Optimization of high-speed railway pantographs for improving pantograph-catenary contact

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Abstract A crucial system for the operation of high-speed trains is the pantograph catenary interface as it is the sole responsible to deliver electrical power to the train. Being the catenary a stationary system with a long lifespan it is also less likely to be redesigned and upgraded than the pantographs that fit the train vehicles. This letter proposes an optimization procedure for the improvement of the contact quality between the pantograph and the catenary solely based on the redesign of the pantograph head suspension characteristics. A pantograph model is defined and validated against experimental dynamic characteristics of existing pantographs. An optimization strategy based on the use of a global optimization method, to find the vicinity of the optimal solution, followed by the use of a deterministic optimization algorithm, to fine tune the optimal solution, is applied here. The spring stiffness, damping characteristics and bow mass are the design variables used for the pantograph optimization. The objective of the optimal problem is the minimization of the standard deviation of the contact force history, which is the most important quantity to define the contact quality. The pantograph head suspension characteristics are allowed to vary within technological realistic limits. It is found that current high-speed railway pantographs have a limited potential for mechanical improvements, not exceeding 10%–15% on the decrease of the standard deviation of the contact force.

Keywords high-speed, railway, pantograph

A limitation on the velocity of high-speed trains concerns the ability to supply the proper amount of energy required to run the engines, through the catenary-pantograph interface.1 Due to the loss of contact not only the energy supply is interrupted but also arcing between the collector bow of the pantograph and the contact wire of the catenary occurs leading to their deterioration of the functional conditions of the two systems. The increase of the average contact force would improve the energy collecting capabilities but would also lead to a rapid wear of the registration strip of the pantograph and of the contact wire.2 The topology of a pantograph must address three stages of its operation: lift the pantograph head to contact wire height and compensate for spans with lower catenary heights, generally with frequencies of 1–2 Hz; handle the displacements with middle range frequencies associated to steady-arms passage, i.e., up to 10 Hz; deal with the higher frequency but low amplitude events.3 Typically the pantograph head, with its suspension, is responsible to handle the high-frequency excitations while the lower stage, with the pneumatic bellow, deal with the low frequency.

A large majority of the pantographs in operation have been developed with particular catenary systems in mind, forming national pantograph-catenary pairs such as the CX–LN2, which prevails in the French networks, DSA380–Re330, common in the German high-speed lines, and ATR95–C270, used in Italy. However, present trends for interoperability result in new pantograph design requirements allowing operations in different catenary systems. It is accepted that the improvement of the current collection capabilities requires lighter bows, which in turn suggests the use of new materials and construction concepts and that the head suspension is adjusted accordingly.4 The need for the pantographs to have low aerodynamic drag and noise emission and to be compatible for cross-border operation sets some of the development directions for improved collectors.5

The norm EN50367 c) specifies the technical criteria for the interaction between pantograph and overhead line. The experimental data on the contact force allows obtaining the most important parameters required to evaluate the quality of the contact. The norm EN50367 c) specifies the following thresholds for pantograph acceptance.

1) Mean contact force $F_m$  
2) Standard deviation $\sigma < 0.3 F_m$  
3) Max contact force $F_{max} < 350$ N  
4) Max CW uplift at steady-arm $d_{up} \leq 120$ mm  
5) Max pantograph vertical amplitude $\Delta_z \leq 80$ mm  
6) Percentage of real arcing $NQ \leq 0.2$%

Different numerical approaches exist in the literature to handle the pantograph-catenary interaction. References 6 and 7 propose the use of a finite-differences method to describe the catenary and a multibody approach for the pantograph. The catenary used is planar and a co-simulation procedure controls the coordination between the dynamic simulation of the pantograph and catenary dynamics. More recently other au-
thors proposed the use of the finite element method to deal with the pantograph and catenary dynamics. Al- though the catenary models used are fully spatial, only lumped mass pantograph models can be used with these approaches. Ambrosio et al. proposed the representation of the catenary spatial dynamics using the finite element method while the pantograph is handled by a general multibody dynamics approach. A co-simulation procedure is used to ensure the synchronization between the finite elements and the multibody part of the code, pretty much in line with other co-simulation approaches used in different areas of computational mechanics.

In this work a realistic model for a high-speed pantograph model is presented, being its identification discussed. An optimization procedure, based on a genetic algorithm, is used to study the improvement of the pantograph dynamics to improve the pantograph-catenary interaction, described in terms of the quantities identified in the norm EN50367. The catenary model is developed in finite elements and the pantograph model dynamics is studied with a multibody dynamics code being their interaction done through a co-simulation approach using a proper penalty contact formulation. All analysis are developed accounting only for the contact interaction and disregarding the aerodynamic forces according to the European norm for experimental acquisition of the contact force. The potential for pantograph enhancement is, finally, discussed.

The motion of the catenary is characterized by small rotations and small deformations, in which the only nonlinear effect is the slacking of the droppers. The axial tension on the contact, stitch and messenger wire is constant and cannot be neglected in the analysis. All catenary elements, contact and messenger wires are modeled by using Euler–Bernoulli beam elements that include axial tension, in particular to represent the mes-

The contact between the registration strip of the pantograph and the contact wire of the catenary, from the contact mechanics point of view, consists in the contact of a cylinder made of copper with a flat surface made of carbon. The contact problem can be treated by a penalty formulation where the contact force is defined in function of the relative penetration between the two cylinders. In this work, the Hertzian type contact force including internal damping is used. A suitable representation of the contact model is written as

\[
M \ddot{q} + C \dot{q} + K q = f,
\]

where \( M \), \( C \) and \( K \) are the finite element global mass, damping and stiffness matrices. Proportional damping is used to evaluate the global damping matrix, i.e., \( C = \alpha K + \beta M \) with \( \alpha \) and \( \beta \) being suitable proportionality factors. The nodsal displacements vector is \( x \) while \( v \) is the vector of nodal velocities, \( a \) is the vector of nodal accelerations and \( f \) is the force vector given by

\[
f = f_c + f_a + f_d,
\]

being \( f_c \) the pantograph contact forces, \( f_a \) the aerodynamic forces, and \( f_d \) the dropper slacking compensating terms. For typical catenary finite element models the Newmark family of integration algorithms are suitable methods for the integration of the equations of motion.

The mechanical system that guarantees the required characteristics of the trajectory of the pantograph head during rising is generally made up by a four-bar linkage for the lower stage and another four-bar linkage for the upper stage. Another linkage between the head and the upper stage of the pantograph ensures that the bow is always leveled. In order to control the raise of the pantograph one bar of the lower four-bar linkage is actuated upon by a pneumatic actuator. Regardless of using multibody or lumped mass pantograph models, the equations of motion for a constrained multibody system (MBS) of rigid bodies, such as a pantograph, are written as

\[
\begin{bmatrix}
M & \Phi_q^T \\
\Phi_q & 0
\end{bmatrix}
\begin{bmatrix}
\ddot{q} \\
\lambda
\end{bmatrix}
=
\begin{bmatrix}
g \\
\gamma
\end{bmatrix},
\]

where \( M \) is the system mass matrix, \( \ddot{q} \) is the vector with the state accelerations, \( g \) is the generalized force vector, which contains all external forces and moments, including the contact force that also appears in the catenary loading vector depicted by Eq. (2). \( \lambda \) is the vector that contains \( m \) unknown Lagrange multipliers associated with \( m \) constraints, for which \( \Phi_q \) is the Jacobian matrix and \( \gamma \) is the right side of acceleration equations containing the terms that are function of velocity, position and time.

In dynamic analysis, a unique solution is obtained when the constraint equations are considered simultaneously with the differential equations of motion with proper set of initial conditions, i.e., a set of initial conditions that fulfills the position and velocity constraint equations. In each integration time step, the accelerations vector, \( \ddot{q} \), together with velocities vector, \( \dot{q} \), are integrated in order to obtain the system velocities and positions at the next time step. This procedure is repeated up to final time will be reached. Due to the long simulations time typically required for pantograph-catenary interaction analysis, it is also necessary to implement constraint violations correction methods, or even the use of the coordinate partition method for such purpose.

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\[
f_c = K \delta^n \left[ 1 + \frac{3(1 - e^2)}{4} \frac{\dot{\delta}}{\delta^{(-)}} \right],
\]

where \( f_c \) is the magnitude of vector \( f_c \) in Eq. (2), the penalty term \( K \) is the generalized contact stiffness, \( e \) is the restitution coefficient, \( \dot{\delta} \) is the relative penetration velocity and \( \delta^{(-)} \) is the relative impact velocity. The proportionality factor \( K \) is obtained from the Hertz contact theory as the external contact between two cylinders having their axis perpendicular to each other.
When using the same code to simulate both catenary and pantograph models the type of dynamic analysis must be the same for both, i.e., if the catenary is modeled with linear finite elements the pantograph dynamics must also be linear. This is not compatible with the use of more general, and realistic, multibody pantograph models for which large rotations may exist or that may follow curved tracks. In order to take advantage of the more adequate type of dynamic formulations to be used for catenary and for pantograph a co-simulation between finite elements and multibody approaches is used here.

Pantograph models used for the study of the pantograph-catenary interaction can be of two types: lumped mass or multibody models. Due to the fact that they can be studied in common linear finite element codes, also used for the analysis of the catenary, the lumped mass models are the most common appearing in the literature. However, the lumped mass pantograph models are mathematical abstractions that need to be identified in laboratory tests and, for which, almost none of the model parameters have any physical significance. For a typical three-stage lumped mass pantograph model used for high-speed applications, as the one depicted in Fig. 1, only the top mass, stiffness and damping coefficients can be kept similar to the pantograph head mass and suspension characteristics.

Physical laboratories, as that one existing at Politecnico di Milano, can be used to measure the experimental response of a physical prototype of a pantograph in terms of frequency response functions (FRF). With such FRF available, the lumped mass model of the pantograph is identified by setting a virtual testing laboratory, as the one depicted in Fig. 2, in which the pantograph model is simulated. The model parameters are tuned such that the model FRF match those measured experimentally.

The multibody model of the test rig is composed of the excitation system and the pantograph model, connected via a rigid joint. This setup allows to excite the head of the pantograph with the same contact force, frequency and amplitude used in the physical laboratories and to collect the accelerations in any point of the pantographs to build the FRF. The time history of the prescribed displacement of the excitation bar, $z(t)$, generated from the test data supplied by the physical testing laboratories, starts with the bar being raised or lowered to the testing position followed by a sinusoidal excitation with prescribed frequency and amplitude. The flowchart of the virtual laboratory is shown in Fig. 3.

To ensure that the virtual laboratory tests are performed in a suitable manner, the number of cycles of excitation at each frequency must be such that a stationary behavior of the system is obtained before its dynamic response is considered for the evaluation of the FRF. The process is shown in Fig. 4.

By setting the head mass and suspension characteristics of the lumped mass pantograph model to the values obtained from measuring the component and identifying all other parameters of the model, the validated three-stage pantograph model depicted in Table 1 is obtained.

The FRF corresponding to the pantograph head mass, $m_3$, upper frame, mass $m_2$, and pantograph knee,
The optimization problem associated to the improvement of the pantograph for the catenary-pantograph interaction is defined as

$$
\text{minimize } F_0(u_i)
$$

subject to:

$$
\begin{align*}
\{ f_j(u_i) &= 0, & j = 1, 2, \cdots, n_{ec}, \\
\{ f_j(u_i) &\geq 0, & j = (n_{ec} + 1), (n_{ec} + 2), \cdots, n_{tc}, \\
\begin{cases}
&u_i^{\text{lower}} \leq u_i \leq u_i^{\text{upper}}, \\
&i = 1, 2, \cdots, n_{sv},
\end{cases}
\}
\end{align*}
$$

where the objective function is denoted by $F_0$, $u_i$ are the design variables, $n_{ec}$, $n_{tc}$ and $n_{sv}$ are the number of equality constraints, inequality constraints and bounded design variables. In this work the standard deviation of the contact force is used as the measure to minimize. The rationale for this choice is that the ultimate goal is to minimize the mean pantograph contact force, which is nowadays defined by the TSI. Such mean contact force can be controlled using the lower stage pneumatic actuator provided that the standard deviation of the contact force remains equal or below 30% of the mean force. Therefore, minimizing the standard deviation allows reducing the mean contact force, if permitted by regulations, and ensures that the pantograph performs with a better safety margin, as the homologation limit of $\sigma_{\text{std}} < 0.3 F_{\text{mean}}$ is more easily achieved. Therefore,

$$
F_0(u_i) = \sigma_{\text{std}}.
$$

The design variables used in the optimization process are the only ones that have physical meaning in the lumped mass pantograph models, i.e., the mass, stiffness and damping of the upper stage of the three-stage models used. With reference to Fig. 1, the vector of design variables is, therefore,

$$
u = \begin{bmatrix} m_3 & k_{2-3} & c_{2-3} \end{bmatrix}^T.
$$

The constraints used in the optimization process are the side constraints of the design variables and any functional constraint. The technological constraints are expressed here as side constraints being

$$
0.8 m_{\text{ref}} \leq m_3 \leq 1.2 m_{\text{ref}},
$$
0.8 k_{ref} \leq k_{2-3} \leq 1.2 k_{ref},
0.1 c_{ref} \leq c_{2-3} \leq 100 c_{ref}.

(8)

A constraint maintained in the optimization process is the mean contact force $F_{\text{mean}} = 150$ N. This constraint recognizes the fact that by reducing the mean contact force there are variations on the standard deviation, which is generally reduced. Therefore the constraint ensures that the reduction of the standard deviation is at the cost of the functional characteristics of the pantograph and not of the operational conditions.

The choice of the optimization methods suitable for the solution of the optimal problem is of crucial importance. The problem of the characterization of the optimal pantograph is non-convex and subjected to large numerical noise. Besides these factors, there is no assurance that the design space is continuous. All these factors lead to the existence not only of local minima but also to convergence difficulties when using deterministic optimization algorithms, i.e., algorithms in which the search is based on the definition of gradients. Therefore, global optimization algorithms are selected to address the problem, among which a genetic algorithm is the choice.

The genetic optimization algorithm used here is available in Matlab. All default values of the ge function are selected with the exception of the number of individuals (i.e., the number of sets $u$ for which the function evaluation is done), which is $N_{\text{individuals}} = 60$, the maximum number of generations, which is $N_{\text{generations}} = 10$ and the maximum number of generations to run without having an improvement above the default limit, which is $S_{\text{tall}} L_{\text{limit}} = 5$. In order to ensure any feasible improvement of the best design detected by the genetic algorithm a deterministic algorithm is used to find the minimum close to such point. The sequential quadratic programming (SQP) method is used for the purpose.

The lumped mass pantograph model, depicted in Fig. 1, and with the data provided in Table 1, is the reference model for the optimization. The pantograph runs on a high-speed catenary of the simple type. The design space of the pantograph is searched by the genetic optimization algorithm with 7 generations, in which 420 analysis are performed. The result is further improved by the SQP optimization taking 4 iterations more to reach the best pantograph design.

The search of the design space is depicted in Fig. 6, where each point is a different pantograph design. It can be observed that the optimal pantograph has a bow mass and head suspension damping as low as the design space allows while the suspension stiffness increases reaching its maximum allowed limit.

The performance of the genetic algorithm is depicted in Fig. 7, in which the values of the worst, mean and best standard deviations for each generation are represented. It is noticed that the best design is obtained after 5 generations while the average values of the objective function decrease in every generation. This indicates that the best individuals are already included in
the last generation and that, most likely, they can not be improved.

The resulting model for the optimal pantograph is depicted in Fig. 8, together with the model of the reference pantograph, i.e., the model that results from the experimental identification of an existing prototype. The optimal pantograph performance on the reference catenary exhibits a standard deviation of the contact force $\sigma_{\text{std}} = 37$ N, which is an improvement of 11% of its contact quality.

To have a better understanding of the behavior of the optimal pantograph in the actual interaction with the catenary it is relevant to analyze the contact force characteristics that are required by the regulation. Figure 9 presents the time history of the contact force in selected spans of the catenary. It is observed that the general behavior of the reference and optimized pantographs is similar, except for the peak forces, in which the optimized pantograph exhibits lower maxima.

The statistical characteristics of the contact forces are depicted in Fig. 10 in terms of the maximum and minimum contact forces, amplitude, standard deviation and statistical minimum force. The improvement on the pantograph performance can be observed not only on the decrease of the standard deviation but also on the reduction of the maximum force and increase of the minimum contact force.

Another relevant aspect of the pantograph performance concerns the RMS-value of the contact forces at relevant frequency ranges, in particular for the first harmonic, span passing frequencies and dropper passing frequencies. It is observed in Fig. 11 that the RMS-values of the optimal pantograph are reduced in all frequency ranges for the optimized pantograph.

Another aspect of the pantograph catenary interaction that requires attention is the maximum uplift on the catenary caused by the pantograph contact. Figure 12 shows the uplift in a selected support of an intermediate span of the catenary for the reference and optimal pantographs. It is observed that the changes in the pantograph have no effect on the steady-arm uplift. The same trend is observed in all other steady arms of the catenary, being 6 cm the maximum uplift value observed anywhere in the catenary, which is far from the 10 cm limit imposed by the infrastructure owner.
The results of the pantograph optimization show that the room for improvement of the existing equipment performance on the standard catenaries in which they are operated is rather limited. The potential improvement on a standard high-speed pantograph performance is of about 11%. The results not only show that pantograph developers do a background work to find the best tuning for their performance to the national catenaries where they are expected to be operated. However, no conclusions can be drawn on the cross-border operation of the pantographs as they have not been object of this study. The use of the optimization tools can provide valuable information on the changes required in the pantographs, from the mechanical point of view, for improved quality of operation across border in the same form that the performance of the current catenary-pantograph pair was demonstrated. Further work is required to demonstrate that the optimal pantograph systems are in fact improved for different operating conditions in the networks in which they are accepted, using experimental geometric data of the catenaries in particular.

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