Mechanism and Machine Theory 45 (2010) 1035–1049

Contents lists available at ScienceDirect

Mechanism and Machine Theory

journal homepage: www.elsevier.com/locate/mechmt

Train kinematics for the design of railway vehicle components

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ARTICLE INFO

Article history:
Received 3 March 2008
received in revised form 13 April 2010
accepted 21 April 2010
Available online 13 May 2010

Keywords:
Cartesian coordinates
Guiding constraints
Gangway design
Train working envelops

ABSTRACT

The kinematic analysis of railway vehicles and trainsets allows for the evaluation of the relative motion between the train vehicles and to calculate relative angles and distances which are of fundamental interest in the design of train components. A general methodology based on Cartesian coordinates is proposed for the kinematic analysis of multibody systems being further used to develop the planar models of the trainsets and to perform their kinematic analysis. Besides the standard revolute and composite joints a trajectory following constraint is developed and implemented to guide the vehicle wheelsets on the railroad, which in turn is described by using cubic splines to interpolate a given set of keypoints. The methodology is implemented in a general purpose computer program and applied to the study of a complex trainset. Finally, the methodology and the models developed in this work are validated with respect to the classical procedures used in industry. The objectives of the application of the methodologies proposed are the evaluation of the relative orientation and positions between the carbodies and the appraisal of the feasibility of insertion of the trainset in the geometry of the railroad. Such results are used in the design of vehicle components such as couplers and gangways or railroad functional envelope studies.

1. Introduction

The evaluation of the relative motion between the carbodies in a train involves some of the most complex issues of railway dynamics. Assuming that the flexibility of each carshell does not play any role in the behavior of the trainset, a train can be viewed as a collection of a rigid bodies connected by rigid or flexible links, supported by mechanical systems known as bogies, which include the wheelsets that are the sub-systems ultimately in contact with the railway.

The movement of the wheelsets over the railway is characterized by a complex interaction between wheels and rails with nonlinear and time variable contact forces [1,2]. The rail-wheel contact modeling involves, among other topics, the contact mechanics, friction and rigid body dynamics [3,4]. The result of all contributions to the final motion of the wheelsets exhibits a highly nonlinear behavior, although contained inside displacement limits during normal operating conditions. When the wheelsets travel along the railway in the longitudinal direction there are also appreciable transversal translations and yaw and roll rotations but with the wheels always in contact with the rails, unless there is a derailment not considered in the design. For the developments that follow it is assumed that the motion of each wheelset follows exactly the geometry of the railway. This is referred here as the wheelset reference motion.

The wheelsets are connected to the chassis of the bogies by a set of joints, which control the relative motion between wheels and chassis, and by a set of springs and dampers. This sub-system is designated by primary suspension [1]. In turn, the bogie chassis is connected to a support beam to which one of the carshell connecting shaft is attached. The connection between chassis and support beam is also done by a set of kinematic joints, springs and dampers, designated by secondary suspension [1]. Each train carbody is generally mounted in the top of two bogies. The attachment between the carbody and each bogie is done by a shaft

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doi:10.1016/j.mechmachtheory.2010.04.008

rigidly connected to the carbody that is inserted in a bushing joint located in the support beam of the bogie. This joint allows for a free yaw rotation of the bogie with respect to the carbody. However, the relative roll rotation is penalized by a moment that results from the deformation of the bushing joint.

Assuming a reference motion of the wheelsets, the general motion of each carbody is defined by the dynamics of the bogie subsystems, carbody inertia characteristics, attachments to the bogies and the interaction between the different carbodies. This motion is always three dimensional and depends, among other factors, in the velocity and acceleration of the train set. For example, when a trainset starts to negotiate a curve in the railway, even when reference motion is assumed, each different carbody starts a rolling motion with respect to the other carbodies. This is due to the centrifugal forces resulting from the curved motion and from the degrees-of-freedom of the suspension systems.

Another aspect of importance to the trainset dynamics is related with the geometry and flexibility of the coupling devices between the carbodies of the trainset. For the purpose of what follows it is assumed that the coupling devices are rigid. Two types of coupling devices are accounted for here: a coupler that ensures that one fixed point of a proceeding carbodies is always at a constant distance of another point fixed to the carbody that follows; a coupling of two successive carbodies by a ball joint.

The objective of this work is to develop a general methodology that allows modeling trainsets and simulating the relative motion between carbodies of the trainset when the wheelsets follow a given reference motion. For this purpose it is assumed that the displacement of the points of each bogie, coincident with the center of the wheelset, follow exactly the reference motion of such wheelset. Moreover, the roll motion of each carbody is not described in the methodology implemented. Finally, the cant angle of the curved track and the irregularities are not accounted for in what follows. Two implications of these simplifications are that the suspension systems of the bogies are not modeled and that the inertia forces are not accounted for either. Consequently, the methodology used in the numerical tool now presented accounts only for in-plane motion. Although in most of the cases of practical interest the motion develops in the horizontal plane (*XY*) there are cases in which the motion of importance is in the vertical (*XZ*) plane. Such cases are considered here independent. As the models obtained in this form have only one degree-of-freedom, which is guided by the driving bogie, the relative motion between carbodies can be fully described by a kinematic analysis. The out-of-plane kinematics, the lateral and yaw displacements of the wheelsets with respect to the track reference line, the track irregularities and cant angles are all modeled as perturbations to the wheelset reference motion and to the carbodies geometries. Although the objective of this work is not to present how such perturbations are handled, for the sake of completeness their modeling and integration on a general analysis is also overviewed here. The methodologies proposed in this work are applied to the design of the coupling elements of a trainset of a surface metro. In the process, the procedures are validated through the use of a classic design process in which geometric modeling software is used instead.

2. Kinematic analysis methodology

Kinematic analysis is the study of the motion of a system independently of the forces that cause it. For this purpose, a set of coordinates to describe the system components is chosen, eventual relations among those coordinates representing physical joints between system components are defined and finally a set of drivers are set up to control the degrees-of-freedom of the system. The kinematic analysis is then carried on by solving the set of equations resulting from the kinematic constraints between the system components and from the drivers [5].

2.1. Kinematic analysis

Let a set of coordinates, described by vector $\mathbf{q}$, be defined to represent the position and orientation of each rigid component of multibody system, and depicted in Fig. 1. Depending on the choice of coordinates used to describe the multibody system a larger or smaller set of equations relating these coordinates are defined. These equations, referred to as constraint equations, represent the mutual dependency of the coordinates [5].

The number of degrees-of-freedom of the system is equal to the difference between the number of coordinates and the number of independent constraint equations. In kinematics, each degree-of-freedom of the system must be driven being the equation that describes each motor referred to as driving constraint. Let all equations representing the different kinematic and driving constraints be grouped together. This set of equations, designated by position constraint equations, is written as:

$$\Phi(\boldsymbol{q}, t) = \mathbf{0}. \tag{1}$$

The set of Eq. (1) is nonlinear in the coordinates $\mathbf{q}$ and in the time t, which is used for the driving constraints. Its solution is obtained using the Newton-Raphson method [1].

The time derivative of Eq. (1) provides the velocity equations for the system, given by

$$\dot{\Phi} = 0 \quad \equiv \quad \Phi_{\boldsymbol{q}}\dot{\boldsymbol{q}} = \boldsymbol{\nu} \tag{2}$$

where $\dot{\mathbf{q}}$ is the vector of the velocities and $\boldsymbol{\nu}$ contains the partial derivatives of the position equations with respect to time. Note that $\dot{\mathbf{q}}$ is the time derivative of $\mathbf{q}$ in the planar case.

The time derivative of Eq. (2) leads to the acceleration equations of the system, written as

$$\ddot{\Phi} = \mathbf{0} \;\equiv\; \Phi_{\boldsymbol{q}}\ddot{\boldsymbol{q}} \;= \boldsymbol{\gamma} \tag{3}$$