Engineering Structures 69 (2014) 83–89

Contents lists available at ScienceDirect

Engineering Structures

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

A direct method for analyzing the nonlinear vehicle–structure interaction

S.G.M. Neves [a,b,*], P.A. Montenegro [b], A.F.M. Azevedo [b], R. Calçada [b]

[a] ISEP, Polytechnic Institute of Porto, Rua Dr. António Bernardino de Almeida, 431, 4200-072 Porto, Portugal
[b] Centro de Estudos da Construção, Faculdade de Engenharia, Universidade do Porto, Rua Dr. Roberto Frias, 4200-465 Porto, Portugal

ARTICLE INFO

Article history:
Received 28 September 2013
Revised 24 February 2014
Accepted 25 February 2014
Available online 5 April 2014

Keywords:
Vehicle–structure interaction
Nonlinear contact
Contact element
Dynamic analysis

ABSTRACT

This article presents an accurate, efficient and stable algorithm to analyze the nonlinear vertical vehicle–structure interaction. The governing equilibrium equations of the vehicle and structure are complemented with additional constraint equations that relate the displacements of the vehicle with the corresponding displacements of the structure. These equations form a single system, with displacements and contact forces as unknowns, that is solved using an optimized block factorization algorithm. Due to the nonlinear nature of contact, an incremental formulation based on the Newton method is adopted. The vehicles, track and structure are modeled using finite elements to take into account all the significant deformations. The numerical example presented clearly demonstrates the accuracy and computational efficiency of the proposed method.

1. Introduction

The development of efficient and robust algorithms that can accurately analyze the nonlinear vehicle-structure interaction is still an important issue, especially due to the increase of the corresponding operating speeds.

A vehicle-structure interaction problem is considerably more complex than a typical structural dynamics problem due to the relative movement between the two subsystems and the associated constraint equations relating the vehicle and structure displacements. In a significant number of studies available in the literature about the vehicle–structure interaction, the structure and vehicles are modeled as rigid multibody systems [1,2]. Other authors, such as Antolín et al. [3] and Tanabe et al. [4], proposed formulations that additionally take into account the deformation of the structure. Neves et al. [5] modeled the vehicles and structure using finite elements, thus considering the deformation of both systems.

When the vehicle and structure are considered as a single system, the forces acting on the contact interface are internal forces. Since the vehicle moves relatively to the structure, to avoid calculating and assembling the element matrices at each time step Yang and Wu [6] proposed a new contact element based on a condensation technique that eliminates the degrees of freedom at the contact interface. However, since the matrices of these elements depend on the position of the contact points, the global stiffness matrix is time-dependent and must be updated and factorized at each time step. This procedure may demand a considerable computational effort.

When the vehicle and structure are treated as separate systems, two different approaches can be adopted: variational formulations that consider an additional term in the energy of the system can be used to impose the constraints [7], or the contact forces can be considered explicitly and treated as externally applied loads, being the equilibrium of all forces acting on the contact interface established directly.

In the methods described in [8–11] the contact forces are considered explicitly but are not treated as unknowns of the governing equilibrium equations. An iterative procedure is used to ensure the coupling between the two subsystems. These methods may exhibit a slow rate of convergence, especially when unilateral contact is considered or a large number of contact points are required. To overcome these limitations, Neves et al. [5] developed an accurate, efficient and robust algorithm to analyze the vertical vehicle–structure interaction, referred to as the direct method, in which the governing equilibrium equations of the vehicle and structure are complemented with additional constraint equations that relate the displacements of the contact nodes of the vehicle with the corresponding nodal displacements of the structure, with no

* Corresponding author at: ISEP, Polytechnic Institute of Porto, Rua Dr. António Bernardino de Almeida, 431, 4200-072 Porto, Portugal. Tel.: +351 22 508 1901; fax: +351 22 508 1446.

E-mail addresses: sgm.neves@fe.up.pt (S.G.M. Neves), pedro.montenegro@fe.up.pt (P.A. Montenegro), alvaro@fe.up.pt (A.F.M. Azevedo), ruiabc@fe.up.pt (R. Calçada).

http://dx.doi.org/10.1016/j.engstruct.2014.02.027
0141-0296/

separation being allowed. These equations form a single system, with displacements and contact forces as unknowns, that is solved directly using an optimized block factorization algorithm. The Lagrange multiplier method and the direct method are equivalent and lead to identical systems of linear equations. The main advantage of the direct equilibrium of forces, when compared with the variational formulations, is a better understanding of the physical meaning of the contact forces, which is particularly important in complex problems such as the vehicle–structure interaction.

In the present article a search algorithm is used to detect which elements are in contact, being the constraints imposed when contact occurs. The time integration is performed using the α method since it provides numerical dissipation in the higher modes while maintaining second-order accuracy [12]. The proposed methodology is implemented in MATLAB [13]. The vehicles and structure are modeled with ANSYS [14], being their structural matrices imported by MATLAB.

2. Contact and target elements

When studying the contact between two bodies, one conventionally has a contact surface, and the other a target surface (see Fig. 1). A two-dimensional node-to-segment contact element is used in the present formulation.

The direct method [5] introduces additional variables in the system to impose the contact conditions, whereas in the penalty method no additional variables are required. Increasing values of the penalty parameter lead to more accurate solutions, but the coefficient matrix might become ill-conditioned. In railway engineering the number of contact points is usually small when compared with the total size of the problem. For this reason, the use of the direct method leads to a small additional computational cost but has the advantage of avoiding ill-conditioned systems.

In the formulation proposed in [5] the contact constraint equations are imposed using the direct method, with no separation being allowed. In the present formulation a search algorithm is used to detect which elements are in contact, being the constraints imposed when contact occurs. Since in the present formulation only the frictionless contact is considered, the constraint equations are purely geometrical and relate the displacements of the contact node with the displacements of the corresponding target element.

Fig. 2 shows the two-dimensional node-to-segment contact element implemented in the present formulation and the local coordinate system (ξ_1, ξ_2, ξ_3) of the contact pair. The ξ_2 axis always points towards the contact node, being the two elements separated by an initial gap g. The forces acting at the contact interface are denoted by X and the superscripts CE and TE indicate contact and target elements, respectively.

According to Newton's third law, the forces acting at the contact interface must be of equal magnitude and opposite direction, i.e.,

$$\mathbf{X}^{CE} + \mathbf{X}^{TE} = \mathbf{0} \tag{1}$$

The displacement vector of an arbitrary point is defined by two translations, v_{ξ_1} and v_{ξ_2}, and a rotation θ_{ξ_3} about the ξ_3 axis. Since this type of contact element neglects the tangential forces and moments transmitted across the contact interface, the contact constraint equations only relate the displacement v_{ξ_2} of the contact node with the corresponding displacement of the auxiliary point k. Each constraint equation is defined in the local coordinate system of the contact pair and comprises the non-penetration condition for the normal direction. These equations are given by

$$\mathbf{v}^{CE} - \mathbf{v}^{TE} \geq -\mathbf{g} + \mathbf{r} \tag{2}$$

where $\mathbf{r}$ are the irregularities between the contact and target elements. The gaps are always positive and a positive irregularity implies an increase of the distance between the contact and target elements (see Fig. 2).

3. Equations of motion

3.1. Force equilibrium

The α method is an implicit time integration scheme that is generally accurate and stable [12]. Assuming that the applied loads are deformation-independent and that the nodal point forces corresponding to the internal element stresses may depend nonlinearly on the nodal point displacements, the equations of motion of the vehicle–structure system given in [5] may be rewritten in the form

$$\mathbf{M}\mathbf{a}^{t+\Delta t} + \mathbf{C}[(1+\alpha)\dot{\mathbf{a}}^{t+\Delta t} - \alpha\dot{\mathbf{a}}^{t}] + (1+\alpha)\mathbf{R}^{t+\Delta t} - \alpha\mathbf{R}^{t} = (1+\alpha)\mathbf{F}^{t+\Delta t} - \alpha\mathbf{F}^{t} \tag{3}$$

where $\mathbf{M}$ is the mass matrix, $\mathbf{C}$ is the viscous damping matrix, $\mathbf{R}$ are the nodal forces corresponding to the internal element stresses, $\mathbf{F}$ are the externally applied nodal loads and $\mathbf{a}$ are the nodal displacements. The superscripts t and $t+\Delta t$ indicate the previous and current time steps, respectively.

To solve Eq. (3) let the F type degrees of freedom (d.o.f.) represent the free nodal d.o.f., whose values are unknown, and let the P type d.o.f. represent the prescribed nodal d.o.f., whose values are known. Thus, the load vector can be expressed as

$$\mathbf{F}_F = \mathbf{P}_F + \mathbf{D}_{FX}^{CE}\mathbf{X}^{CE} + \mathbf{D}_{FX}^{TE}\mathbf{X}^{TE} \tag{4}$$

$$\mathbf{F}_P = \mathbf{P}_P + \mathbf{D}_{PX}^{CE}\mathbf{X}^{CE} + \mathbf{D}_{PX}^{TE}\mathbf{X}^{TE} + \mathbf{S} \tag{5}$$

where $\mathbf{P}$ corresponds to the externally applied nodal loads whose values are known and $\mathbf{S}$ are the support reactions, whose values are unknown. Each matrix $\mathbf{D}$ relates the contact forces, defined in the local coordinate system of the respective contact pair, with the nodal forces defined in the global coordinate system (see Fig. 2).

Substituting Eq. (1) into Eqs. (4) and (5) leads to

$$\mathbf{F}_F = \mathbf{P}_F + \mathbf{D}_{FX}\mathbf{X} \tag{6}$$

$$\mathbf{F}_P = \mathbf{P}_P + \mathbf{D}_{PX}\mathbf{X} + \mathbf{S} \tag{7}$$

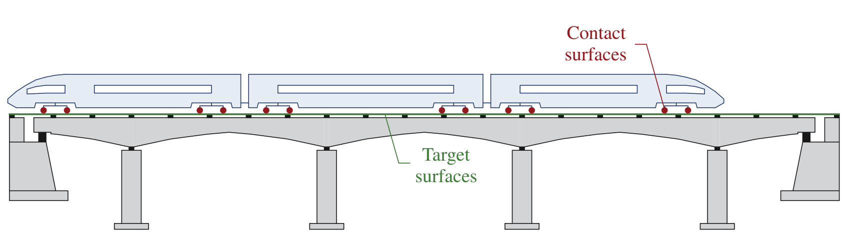

Fig. 1. Contact pair concept.