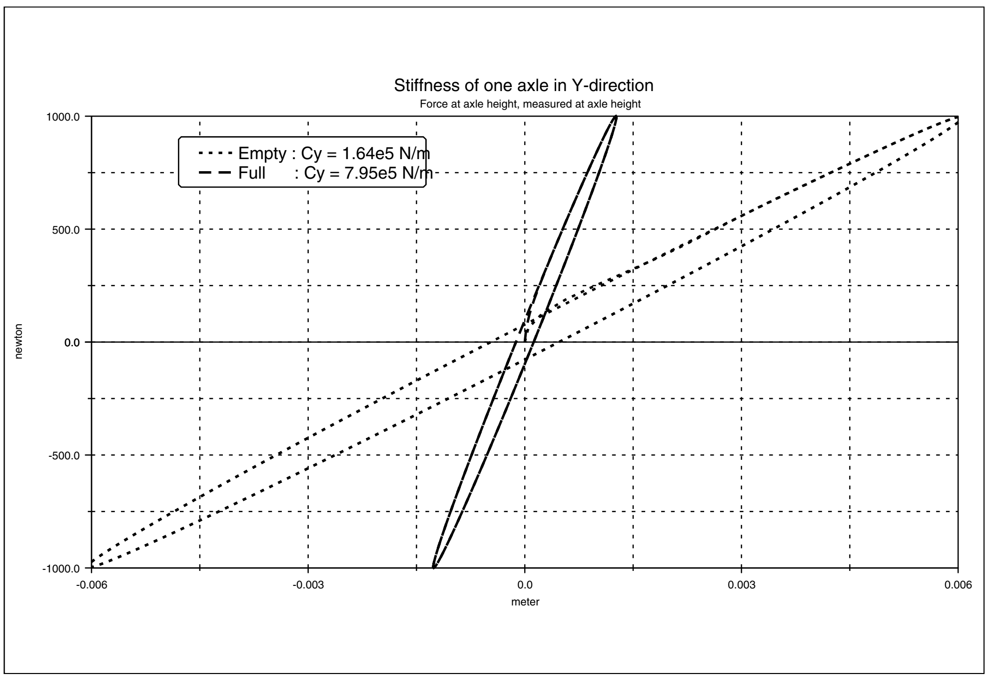


Figure 1.7: Geometric Y-stiffness as measured in model

# Chapter 2

# The complete vehicle model

The complete model of the freightcar is defined using a large amount of design variables. An overview of the model is shown in figure 2.1. In ADAMS terminology, design variables are used to define parametrisation of geometrical and other data in a model.

The complete parametrisation of the vehicle is performed in a strict hierarchical approach: the chassis frame is defined relative to the global frame, the reference points of the axle suspensions and the brake frames are defined relative to the vehicle chassis. The method is tuned to obtain a model that takes a minimum of redundant parameters.

In general due to this approach, the parameters can be divided in a number of groups:

1. overall vehicle parameters, mass, inertia values, amount of load, location and orientation of the vehicle.
2. Axle layout parameters such as distance between axles and wheels, location of the leafsprings, radius of the wheels. These parameters are still independent of the type of suspension used in the model (i.e. leaf spring vs. air spring).
3. Parameters to define inertia, topology and geometry of a single suspension element
4. parameters to define the force-deflection properties of the suspensions.
5. parameters to define locations and inertia for the brake frames connected to the axles.

In the current model, the four suspensions on the two axles are defined identically. The model can easily be enhanced to investigate differences between the suspension elements.

The different groups of parameters will be explained separately. The parameters to define the force-deflection properties of the suspensions have already been discussed in the previous chapter.

## 2.1 Overall vehicle parameters

The model is ideally defined in an inertial frame with the z-axis parallel to gravity. For the vehicle orientation, the local x-axis is parallel to the drive direction. Thus, the positive y-axis points to the right when looking in the drive direction.

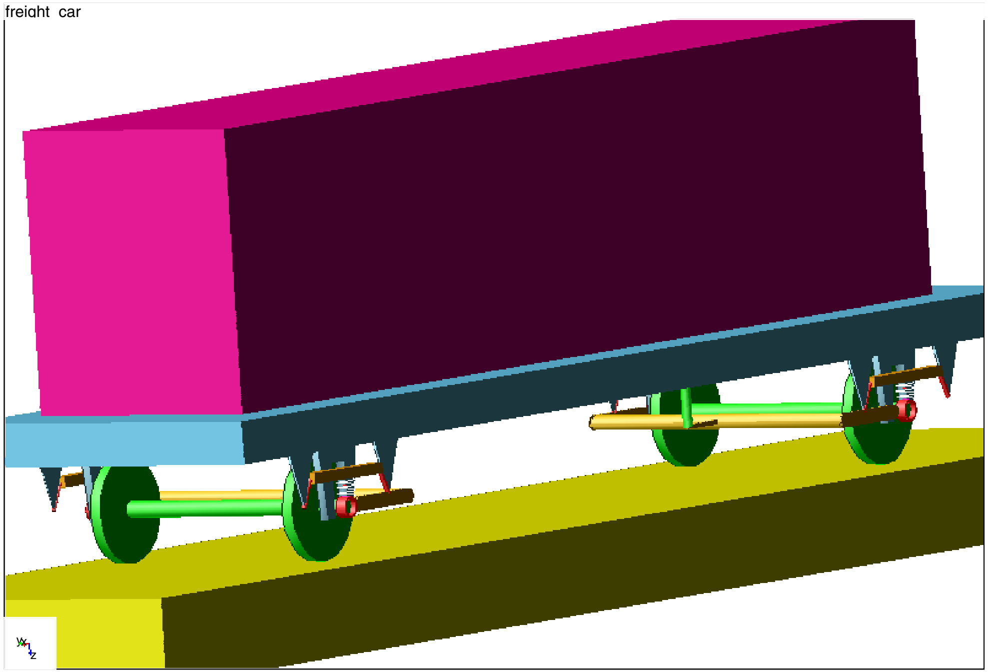


Figure 2.1: Full freightcar model topology

The reference point of the complete vehicle is defined at a global *marker* called `LPRF_car`. This point is in the centre point between front and rear axle on the ground surface. The initial location and orientation of the vehicle chassis (part `Chassis`) are fixed to this marker. Thus, the initial setting of the vehicle can be moved and rotated by replacing `LPRF_car`. Local coordinates of components of the chassis are therefore defined with respect to the chassis reference marker `LPRF_car`.

The basic geometry of the chassis is defined by two blocks, one for the empty vehicle and one for the load of the vehicle. The current amount of load is reflected in the inertia properties of the chassis and by visualisation of the local coordinate and size of the block for the load. Using variable `Car_Loaded`, the user can set the vehicle to be empty (`Car_Loaded`=0.0) or fully loaded (`Car_Loaded`=1.0). Intermediate values will be defined using linear interpolation.

For generality reasons, in this model, the main axes of inertia are parallel to the rail frames resulting in zero inertia coupling terms. The different parameters are placed in a table. Besides the names, values and explanations of the variables, the table also contains a column to denote the Function of the variable. In this column:

*Vis* represents a variable used for visulisation only,

Ine represents a variable used for inertia calculations,

Geo represents a geometrical variable used in the calculations,

Dep a variable that depends on othe variables,

Frc a variable defining force-deflection properties

| Name | Value | Function | Comment |
|---|---|---|---|
| L_Chassis | 16.0 | Vis | Length of the chassis shape |
| B_Chassis | 2.9 | Vis | Width of the chassis shape |
| Z_Chassis | -1.45 | Vis | local Z-coord. bottom of chassis block |
| Zcg_Chassis | -1.2 | Ine | local X-coord. of chassis cg |
| Xcg_Chassis | 0.31 | Ine | local Z-coord. of chassis cg |
| M_Chassis | 9150.0 | Ine | Mass of the empty vehicle |
| Ix_Chassis | 6500.0 | Ine | local X-rot. inertia empty vehicle |
| Iy_Chassis | 2.0e05 | Ine | local Y-rot. inertia empty vehicle |
| Iz_Chassis | 2.0e05 | Ine | local Z-rot. inertia empty vehicle |
| Car_Loaded | 1.0 | Ine | Switch to denote car load (0.0, ...1.0) |
| M_Load | 3.674e4 | Ine | Mass of the load |
| Ix_Load | 4.0e4 | Ine | local X-rot. inertia Load |
| Iy_Load | 2.0e5 | Ine | local Y-rot. inertia Load |
| Iz_Load | 2.0e5 | Ine | local Z-rot. inertia Load |
| Lx_Load | 13.0 | Vis | Length of full load |
| Ly_Load | 2.0 | Vis | Width of full load |
| Lz_Load | 2.5 | Vis | Height of full load |
| Dx_Cg_Load | 0.5 | Ine | Local x-coord. of cg of full load |
| Dz_Cg_Load | -0.6 | Ine | Local z-coord. of cg of full load |

## 2.2 Axle layout parameters

The vehicle has two axles, `Axle_1` at the front and `Axle_2` at the rear side. The axles are equally spaced about the reference point of the chassis. The visualisation of the axles doesnot contain the brake elements. The inertia properties therefore are defined disrespective of the geometrical shape of it. At the two ends of the axles parts axle box (called `Box_..`) parts are connected using revolute joints. The centre points of the axle boxes is placed at the y-coordinate of the leaf springs and at height `R_Wheel` above the ground surface. The volume of the axle boxes is used to define its inertial properties (its effect on overall vehicle behaviour being small).

## 2.3 Suspension element parameters

## 2.4 Brake frame parameters

| Name | Value | Function | Comment |
|---|---|---|---|
| A_Axles | 10.0 | Geo | Distance from front to rear axle |
| B0_Axle | 0.75 | Geo | Half distance between wheels |
| R_Axles | 7.5E-02 | Vis | Radius of axle cylinder |
| R_Wheel | 0.46 | Geo | Radius of wheel cylinder |
| B_Wheel | 0.1 | Vis | Width of wheel cylinder |
| M_Axle | 1345.0 | Ine | Mass of an axle |
| Ix_Axle | 820.0 | Ine | local X-rot. inertia axle |
| Iy_Axle | 150.0 | Ine | local Y-rot. inertia axle |
| Iz_Axle | 810.0 | Ine | local Z-rot. inertia axle |
| W_Axlebox | 0.12 | Ine | Width of an axle box |
| R_Axlebox | 0.15 | Ine | Radius of an axle box |

| Name | Value | Function | Comment |
|---|---|---|---|
| Y_Leafspring | 1.0 | Geo | Local y-coord. leafsprings |
| Z_Leafspring | -0.8 | Geo | Local z-coord. leafsprings |
| W_Leafspring | 0.1 | Vis | Width of the leafsprings |
| L_Leafspring | 1.2 | Geo | Length of the leafsprings |
| M_Leafspring | 100.0 | Ine | Mass of the leafsprings |
| Ix_Leafspring | 5.0 | Ine | local X-rot. inertia leafspring |
| Iyz_Leafspring | 30.0 | Ine | local Y- and Z-rot. inertia leafspring |
| X_Chain_Seat | 0.75 | Geo | Local X-coord. of the Bottom chain point |
| Y_Chain_Seat | 1.0 | Geo | Local Y-coord. of the Bottom chain point |
| Z_Chain_Seat | -0.51 | Geo | Local Z-coord. of the Bottom chain point |
| Z_Chain_Top | 0.0 | Geo | Local Z-coord. of the Top chain point |
| L_Chain | .... | Dep | Length of the chains |
| Z_Fric_XY | -0.2 | Geo | Z-coordinate where XY-friction force is applied |

| Name | Value | Function | Comment |
|---|---|---|---|
| X_Brake_Frame | -1.0 | Geo | X-coord. brake frame wrt. axle frame |
| Z_Brake_Frame | -0.5 | Geo | Z-coord. brake frame wrt. axle frame |
| R_Brake_Frame | 7.1E-02 | Ine | specific Radius of the brake frame |
| X_Brake_Link | -2.0 | Geo | X-coord. brake link wrt. axle frame |
| Z_Brake_Link | -1.0 | Geo | Z-coord. brake link wrt. axle frame |
| R_Brake_Link | 4.0E-02 | Ine | specific Radius of the brake link |
| L_Brake_Arm | 0.8 | Geo | |
| L_Brake_Clink | .... | Dep | |
| Cxz_Brakerubber | 5.3E+07 | Frc | Radial stiffness of brake frame rubber |
| Rxz_Brakerubber | 1.0E+04 | Frc | Radial damping of brake frame rubber |
| Cy_Brakerubber | 2.6E+06 | Frc | Axial stiffness of brake frame rubber |
| Ry_Brakerubber | 1000.0 | Frc | Axial damping of brake frame rubber |

# Bibliography

[1] "Measurement and representation of the Mechanical Properties of Truck Leaf Springs", P.S. Fancher, R.D. Ervin, C.C. MacAdam and C.B. Winkler, SAE Paper 800905

[2] 'Simulation des Fahrverhaltens zweiachsiger Güterwagen', Holger Lange, ZEV + DET. Glas. Ann. 122, Nr. 3, March 1998