Engineering Structures 160 (2018) 473–487

Contents lists available at ScienceDirect

Engineering Structures

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

Dynamic and structural integrity analysis of a complete elevator system through a Mixed Computational-Experimental Finite Element Methodology

Dimitrios Giagopoulos[a,*], Iraklis Chatziparasidis[a,b], Nickolas S. Sapidis[a]

[a] Department of Mechanical Engineering, University of Western Macedonia, Kozani 50100, Greece
[b] Research & Development Department, Kleemann Hellas SA, Kilkis Industrial Area 61100, P.O. Box 25, Greece

ARTICLE INFO

Keywords:
FE modeling
Large scale structures
Model updating
Stress evaluation
Structural dynamics

ABSTRACT

A systematic dynamic and structural integrity analysis of an elevator system, under real dynamic load conditions, is presented in this work. The proposed method constitutes an important innovation for the field "analysis and design optimization of elevator systems". Current elevator design requirements lead to development of lightweight structures which often are of higher complexity. In order to achieve design optimization of these systems, it is important to develop an accurate Finite Element Analysis methodology, to support their precise numerical modeling, also in the worst loading case for their operation. One of the worst loading cases for an elevator system is when the elevator falls free and the safety gear is activated until the elevator stops. Each elevator, when installed in a building, is tested in this loading scenario. This means, that should be designed to withstand in these loads and not to have any damage after this test. One of the special features of this paper is that the study was performed on a complete industrial elevator system, including all details/complexities of a commercial system. Comparison of the numerical and experimental data verifies that the proposed "mixed computational-experimental" analysis method is quite reliable, concluding application/verification of the method on a complete industrial elevator system.

1. Introduction

Ever evolving elevator design requirements frequently require improvements and modifications of either specific mechanical components or even of entire structures. In order to achieve design optimization of these systems, it is important to develop an accurate Finite Element Analysis methodology, to support their precise numerical modeling, also in the worst loading case for their operation. The worst loading case for an elevator system is when the elevator falls free and the safety gears are activated until the elevator stops. The synchronized activation of the safety gears is a very important issue. However, because of the inertia of the activation system and of manufacturing tolerances deferred activation of safety gears could happen. To study the effects of this on the elevator, the main finite element analysis used in the industry is static structural analysis. This analysis makes a series of assumptions/simplifications, resulting to a simulation not sufficiently accurate for the present problem. The case of the non-synchronous activation of the safety gears cannot be simulated at all with a static analysis. So, to solve these modeling issues, one reasonable approach is to develop an accurate mixed computational-experimental procedure, to simulate accurately the dynamic behavior. This is the main research issue addressed by the present work.

The equations of motion of a large and geometrically complex structural system are set up by applying classical finite element techniques. As the order/complexity of these models increases, existing numerical and experimental methodologies for determining their dynamic response become inefficient. To cure this problem, appropriate substructuring methods in either the time or the frequency domain have been developed and are usually employed [1–5,19–21]. A basic ingredient of these methods is the determination of sets of component modes allowing a drastic reduction in the numerical dimension of the system examined [1,2,6,7]. In addition, there are many occasions in practice where exact technical characteristics, including those related to strongly nonlinear action of some structural components, are not known. In such cases, appropriate mixed methodologies, involving generation and processing of information related to the system dynamic response by a combination of numerical and experimental techniques, are applied. Specifically, some of the components of the structure are modelled numerically while the remaining components are modelled experimentally [8,9,11]. Also, in structures that are mostly subjected to fatigue, it is necessary to develop a high fidelity finite element model, based on which the fault locations and the lifetime of the construction

* Corresponding author.
E-mail addresses: dgiagopoulos@uowm.gr (D. Giagopoulos), i.chatziparasidis@kleemannlifts.com (I. Chatziparasidis), nsapidis@uowm.gr (N.S. Sapidis).

https://doi.org/10.1016/j.engstruct.2018.01.018
Received 19 January 2017; Received in revised form 7 January 2018; Accepted 8 January 2018
0141-0296/

can be accurately determined. In such cases, appropriate finite element model updating methods are applied combining a substructuring approach with a near real-time system identification scheme, namely the Unscented Kalman Filter (UKF) [13–15,22–24]. The former aims at isolating and locally updating individual structural subsystems of a large-scale structure, while the latter, in contrast to other alternatives (e.g. the Extended Kalman Filter), offers a number of advantages in treating nonlinear systems, such as a derivative free calculation and a capacity for handling higher order nonlinearities.

The main objective of the present work is to detail an appropriate mixed numerical-experimental methodology to accurately predict dynamic response and identify critical points in an elevator system [10]. The special feature of this research is that this study is performed on a real industrial elevator system, characterized by a quite complex structure. The complete elevator system examined consists of three main parts. The first part includes all the supporting parts, like the guide rails, hydraulic piston, wire ropes, etc. Usually, no information about the specific dynamic characteristics of this part is available. The second part includes the chassis of the elevator, while the third part includes the cabin which is mounted onto the chassis. For the chassis and the cabin, a detailed finite element model is developed, leading most-often to an excessive number of degrees of freedom. Therefore, a mixed modeling method is employed for solving and investigating the specific problem. In this way, complications arising from inadequate information on the parameters of the supporting components are avoided. Additionally, the present methodology includes a drastic reduction of the original degrees of freedom of the systems examined.

The method developed in the present work can be used to determine the dynamic behavior of the elevator main substructures (chassis and cabin). Also, this leads to an accurate identification of points where critical (largest) stresses appear. This is done by applying a numerical method for determining the equations of motion for the chassis and cabin, while the dynamic characteristics of the remaining components are taken into account through the application of appropriate experimental measurements.

The procedure proposed for solving and analyzing this specific problem includes the following steps. First, the chassis and the cabin of the elevator are modelled by discretizing them geometrically according to the FE method. In order to verify the accuracy of the FE model and also identify the braking forces acting on the system, one first examines only the elevator chassis. The initial FE model of the chassis is updated and validated through an experimental investigation of its dynamic response when the elevator stops using instantaneous or progressive safety gear. These experimental tests are performed under real operating conditions, using an experimental device that was designed exactly for this purpose and aimed at recording the acceleration time histories at the connection points of the chassis with the safety gear and at other locations used as reference points. The acceleration time histories at the connection points are subsequently used as base excitation for the FE model of the chassis and the corresponding stresses developed are evaluated. On the basis of these numerical results, the critical points of the chassis are specified, as those corresponding to larger stresses. In order to test the reliability of the proposed method, strain gauges are placed at the critical points of the chassis and measurements are carried out, under similar dynamic load conditions, in order to experimentally verify the stresses calculated above.

The organization of this paper is as follows. In the following section, an appropriate methodology is proposed for analyzing the stress in a chassis-cabin of a complete industrial elevator system in emergency cases, based on "dimensional reduction" and on using experiments to specify a base excitation for a simplified FE model of the complete system. Then, in the third section, the effectiveness and accuracy of this methodology is examined. Here, the reliability of the methodology is tested first in detail on the elevator chassis with the platform of the cabin and next on the full elevator system (chassis and cabin). The paper concludes with a summary of the obtained results.

2. Outline of the proposed mixed computational-experimental methodology

The equations of motion of mechanical systems with complex geometry are commonly set up by applying finite element techniques. Quite frequently, a systematic investigation of the dynamics of a large scale mechanical structure leads to models involving an excessive number of degrees of freedom. Therefore, a computationally efficient solution requires application of methodologies reducing the numerical dimension of the original model [1–5,7]. Next, the basic steps of a time domain reduction method are briefly presented.

For simplicity, consider a mechanical system consisting of two subsystems, say A and B. Moreover, let the equations of motion for subsystem A be derived from the following classical equation

$$\hat{M}_A\ddot{\underline{x}}_A + \hat{C}_A\dot{\underline{x}}_A + \hat{K}_A\underline{x}_A = \hat{\underline{f}}_A \tag{1}$$

where $\hat{M}_A$, $\hat{C}_A$ and $\hat{K}_A$ are, respectively, the mass, damping and stiffness matrix of the subsystem A, with the vector $\hat{\underline{f}}_A(t)$ representing the external forcing. For a typical model, the number of these equations may be quite large. However, for a given level of forcing frequencies, it is possible to reduce significantly the number of the original degrees of freedom, without sacrificing the accuracy in the numerical results, by applying standard component mode synthesis methods [1,12]. This can be achieved through an approximate Ritz transformation of the form

$$\underline{x}_A = \Psi_A\underline{q}_A \tag{2}$$

The transformation matrix Ψ_A includes an appropriately chosen set of the lowest frequency normal modes of component A, corresponding to support-free conditions [1,7]. The number of these modes depends on the accuracy required in the response frequency range examined. Consequently, the matrix Ψ_A is completed by a set of static correction modes of component A, MacNeal's Method [2,7,18]. Employing this transformation (2), the original set of equations (1) can be replaced by this considerably smaller set of equations, expressed in terms of the new generalized coordinates $\underline{q}_A$:

$$M_A\ddot{\underline{q}}_A + C_A\dot{\underline{q}}_A + K_A\underline{q}_A = \underline{f}_A \tag{3}$$

where

$$M_A = \Psi_A^T\hat{M}_A\Psi_A,\quad C_A = \Psi_A^T\hat{C}_A\Psi_A,\quad K_A = \Psi_A^T\hat{K}_A\Psi_A \quad \text{and} \quad \underline{f}_A = \Psi_A^T\hat{\underline{f}}_A$$

Moreover, the set of unknowns can be split in the form

$$\underline{q}_A = (\underline{p}_A^T\ \underline{x}_b^T)^T$$

where $\underline{p}_A$ includes coordinates related to the response of internal degrees of freedom of component A, while $\underline{x}_b$ includes the boundary points of component A with component B. Next, similar sets of equations of motion are obtained for component B. Namely, the equations of motion are first set up in the form

$$M_B\ddot{\underline{q}}_B + C_B\dot{\underline{q}}_B + K_B\underline{q}_B = \underline{f}_B \tag{4}$$

with coordinates

$$\underline{q}_B = (\underline{p}_B^T\ \underline{x}_b^T)^T$$

Then, a proper combination of equations (3) and (4) leads to the equations of motion of the composite system in the classical form

$$M\ddot{\underline{q}} + C\dot{\underline{q}} + K\underline{q} = \underline{f} \tag{5}$$

with coordinates

$$\underline{q} = (\underline{p}_A^T\ \ \underline{p}_B^T\ \ \underline{x}_b^T)^T.$$

The stiffness matrix of the composite system can be obtained by considering the total potential energy of the system. Likewise, the mass matrix of the composite system is obtained by considering the corresponding kinetic energy, while the forcing vector is determined by