

## PantoCat statement of method

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# PantoCat Statement of Method

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## Abstract

The Pantograph-Catenary Dynamic Interaction Analysis Program (`PantoCat`) addresses the need for a dynamic analysis code able to analyze models of the complete overhead energy collecting systems that include all mechanical details of the pantographs and the complete topology and structural details of the catenary. `PantoCat` is a code based on finite elements method, for the catenary, and on multibody dynamics methods, for the pantograph, integrated via a co-simulation procedure. A contact model based on a penalty formulation is selected to represent the pantograph-catenary interaction. `PantoCat` enables models of catenaries with multiple sections, including their overlap, the operation of multiple pantographs and the use of any complex loading of the catenary or pantograph mechanical elements including aerodynamic effects. The models of the pantograph and catenary are fully spatial being simulated in tangential or curved tracks, with or without irregularities and perturbations. User-friendly interfaces facilitate the construction of the models while the post-processing facilities provide all quantities of interest of the system response according to the norms and industrial requirements.

## 1. Introduction

`PantoCat` is a software that allows modeling and performing the dynamic simulation of the pantograph-catenary interaction. The program includes 3 modules that, being able to operate independently, are interfaced in the same user environment: `PantoCatFEM`, which is a Finite Element dynamic analysis code responsible for handling the catenary dynamics, `PantoCatMB` that is a Multibody Dynamics analysis code responsible to handle the dynamic simulation of the pantograph, and `PantoCatPro` that handles the models initialization and the results post-processing. The dynamic analysis code use methods defined in the time domain that handle all nonlinear effects present in the pantograph and catenary dynamics, such as the dropper slacking, friction, large rotations and nonlinearities of the pantograph system or the contact developing between components of the pantograph or between the pantograph registration strips and the catenary contact wires.

The pantograph dynamic analysis code was first developed as a tool for the fully detailed modelling and analysis of realistic pantographs, including general motion trajectories and control models [1][2]. This module, `PantoCatMB`, is featured to be interfaced with catenary modelling and analysis software via co-simulation approaches [3]. In the scope of the European project EUROPAC the software, with the name `Europacas-MB` that is the basic version of the current module, was fully tested and advanced modelling features specific to pantograph mechanical systems were made available. However, the use of `Europacas-MB` in the framework of system optimization or other specific task soon showed limitations, due to the interactive features of the companion software [4][5]. These limitations were overcome by the development of the catenary dynamic analysis module, `PantoCatFEM`, whose implementation addresses not only the common dynamic analysis in a co-simulation

environment with `PantoCatMB` but also design optimization, active control or any other environment in which it is necessary to run batches of simulations [6]. The advanced features of the software allow for the simulation of sophisticated and detailed models of the pantograph and catenary that have large sets of data, which are cumbersome, if not impossible, to manipulate by hand. The set of output data includes not only the contact forces but also all the kinematics of each mechanical element of the pantograph and catenary and the internal forces on both systems. In order to provide user friendly interfaces the `PantoCatPro` module was released. The complete set of modules is simply designated by `PantoCat`. Within the European project `PantoTRAIN` the software was fully tested and used for a wide range of scenarios, being its results compared with those of other software or with inline experimental data, when available, building not only the confidence on its quality but also identifying the required features for its use in practical applications [7]. Models for many of the European catenaries have been developed and analysed with `PantoCat`, being it able to handle stitch wire, composite or simple catenary types with one or more contact wires.

The models for the catenaries are developed using the finite element method for tangent and curved tracks being the analysis fully 3-dimensional. The pantograph models may be lumped mass or fully 3-dimensional multibody and their base motion is defined as if the pantograph is roof-mounted in a vehicle that follows the track for which the catenary is developed. With this approach it is possible to include pantograph motion perturbations originated either from the general vehicle railway dynamics or from any other source [8]. Due to the detailed pantograph modelling, wind forces acting on the pantograph elements can be included in the analysis [9]. Basically, any mechanical element existing in current pantograph construction technology can be included in the `PantoCatMB` models [10]. The catenary and pantograph dynamic analysis codes run in a co-simulation environment. It is possible to simulate single or multiple pantograph operations in catenaries that may include overlap sections [11][12]. All outputs considered in current regulations are standard in the `PantoCatPro` code. The current output of `PantoCat` include:

- Kinematics of the overhead system, i.e., displacement, velocity and acceleration of all nodes of the model
- Kinematics of all components of the pantograph, i.e., displacement, velocity and acceleration of the center of mass of all components or of any particular point
- Contact forces between the pantographs registration strips and the catenary contact wires, raw and filtered
- Position of the contact points in the contact wires and registration strips
- Joint reaction forces between the pantograph mechanical elements.
- Forces in the catenary droppers
- Uplifts of the catenary steady arms.
- Statistical parameters of the contact forces including average, standard deviation, maximum, minimum, number of contact losses, etc.
- Histograms of the contact forces
- PSD of the contact forces.
- RMS of the contact forces.
- Animation of the catenary and pantograph kinematics

The software `PantoCatFEM` is written in Matlab as well as the `PantoCatPro` pre and post processing, including its graphical user interface. The multibody code `PantoCatMB` is programmed in Fortran95 while the post-processed results are displayed in Microsoft Excel.

## 2. Methods applied in the benchmark

The `PantoCat` code is structured into two independent modules that handle the catenary dynamics, `PantoCatFEM` which is a finite element module programed in Matlab, and the pantograph, `PantoCatMB` which is a multibody dynamics module programed in Fortran. These modules run in a co-simulation computational environment being their interaction achieved via the contact force between the contact strips of the pantographs and the contact wire of the catenary for which a penalty contact force formulation is used.

### 2.1 Catenary Analysis Module and Models

The finite element method is used to describe the catenary dynamics. The equilibrium equations for the catenary structural system are assembled as [12]

$$\mathbf{M}\mathbf{a} + \mathbf{C}\mathbf{v} + \mathbf{K}\mathbf{x} = \mathbf{f} \quad (1)$$

where  $\mathbf{M}$ ,  $\mathbf{C}$  and  $\mathbf{K}$  are the finite element global mass, damping and stiffness matrices of the finite element model of the catenary. All catenary elements, contact and messenger wires are modelled by using Euler-Bernoulli beam elements. Due to the need to represent the high axial tension forces the beam finite element used for the messenger, stitch and contact wire, designated as element  $i$ , is written as

$$\mathbf{K}_i^e = \mathbf{K}_L^e + F \mathbf{K}_G^e \quad (2)$$

in which  $\mathbf{K}_L^e$  is the linear Euler-Bernoulli beam element,  $F$  is the axial tension and  $\mathbf{K}_G^e$  is the element geometric matrix. The droppers and the registration and steady arms are also modelled with the same beam element but disregarding the geometric stiffening. The mass of the gramps, attaching droppers to wires, are modelled here as lumped masses.

Proportional damping is used to evaluate the damping matrix of each finite element, i.e,  $\mathbf{C}^e = \alpha^e \mathbf{K}^e + \beta^e \mathbf{M}^e$  with  $\alpha^e$  and  $\beta^e$  being proportionality factors associated with each type of catenary element, such as dropper, messenger wire, stitch wire, etc. Alternatively, the global damping matrix is evaluated with the same proportionality factors associated to all structural elements, i.e,  $\mathbf{C} = \alpha \mathbf{K} + \beta \mathbf{M}$ .

The nodal displacements vector is  $\mathbf{x}$  while  $\mathbf{v}$  is the vector of nodal velocities,  $\mathbf{a}$  is the vector of nodal accelerations and  $\mathbf{f}$  is the force vector, written as

$$\mathbf{f} = \mathbf{f}_{(c)} + \mathbf{f}_{(a)} + \mathbf{f}_{(d)} \quad (3)$$

which contains the pantograph contact forces,  $\mathbf{f}_{(c)}$ , the aerodynamic forces,  $\mathbf{f}_{(a)}$ , and the dropper slacking compensating terms,  $\mathbf{f}_{(d)}$ . Equation (1) is solved for  $\mathbf{x}$  or for  $\mathbf{a}$  depending on the integration method used.

The integration of the nodal accelerations uses a Newmark family integration algorithm. The contact forces are evaluated for  $t + \Delta t$  based on the position and velocity predictions for the FE mesh and on the pantograph predicted position and velocity. The finite element mesh accelerations are calculated by

$$(\mathbf{M} + \gamma \Delta t \mathbf{C} + \beta \Delta t^2 \mathbf{K}) \mathbf{a}_{t+\Delta t} = \mathbf{f}_{t+\Delta t} - \mathbf{C} \tilde{\mathbf{v}}_{t+\Delta t} - \mathbf{K} \tilde{\mathbf{d}}_{t+\Delta t} \quad (4)$$

Predictions for new positions and velocities of the nodal coordinates of the linear finite element model of the catenary are found as

$$\tilde{\mathbf{d}}_{t+\Delta t} = \mathbf{d}_t + \Delta t \mathbf{v}_t + \frac{\Delta t^2}{2} (1 - 2\beta) \mathbf{a}_t \quad (5)$$

$$\tilde{\mathbf{v}}_{t+\Delta t} = \mathbf{v}_t + \Delta t(1-\gamma)\mathbf{a}_t. \quad (6)$$

Then, with the acceleration  $\mathbf{a}_{t+\Delta t}$  the positions and velocities of the finite elements at time  $t+\Delta t$  are corrected by

$$\mathbf{d}_{t+\Delta t} = \tilde{\mathbf{d}}_{t+\Delta t} + \beta \Delta t^2 \mathbf{a}_{t+\Delta t} \quad (7)$$

$$\mathbf{v}_{t+\Delta t} = \tilde{\mathbf{v}}_{t+\Delta t} + \gamma \Delta t \mathbf{a}_{t+\Delta t}. \quad (8)$$

In the current applications, to highspeed catenary dynamics, the coefficients used in the integration scheme depicted by Equations (4) through (8) are  $\beta = 1/4$  and  $\gamma = 1/2$ .

The droppers slacking are also corrected in each time step. Although the droppers perform as a bar during extension their stiffness during compression is either null or about  $1/100^{\text{th}}$  of the extension stiffness. As the droppers stiffness is included in the stiffness matrix  $\mathbf{K}$  as a bar element, anytime one of them is compressed such contribution for the catenary stiffness has to be removed, or modified. In order to keep the dynamic analysis linear the strategy pursued here is to compensate the contribution to the stiffness matrix by adding a force to vector  $\mathbf{f}$  equal to the bar compression force

$$\mathbf{f}_{(d)t+\Delta t} = \mathbf{K}_{dropper}^e \mathbf{B} \tilde{\mathbf{d}}_{t+\Delta t} \quad (9)$$

where the Boolean matrix  $\mathbf{B}$  simply maps the global nodal coordinates into the coordinates of the dropper element.

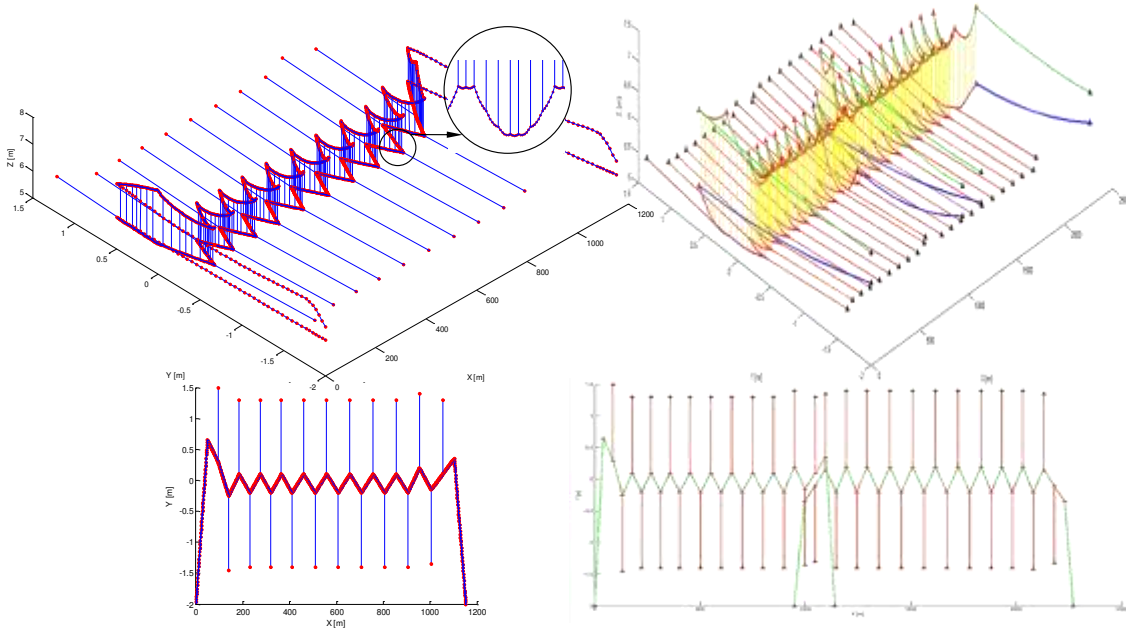


Figure 1: Finite element models of a catenary: single section with the sag highlighted and a plant view displaying the stagger; multiple sections and a plant view showing the overlap.

The correction procedure expressed by using Equations (5) through (9) and solving Equation (4) is repeated until convergence is reached for a given time step, i.e., until  $|\mathbf{d}_{t+\Delta t} - \tilde{\mathbf{d}}_{t+\Delta t}| < \varepsilon_d$  and  $|\mathbf{v}_{t+\Delta t} - \tilde{\mathbf{v}}_{t+\Delta t}| < \varepsilon_v$  being  $\varepsilon_d$  and  $\varepsilon_v$  user defined tolerances. At least 6 iterations must be allowed for the convergence process, although it is recommended that a maximum of 10 iterations is defined in order to prevent that residual compressive forces appear in the dropper elements.

The European project Pantotrain presented recommendations on the minimum number of elements to be used for the discretization of each structural component of the catenary finite element models [5]. The models studied here use 6 beam elements, between droppers, to represent the contact, messenger and stitch wires. For the droppers, steady arms and registration arms, if needed, a single beam element is used. In order to preserve the bar behaviour of the droppers under traction, steady arms and registration arms the moments of inertia of the beam elements used for their representation are kept to a minimum, lower than  $10^{-11}$ . In this form not only the numerical stability of the methods used in the solution of the equations of motion is ensured but also the use of these residual values do not represent any rotational stiffness of the bar components of the model.

The wave travelling velocity and the dissipative effects of the damping on the catenary are of crucial importance for its dynamic response. Therefore, the catenary model allows differential damping coefficients for its different structural components. The two entering and two exiting spans on each catenary section, where no contact with the pantograph exists, account for about 100 m of wire length in each end. Furthermore, the boundary conditions for the contact and messenger wire correspond to a spring-damper element. Besides increasing the realism of the catenary models, these two modelling features ensure that the reflection of the elastic wave does not influence the pantograph contact during the dynamic analysis, due to the lengths of the entering and exiting spans and that the elastic wave is attenuated, due to the boundary conditions.

## 2.2 Pantograph Analysis Methods and Models

A typical multibody model is defined as a collection of rigid or flexible bodies that have their relative motion constrained by kinematic joints and that are acted upon by external forces. The forces applied on the system components may be the result of springs, dampers, actuators or external applied forces describing gravitational, contact/impact or other forces. The pantograph models, being lumped mass or detailed 3-dimensional, may use any of the features available in multibody methodologies [14].

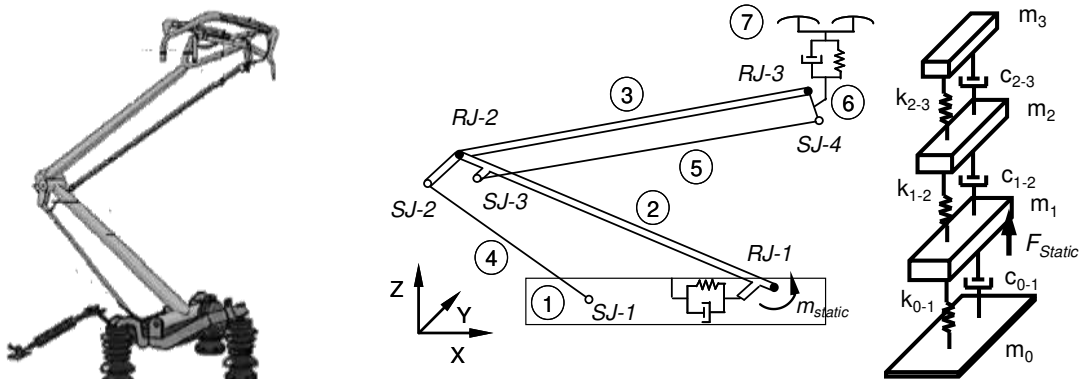


Figure 2: Typical pantograph and its multibody and lumped mass models.

The equations of motion for a constrained multibody system of rigid bodies are written as a system of differential algebraic equations, solved for  $\ddot{\mathbf{q}}$  and  $\boldsymbol{\lambda}$  as [14]

$$\begin{bmatrix} \mathbf{M} & \boldsymbol{\Phi}_q^T \\ \boldsymbol{\Phi}_q & \mathbf{0} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}}_r \\ \boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{g} \\ \boldsymbol{\gamma} \end{bmatrix} \quad (10)$$

where  $\mathbf{M}$  is the system mass matrix,  $\ddot{\mathbf{q}}$  is the vector that contains the state accelerations,  $\boldsymbol{\lambda}$  is the vector that contains  $m$  unknown Lagrange multipliers associated with  $m$

holonomic constraints,  $\mathbf{g}$  is the generalized force vector, which contains all external forces and moments, and  $\Phi_q^T \lambda = \mathbf{f}^{(c)}$  is the vector of constraint reaction forces.

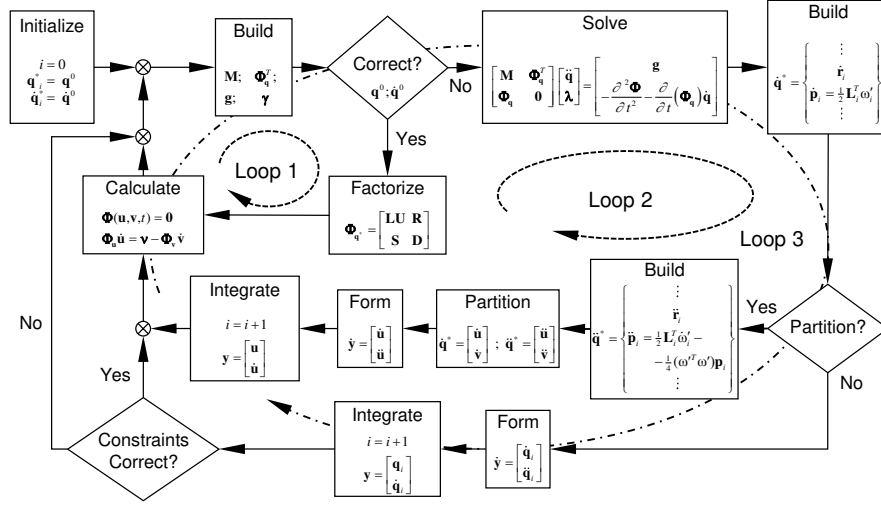


Figure 3: Flowchart with the forward dynamic analysis of a multibody system

In dynamic analysis, a unique solution is obtained when the constraint equations are considered simultaneously with the differential equations of motion and a proper set of initial conditions is specified. In each integration time step, the accelerations vector,  $\ddot{\mathbf{q}}$ , together with velocities vector,  $\dot{\mathbf{q}}$ , are integrated in order to obtain the system velocities and positions at the next time step. This procedure is repeated until the final time is reached, as depicted in Figure 3. An integration algorithm with variable time-step and integration order is used to solve the multibody equations of motion [15].

To initialize the solution process the positions and velocities of the mechanical elements must be compatible with the kinematic constraint equations. Such constraint fulfilment is ensured by loop1 of the solution scheme described in Figure 3. The set of differential algebraic equations of motion, Equation (10) does not use explicitly the position and velocity equations associated to the kinematic constraints. Consequently, when using the route labelled as loop 3 in Figure 3, the original constraint equations are rapidly violated due to the integration process. Thus, in order to stabilize or keep under control the constraints violation, Equation (10) is solved by using the Baumgarte stabilization method or the augmented Lagrangean formulation [16]. Due to the long simulations time typically required for pantograph-catenary interaction analysis, it is also necessary to use of a coordinate partition method, as implied in loop 2 of Figure 3, whenever the stabilization of the constraints is not possible otherwise.

### 2.3 Pantograph-Catenary Contact Force Model

The contact problem is treated with a penalty formulation in which the contact force is a function of the relative penetration between the two cylinders. The contact model used here includes hysteresis damping in the impact between bodies in the systems

$$F_N = K \delta^n \left[ 1 + \frac{3(1-e^2)}{4} \frac{\dot{\delta}}{\dot{\delta}^{(-)}} \right] \quad (11)$$

where  $K$  is the generalized stiffness contact,  $e$  is the restitution coefficient,  $\dot{\delta}$  is the relative penetration velocity and  $\dot{\delta}^{(-)}$  is the relative impact velocity.  $K$  can be obtained

from the Hertz contact theory as the external contact between two cylinders with perpendicular axis [17]. Although standard EN50318 specifies a value of  $K=50 \cdot 10^3$  N/m, the findings of the European Project PantoTRAIN [7] suggest that a more realistic value is  $K=200 \cdot 10^3$  N/m for current highspeed catenaries. Although all parameters used in the PantoCat code are user inputs, the parameters used in the contact model for the purpose of this benchmark are  $K=200 \cdot 10^3$  N/m,  $e=1$  and  $n=1$ .

## 2.4 Numerical Integration Procedures

Linear finite elements provide all modelling features for the development of the catenary dynamic analysis while (nonlinear) multibody mechanisms include all modelling features required for any type of pantograph model. In order to take the best advantage of the two different types of dynamic analysis a co-simulation environment between PantoCatFEM and PantoCatMB codes is implemented in the PantoCat program. The multibody code provides the finite element code with the positions and velocities of the pantographs registration strips. The finite element code calculates the contact force, using the contact model represented by Equation (11), and the location of the application points in the pantographs and catenary, using geometric interference functions. The contact forces are applied to the catenary, in the finite element code, and to the pantograph model, in the multibody code, as implied in Figure 4. Each code handles separately the equations of motion of each sub-system based on the shared force information.

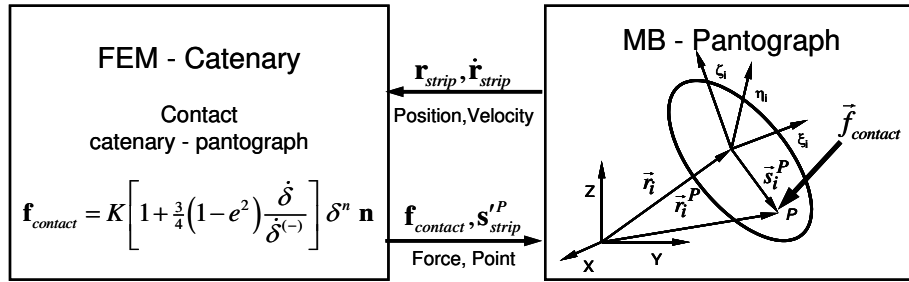


Figure 4: Co-simulation between a finite element and a multibody code

The key of the synchronization procedure between the multibody and finite element codes is the time integration step, ensuring the correct dynamic analysis of the pantograph-catenary system, including intermittent contact. The finite element integration code is of the Newmark family and has a constant time step that is small enough not only to assure the stability of the integration of the catenary but also to capture the initiation of the contact between the pantograph registration strip and the contact wire of the catenary. The multibody code uses a predictor-corrector integrator that can be an Adams-Bashforth or the Gear algorithm, with variable order and time-step [14]. The only restriction that is imposed in the integration algorithm of the multibody module is that its time step cannot exceed the time step of the finite element code. Both modules can start independently from each other, i.e., the catenary finite element model and the pantograph multibody model include the initial conditions for the start of the analysis expressed in terms of the initial positions and velocities of all components of the systems.

In order to ensure that the initiation and loss of contact is captured a maximum time step of  $10^{-3}$  s is allowed. Note that in some particular applications, such as those focusing analysis of irregularities and singularities in the contact wire, the maximum allowable time step may have to be reduced.

The co-simulation procedure was validated for a very wide number of scenarios of tangent tracks that included models of different catenary types and various lumped mass pantograph models at different operating velocities. All results obtained using the co-simulation approach are indistinguishable from the results obtained when the pantograph lumped mass models are analyzed with the finite element code without co-simulation.

## 2.5 Models and Analysis Initialization

The initialization of the catenary and pantograph models, i.e., the initial positions and velocities of the catenary finite element nodes and of the pantograph body components have different requirements due to the different methods used in the solution of their dynamics. The definition of the initial conditions for the catenary finite element model poses a different set of challenges. Recognizing that the geometric specification of the catenary geometry is defined for its static deformed state due to gravitational loading, the definition of the finite element nodal positions of the catenary model in its undeformed configuration are found before the dynamic analysis starts. For this purpose an optimization problem is defined to minimize the function that quantifies the distance between the static deformed geometry of the contact wire and its specified position, as

$$\begin{aligned} \min \mathcal{F}(\mathbf{b}) &= \sum_{i=1}^n (z_i^r - z_i)^2 \\ \text{s.t. } \mathbf{l} &\leq \Psi(\mathbf{b}) \leq \mathbf{u} \end{aligned} \quad (12)$$

in which the vector of design variables is  $\mathbf{b}=[z_1, z_2, \dots, z_n]^T$ , being  $z_i$  the position of node  $i$  of the messenger wire along the height direction, assuming the position of the dropper node in the contact wire as unchanged. The design variables are equivalent to the dropper length. Only the nodes of the messenger wire located at the droppers and steady arm are considered in vector  $\mathbf{b}$ . The reference height of each node  $i$ , i.e., the height of each node of the messenger wire at the steady arm, specified as input for the catenary geometry is defined as  $z_i^r$ . The constraints of the optimization problem include the orientations of steady arms and lengths of the droppers. Note that the catenary axial tensioning of the wires and the gravitational loading lead to catenary deformations require a nonlinear static analysis due to the large displacements and rotations of the finite element mesh. During the optimization problem a solution for a static analysis is required, every iteration, as if the deformation of the catenary model is linear. Although good solutions for the initial catenary positions have been obtained with the approach used here there is no guarantee that a good solution for the initial catenary configuration is always obtained.

In the case of the multibody pantograph model the initial velocities of all mechanical elements must be not only compatible with the forward velocity of the system along the track but also consistent with the kinematic constraints used [16]. In case of small errors the `PantoCatMB` module initialization, the code ensures their correction via loop 1 in Figure 3 so that the state variables ensure the kinematic consistency of the model.

Having the pantograph and catenary properly initialized, with the position of the pantograph contact strips located under the contact wire, in its close vicinity but without contact, the dynamic analysis starts. Some trial runs may be necessary before the pantograph lifting force/moment is fine tuned to ensure that the average contact force in the catenary-pantograph interface meets the required average force [18]. In any dynamic analysis of the system the dynamic response is not collected, or processed, during the first 2-3 spans after contact between the pantograph and catenary starts. Such transient response is discarded from any further analysis.

### 3. Additional methods available and not used in the benchmark

The `PantoCat` program has a number of analysis features that allows the study of pantograph and catenary models with characteristics that go well beyond the requirements of the benchmark. In terms of dynamics analysis `PantoCat` allows for:

- The application to the pantograph base of the kinematics of the train roof, including all disturbances resulting from track/wheel interaction, vehicle suspensions and operating conditions of the train.
- Generation of catenary geometries consistent with general track geometries including horizontal and vertical curves.
- Dynamics of the pantograph models that represent all of its constructive details including imperfections of the mechanical joints, dynamics of the pneumatic actuators, flexibility of the system components, fully spatial kinematics of all system components and any nonlinearity of the suspension mechanical elements.
- The simulation of catenary models with multiple sections, including realistic representations of the overlapping in the transition between sections.
- The introduction of active control by providing for the effect all necessary interfaces to test active control algorithms in realistic pantograph models.

Although without direct influence on the analysis features implemented in the `PantoCat` code, the ability to input catenary data in a neutral format widens the applicability of the program. By using the data format agreed in the `PantoTrain` project, implemented in an Excel database file, all geometric and material characteristics of the catenary are specified. This allows also the inclusion of any particular element and singularity in the catenary construction such as defects or dampers, as those being tested in some Japanese catenaries [19].

The implementation of other advanced features in `PantoCat`, such as the ability to include the flexibility of the pantograph mechanical elements in the dynamic analysis, is underway. Some of the initial results are available showing that in particular conditions there is an influence of the system flexibility in the quality of the contact [20].

### 4. Validation of the software

The norm EN50318 [21] provides two validation steps, defined as steps 1 and 2, for the assessment of a simulation method, one by comparison with other validated simulation methods and other by comparison with line tests. On step 1 simple reference models are provided for catenary and pantograph. For a successful validation it is necessary to obtain a given number of simulation output parameters within given ranges. Following step 1, which gives assurance on the precision and accuracy of the simulation tool, step 2 states a required accuracy of the method used concerning key parameters extracted from line tests. The `PantoCat` code has been validated according to both steps of the EN50318 norm.

In the first step of EN50318 reference models of the catenary and pantograph are defined being the simulation outputs deemed to fall within pre-defined ranges, for two different pantograph speeds. The pantograph is represented by a two stage lumped mass model while the catenary model is obtained by the basic geometry and material norm definitions, for a length of ten spans. These minimal modelling data requirements are short of the detailed definition of the catenary characteristics used in the benchmark, particularly with respect to the mechanical characteristics of the supports and structures. Another critical aspect concerns the norm specification of no damping on the catenary

model, which ultimately leads to the reflection of the wave propagation if neither energy absorbing boundary conditions nor entrance and exiting spans are used for the contact wire. The pantograph-catenary interaction is evaluated for pantograph speeds of 250 km/h and 300 km/h, being the results processed only for the 5<sup>th</sup> and 6<sup>th</sup> spans. The forces are filtered with a cut-off frequency of 20 Hz, fulfilling the norm specifications.

The statistical characteristics of the contact forces defined by the norm as requirement for the 1<sup>st</sup> step of the validation as well as the range of acceptance are shown in Table 1. All quantities required for the software acceptance fall, successfully, inside the ranges specified by EN50318.

speed	[km/h]	250		300	
		Norm	Model	Norm	Model
Mean contact force	[N]	110 - 120	114.6	110 - 120	115.5
Standard deviation	[N]	26 - 31	28.6	32 - 40	34.3
Statistical maximum	[N]	190 - 210	200.4	210 - 230	218.5
Statistical minimum	[N]	20 - 40	28.9	-5 - 20	12.5
Actual maximum	[N]	175 - 210	196.7	190 - 225	195.7
Actual minimum	[N]	50 - 75	52.6	30 - 55	34.7
Maximum uplift at support	[mm]	48 - 55	54	55 - 65	60
Percentage of loss of contact	[%]	0	0	0	0

Table 1: Statistical quantities required by EN50318, and range of acceptance, for the pantograph-catenary simulation software.

The 2<sup>nd</sup> step of the validation procedure consists on modelling existing catenary and pantograph and verifying the correlation of the model response with inline acquired data. The simulation results and acquired data are filtered similarly. The successful validation of the simulation tool, and implicitly of the models developed, requires a maximum deviation of 20% for the standard deviation of the contact force, maximum uplift at the supports and the vertical displacements of the contact point. For this type of validation it is necessary to access not only experimental inline measured data but also the modelling data for the catenary and pantograph of the existing system. The data concerning the LN2 catenary of the TGV Atlantique line and the Faiveley CX pantograph running at an operational speed of 300 km/h, was made available by SNCF for the PANTOTRAIN European project and used here. Two numerical simulations were produced for a pantograph to catenary contact force model, considering elastic contact only or including hysteresis damping.

	Experimental	Elastic	Damped
Maximum [N]	319.3	283.5	298.5
Minimum [N]	73.7	100.6	97.0
Amplitude [N]	245.5	182.9	201.5
Mean [N]	179.8	179.4	179.5
Standard Deviation [N]	44.3	44.4	46.9
Standard Deviation Accuracy [%]	-	0.08	5.92
Statistical Maximum [N]	312.8	312.5	320.3
Statistical Minimum [N]	46.9	46.3	38.6
Contact Loss [%]	0	0	0

Table 2: Statistical parameters for the experimental and simulated contact forces.

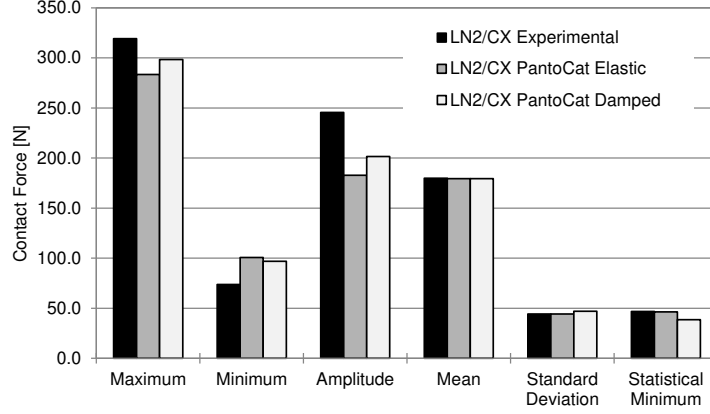


Figure 5: Statistics of the contact force for the experimental and simulated data.

The statistical analysis of the experimental and simulated contact forces are presented on Table 2 and Figure 5. The standard deviation accuracy also presented shows that its values, either for the elastic or damped contact models, are well inside the 20% required accuracy required by the norm for validation. For the elastic contact results the standard deviation is very close to the experimental results, however the deviation from the maximum and minimum contact force is more significant than the ones presented by damped contact result.

## 5. Considerations about the benchmark results

In general, for all important quantities in the study of the pantograph-catenary interaction problem, the dynamic response obtained with all software tested is similar. The few particular differences can be justified by modelling assumptions, sometimes forced by the software capabilities of analysis. The dynamic response obtained for the benchmark, with *PantoCat*, is analyzed here with reference to both modelling assumptions and software analysis capabilities.

### 5.1 Static analysis and initialization

The relevant results of the static analysis of the catenary concerns the position of the contact wire and the elasticity of the catenary, depicted respectively in Figures 4 and 5 of the benchmark general paper [22]. The position of contact wire at the regulation arms obtained by *PantoCat* is basically 5mm lower than what most of the remaining codes obtain, for the planar catenary, as seen in right side of Figure 6. This may be due to either the initialization procedure for the catenary geometry, to the finite element model used or to the model for the regulator arm. For the three-dimensional catenary model the height of the contact wire at the regulation arm, depicted in the left side of Figure 6 is coincident with that of most of the other codes. Note that the optimization procedure depicted by equation (12) is used to fine tune to position the steady arm. All rotations of the steady arms during the analysis are considered small, and consequently no large deviations on the steady arm positions are considered. The steady arms are pinned to a fixed element in one end and pinned to the contact wire in the other end in the models considered here.

The catenary elasticity can be evaluated with a pure static analysis or with a dynamic analysis in which the pantograph moves with a velocity low enough to disregard any dynamic effects on the response. For the load  $F=200\text{N}$  the two methods of identifying the

catenary elasticity, with PantoCat, lead to slightly different results, as shown in Figure 7. For a load  $F=100\text{N}$  there is no difference between the two methods of identifying the catenary flexibility, which suggests that the compression of the droppers, inexistent for the lower force, plays a role in the dynamical system due to the inertia forces.

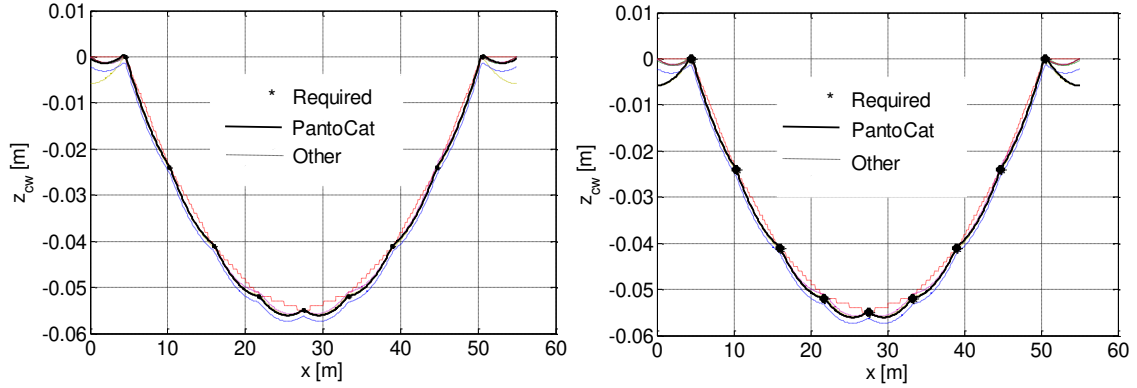


Figure 6: Position of the contact wire across the 6<sup>th</sup> span with 0.1 m reporting step for the 3D model (left) and 2D model (right).

The catenary geometric initialization is a critical step in the process of setting any catenary in general, and in this benchmark in particular. The gravitational forces tend to force the contact wire downwards while the axial tension in the wires tends to raise the messenger and contact wires. In most of the catenaries these two opposite trends almost cancel each other and the equilibrium position of the loaded catenary is not too far from the reference position. From the mechanical point of view this situation means that small variations on the unloaded geometry of the catenary allow for the complete system, upon loading, to reach a predefined geometry while assumptions for linearity of the system behaviour remain valid. However, for some particular catenary geometries, topologies and loadings the assumption of linearity of the catenary during loading may not be valid, due in particular to large rotation of the elements. In these cases, the identification of a preloaded geometry, which after loading is the specified configuration, may be difficult to find without using a nonlinear analysis.

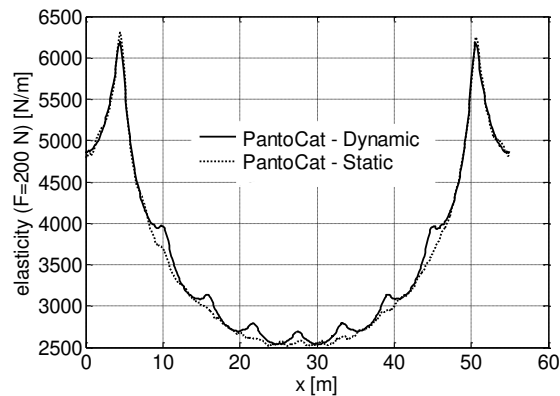


Figure 7: Elasticity of the catenary in the central span for a contact fore  $F=200\text{N}$ , evaluated with PantoCat with a static analysis and with a dynamic analysis using a very slow moving pantograph.

## 5.2 Dynamic analysis

In the catenary models used here the starting and exiting spans are not specified, neither is the type of boundary conditions used to fix the messenger and contact wires. This is a particularly sensitive issue for dynamic analysis at speeds for which the reflection of the catenary elastic deformation wave plays a role in the pantograph-catenary interaction. In order to avoid problems associated to the elastic wave reflection the catenary models used in *PantoCat* have two initial and two terminal spans similar to those of current highspeed lines in which no contact exists, i.e, the two spans account for a total length in the order of 100 m. Furthermore, the boundary conditions for the messenger and contact wires are defined with energy absorption. This is achieved by considering the elements that connect each of the wires to each attachment point as a spring and damper element in which the damper is used to fine tune the energy absorption.

A critical characteristic of the system that affects the pantograph-catenary interaction is the catenary damping. While in the benchmark the damper is completely characterized, in an existing catenary it has to be identified. Ambrosio et al. [6] actually shows that in a multiple pantograph operation scenario, depending on the catenary damping, the contact of the leading pantograph can be heavily affected by the rear, in the case of very lightly damped catenaries, or the inverse, in the case of more damped catenaries.

The contact law used for the pantograph-catenary interaction in this benchmark is not fixed. *PantoCat* allows the user to choose the parameters. According to the findings of the *PantoTRAIN* project [5] the recommended stiffness for the contact is  $F=50-200$  kN and the damping null for the higher stiffness. It is found that variations on the stiffness and damping of the contact law lead to variations in the results that may not be negligible.

The emphasis on the pantograph catenary interaction modelling issues is generally put on the catenary side and not the pantograph because the lumped mass pantograph models result from a system identification being their dynamic response, for the type of displacements observed, basically obtainable with a linear pantograph model. For larger head displacements, or when used in lines with curves, the lumped mass pantograph model cannot be used anymore, being a multibody approach to pantograph modelling unavoidable. However, clear specifications on how to model and use multibody pantographs do not exist yet. Preliminary studies show how the existing laboratory tests used for the identification of the lumped pantograph models can still be used to identify the selected unknown modelling parameters of multibody pantographs [23].

It must be referred that *PantoCat* presents good computational efficiency allied to the accuracy demonstrated in the benchmark. In order to measure such efficiency, for the cases simulated in this benchmark and other cases simulated throughout the life of *PantoCat*, each 1s of real time takes about 11s of computer time for the single pantograph case and 16 s for the two pantograph scenarios in a computer equipped with the Intel i7 2600K, or using another measure, a simulation of a pantograph running at 300 km/h in 1 km of track takes about 132s of computer time.

Finally it must be referred that the benchmark now developed does not allow to understand the importance, or lack of it, of using spatial catenary models in place of the planar models. That is mainly because the pantograph model used in the benchmark is one dimensional, i.e., its mobility is only in the vertical direction and consequently, even if excited in any other direction, its dynamic response is only in the direction it is modeled. Physically realistic pantographs models use a fully three-dimensional representation of all mechanical components being them allowed to develop a spatial motion. The understanding of the differences between planar and spatial catenary models is clarified when interacting with fully three-dimensional pantograph models.

## 6. Conclusions

The PantoCat code is a dynamic analysis software that accepts fully three dimensional finite element models of catenaries and spatial nonlinear multibody models of pantographs. A co-simulation procedure, via the contact force model is used to couple the simulation of both models. By using the finite element method for the dynamic analysis of the catenary and a multibody methodology for the dynamics of the pantograph PantoCat can take advantage of the best features of each method and handle detailed models with complex topologies and geometries. Although many of the features of the software are not used in this benchmark, such as the curved track with curved catenaries, the multibody pantographs or the perturbations of the pantograph trajectory, all features required exist by nature in the original PantoCat. The interaction force between the pantograph and catenary very sensitive to the catenary geometry, effort to develop a more advanced initialization procedure for the geometry of the catenary and for the initial conditions of the pantograph is underway. The benchmark in which the code PantoCat is used considers models for the catenary and pantograph not only more detailed than those used in EN50318 but much closer to the current highspeed railway lines and to the modeling capabilities of modern pantograph-catenary dynamic analysis codes. Among the catenary characteristics not specified in this benchmark that have influence in some of the results it must be emphasized the characteristics of the boundaries of the contact and messenger wires and/or the geometry of the first and last spans on the catenary. Another aspect of the benchmark that must be taken into account is the impossibility to use the dynamic response in the first 2-3 spans after the effective contact between the pantograph head and the catenary contact wire takes place, due to the need to raise the pantograph head until line contact is achieved at a correct contact force.

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