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TOWER CRANE – ASPECTS OF MODEL REDUCTION AND ACTIVE VIBRATION SUPRESSION

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Abstract

Transportation aims at different solutions to move the objects easier, faster and more economical. The solutions call for optimization with various optimization objectives and constraints that depend on specific circumstances. Modeling of proposed design solutions and simulation of machine operations belong to typical engineering tasks which enable assessment of the solutions. As simulations may be rather time-consuming, appropriate model reduction is quite often addressed. A care is needed if nonlinearities are involved in the structural behavior and the first part of the paper handles this aspect on an example of a tower crane. In the second part, the possibility of active vibration suppression is considered on the same tower crane example.

Key words*: tower crane, model reduction, active vibration supression*

1 INTRODUCTION

Transportation as an engineering discipline calls for solutions that enable easier, faster and more economical movement of people, goods, materials, etc. Hence, it implies optimization in the choice of transportation mode, patterns, means, and operations. A necessary prerequisite for the optimization is modeling and simulation of the machine behavior during the process of transportation. In this manner, one can assess the proposed solutions with respect to various criteria, make a choice of parameters and improve them in further analysis.

Modeling is one of the fundamental activities engineers are involved in. Model building is a procedure that explores alternative solutions with the aim of achieving a satisfying compromise between the model complexity and the accuracy of the predicted behavior of the physical system. The two aforementioned objectives are not always easy to conciliate and different techniques are used to achieve what is believed to be the best compromise.

Transportation implies dynamics. Simulation of dynamical behavior of transportation machines belongs to the field of structural dynamics. Well-developed solutions are available to engineers as a part of commercial software packages for FEM (finite element method) and MBS (multi-body system) analysis. They also involve standard solutions for model reduction in order to improve the numerical efficiency. While nonlinearity is intrinsic for a great number of such structures, linear models are still used in many cases as a very good approximation of the actual system behavior within a certain, typically small domain in the vicinity of the original system configuration. If, however, the considered physical behavior cannot be described accurately enough by a linear model, then a nonlinear model becomes an imperative. Switching from linear to nonlinear analysis implies a much greater numerical effort required to solve the resulting nonlinear system of equations.

There are, however, cases characterized by local nonlinearities. In such a case, a large portion of the structure can be adequately described by a linear mode, while a local substructure demands to account for nonlinear behavior. Highly efficient modeling would demand to use a kind of mixed linear – geometrically nonlinear model. This is the aspect addressed in the first part of the paper. A tower crane model is used for the illustration of the proposed idea. The second part considers the idea of active vibration suppression of dynamically loaded structures. The idea can bring considerable savings in operational times of different machines and improve their safety and robustness. The same model of the tower crane is used for the illustration.

2 COMBINED LINEAR – GEOMETRICALLY NONLINEAR APPROACH AND MODAL SUPERPOSITION

2.1. Theoretical background

The FEM equation of transient structural dynamics can be generally given as:

$$[M]\,{}^{t}\{\ddot{u}\}+{}^{t}[C]\,{}^{t}\{\dot{u}\}={}^{t}\{f_{ext}\}-{}^{t}\{f_{int}\} \qquad (1)$$

where $[M]$ and $[C]$ are the structural mass and damping matrices, $\{f_{ext}\}$ and $\{f_{int}\}$ are the external (excitation) and internal (elastic) forces of the FEM assemblage, $\{u\}$are the structural displacements with dots above denoting time derivatives (acceleration and velocity), while the left superscript denotes at which moment of time the quantity is taken.

If a linear model is considered, the change of the structural configuration is neglected and the material law is assumed to be constant, which allows the computation of the internal forces based on the structural stiffness matrix $[K]$ determined for the initial structural configuration:

$${}^{t}\{f_{int}^{\,lin}\}=[K]\,{}^{t}\{u\} \qquad (2)$$

with $\{u\}$ denoting the current nodal displacements.

A nonlinear analysis is rigorous and, thus, determines the internal forces based on the current stress state in the structure, $\{\sigma\}$, and current configuration:

$$ {}^{t}\{f_{int}^{nl}\} = \int_{{}^{t}V} {}^{t}[B]\,{}^{t}\{\sigma\}\,d\,{}^{t}V \qquad (3) $$

where $[B]$ is the strain-displacement matrix of the FE assemblage.

The essence of the idea proposed in this paper is to conduct structural analysis by computing structural internal forces according to either Eq. (2) or Eq. (3), depending on the suitability of the expressions for a specific structural domain. Eq. (2) is used for the part of the structure for which a linear model can be used, and Eq. (3) for the sub-domains that demand a nonlinear model. This approach is reasonable for structures characterized by local nonlinearities, i.e. where sub-domains demanding Eq. (3) are relatively small compared to the whole structure. This enables great computational savings as the computation of the linear stiffness matrix can be done in a pre-step prior to simulation.

FE models typically have a large number of degrees of freedom, which could go up to several million and the solution of the resulting system of equations is numerically rather demanding, particularly in structural dynamics. In order to reduce the numerical effort, model reduction is applied. Modal decomposition is a standard technique and it is especially convenient if a structure is excited by a band-limited excitation. It is limited to linear models only and, within the proposed idea, it is applicable to the linear part of the model.

Model reduction is performed by transforming the system equations with the aim of reducing its bandwidth and size. A quite common approach is based on the transformation from nodal to modal space. With carefully chosen modes, the bandwidth of the system matrices can be reduced dramatically. Using eigenmodes $\{\phi_i\}$, which are obtained as a solution to the eigenvalue problem:

$$ \left([K_{uu}] - \omega_i^2 [M_u]\right)\{\phi_i\} = 0 \qquad (4) $$

and which are orthogonal with respect to the mass and stiffness matrices, the transformed equations become actually decoupled. In Eq. (4) ω_i are the eigenfrequencies. Additionally, in most cases, it is the structural response in lower spectrum range that is of interest (modes with higher eigenfrequencies are typically damped out very fast). This fact allows to take only the mode shapes within certain (lower) frequency range, thus reducing the model size dramatically. Introducing the matrix of mode shapes $[\Phi]$ so that $[\Phi]=[\{\phi_1\}\ \{\phi_2\}\ \ldots\ \{\phi_n\}]$, the system matrices and loads are transformed in the following manner [1]:

$$ [\tilde{M}] = [\Phi]^{\mathrm{T}}[M][\Phi], \quad [\tilde{C}] = [\Phi]^{\mathrm{T}}[C][\Phi], $$
$$ [\tilde{K}] = [\Phi]^{\mathrm{T}}[K][\Phi], \quad \{\tilde{f}\} = [\Phi]^{\mathrm{T}}\{f\} \qquad (5) $$

which leads to the following system of equations for the reduced linear part of the model:

$$ [\tilde{M}]\{\ddot{z}\} + [\tilde{C}]\{\dot{z}\} + [\tilde{K}]\{z\} = \{\tilde{f}\} \qquad (6) $$

The coupling between the linear and geometrically nonlinear part of the model can be effectively done by using the explicit time integration scheme as it does not require assemblage of the system matrices.

2.1. Application to a tower crane model with suspended mass

A tower crane is a large span structure and certain points of the structure, such as the working jib tip, may undergo large displacements during the operation of the tower crane. However, compared to the structural dimensions, those displacements are, for typical operational conditions and loads of a tower crane, still in the realm of the physical behavior that can be described by a linear model.

The tower crane performs a combination of motions to reach any point within its working radius. Attached to the very top of the mast is the slewing unit comprised of a gear and motor that gives the crane the ability to rotate. A trolley is fastened to the long working jib and carries the load along the jib. Steel ropes and a hook are used to suspend a load. Due to inertial forces during the transport of the load, which involves rotational motion and translational motion along the working jib, longitudinal and side sways of the load are easily initiated. The forces induced in the ropes are part of the excitation of the tower crane's structure. The load sways are the cause of changing line of action of the steel rope internal forces. Omitting this aspect, erroneous simulation results are obtained for the dynamical behavior of both the hanging load and the structure of the tower crane. A linear model observes only the initial configuration and the internal forces of the rope would act along the line of the original rope direction, which is typically vertical. Pulling the load along the direction of the jib would result in very large displacements as there is no stiffness that would resist this motion in the original configuration (only mass proportional damping).

Hence, the authors apply the idea of combined linear-geometrically nonlinear modeling to the tower crane model with a suspended load. The tower crane structure is considered by a linear model and the modal superposition technique is used to further increase the efficiency of that part of the model. The first 10 eigenmodes of the structure are used. The steel ropes with suspended load are considered separately by a geometrically nonlinear model coupled to the linear model of the tower crane's structure (Fig. 1).

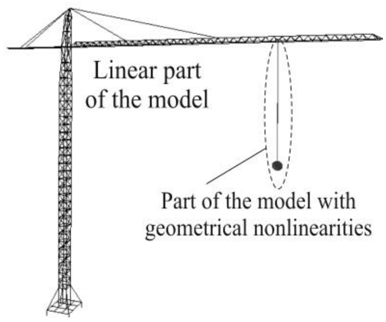

Fig. 1 *Linear – geometrically nonlinear model of a tower crane with suspended load.*

If the model depicted in Fig. 1 is entirely considered as linear, any excitation that acts horizontally onto the suspended mass (perpendicular to the rope) would result in unrealistic large displacements (Fig. 2b), as no stiffness is associated to such a motion of the suspended mass and an artificial enlargement of the rope can be noticed. But taking the local rotations of the steel rope into account resolves the problem successfully (Fig. 2c).

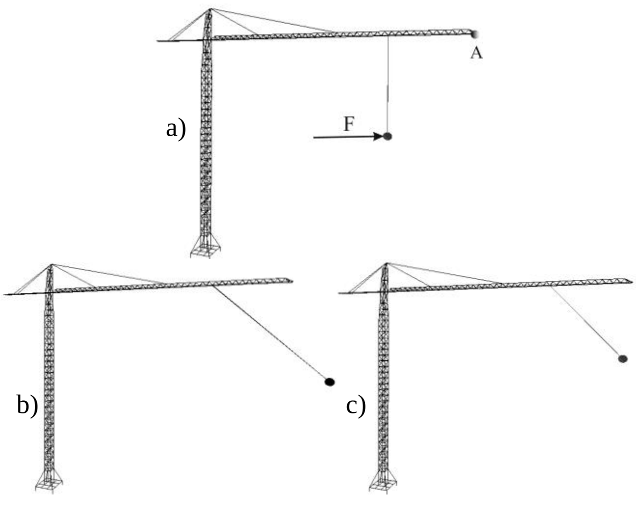

Fig. 2 *Tower crane model: a) initial configuration with excitation force; b) simulation by linear model 2; c) simulation by linear – geometrically nonlinear model*

To demonstrate this effect, a mass of 300 kg suspended on the rope has been exposed to a rather short impulse force of 5 kN in duration of 10^{-3} s (Fig. 2a) and the displacements of the working jib tip (point A in Fig. 2a) have been observed. The linear model yields no horizontal displacements of the working jib tip and in Fig. 3 those displacements would have been given as coinciding with the x-axis. With the linear model, the steel ropes can transfer the force only in vertical direction, while the motion of the mass is strictly horizontal. On the other hand, the combined model can resolve this successfully and Fig. 3a gives the displacement of the working jib tip in the vertical direction, i.e. parallel to the tower crane mast, while Fig. 3b shows the displacement of the same point in the horizontal direction, i.e. parallel to the working jib, predicted by the combined model (solid line) and by rigorous geometrically nonlinear FEM computed in ABAQUS (dashed line) for the simulation period of 3 s. In the conducted analysis, the static computation is performed first in order to determine the statically deformed initial configuration. The displacements at time t = 0 s correspond to this configuration. One may notice that there are differences between the initial displacements computed in ABAQUS and by the presented formulation. This is clearly the consequence of the fact that the computation in ABAQUS is geometrically nonlinear and done with the full FE model, whereas the computation by the presented formulation is strictly linear for the static case (the rope remains vertical) and performed in the modal space with only the first 10 eigenmodes as degrees of freedom. The dynamic analysis shows a relatively good agreement between the two formulations. It should be emphasized at this point that the rigorous geometrically nonlinear computation in ABAQUS is performed with the model containing 1773 degrees of freedom, while the model used with the presented formulation has only 13 degrees of freedom (10 eigenmodes and the 3 displacements of the suspended mass).

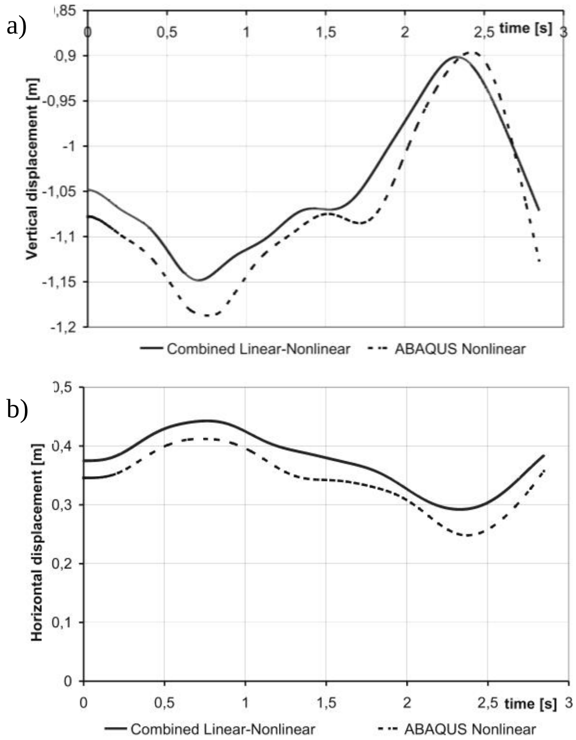

Fig. 3 *Displacements of the working jib tip: a) in vertical direction; b) in horizontal direction*

3 ADAPTIVE STRUCTURES IN THE FIELD OF TRANSPORT

Adaptive structures are characterized by the ability to mimic the behavior of natural systems. The general idea consists in using advanced multifunctional materials in order to design and integrate active elements, i.e. sensors and actuators, into structures and to couple them by means of a controller. Active elements can also be added as additional devices, if such a design better suits the structure or the required actuator force and stroke. The purpose of sensors is to provide signals that contain information about the state of the structure, such as acceleration, velocity, deformation, etc. The sensor signals are transmitted to a controller that implements the control law aiming at the desired structural behavior. This practically means that the controller processes the sensor signals and determines what action should be performed by actuators in order to produce a desired structural behavior. The corresponding signal is then sent to actuators. In this manner, the structure reacts adaptively to external excitations. The approach is most frequently used for active vibration suppression.

3.1. Active vibration suppression of the tower crane

Active vibration suppression improves many aspects of the structural dynamic behavior, ranging from comfort to robustness and safety.

To exemplify the idea of adaptive structural behavior, the same tower crane model already considered above will also be used here. Without going into details of possible solutions for sensors and actuators, they will be idealized here simply by assuming that the sensors can measure desired quantities related to truss elements and that the actuators can produce desired forces that act along truss elements. To meet the observability and controllability conditions [2], it will be assumed that the coupled sensors and actuators are collocated, i.e. positioned in the same truss elements.

Now, there are a number of strategies and algorithms developed to achieve active vibration suppression. A very simple logic will be applied here. It is for a good reason that damping - although being a rather complex effect - is quite often modeled in dynamics as viscous, i.e. velocity proportional. Hence, the idea used here is first to use sensors to gain the information on the strain rate in the truss elements where sensors are placed. In the next step, for each sensor the corresponding force is computed proportional to the measured strain rate. This force is supposed to be generated by the actuator in the very same truss element where the sensor is located and thus suppress the vibration.

The position of active elements is a very important aspect. There are different approaches depending on the objective of adaptive behavior. As mentioned, the objective in this specific case is vibration suppression. Control of complex systems with a large number of degrees of freedom is a prohibitively expensive task. An elegant idea to resolve the problem is to perform control within specific frequency range. As already elaborated, if the focus is put to a certain frequency range, then the dynamical behavior can be adequately described by means of model superposition. In other words, the mode shapes become the degrees of freedom to be controlled. The optimal position of active elements is where they can provide the best observability and controllability. Simply said, they should be located in the truss elements where their effect would be greatest. Hence, the optimal position of active elements is determined based on the strain energy density. The procedure implies to determine first the strain energy density for each of the mode shapes. Furthermore, the mode shapes are given weight factors based to their importance in the overall structural behavior and control. Finally, the weighted modal strain energy densities are superposed. This simple procedure reveals in which truss elements the control would have the largest impact onto the structural vibrations.

Now, considering the crane structure in Fig. 1 (without the suspended mass), upon limiting the number of active truss elements to 4 and the number of mode shapes of interest to 10, this procedure yields the truss elements shown in Fig. 4 as optimal for active elements (sensors and actuators). Over the course of simulation, the strain rate in those elements is determined and used to compute directly proportional actuator forces by means of a predetermined coefficient. It should be emphasized that the simulation is idealized, i.e. it is assumed that within a time-step of the time integration, the actuators can generate the forces proportional to the measured strain rate. In reality, this requires certain time that could be longer than the time-step and the delay would have to be accounted for in order to have a more realistic simulation.

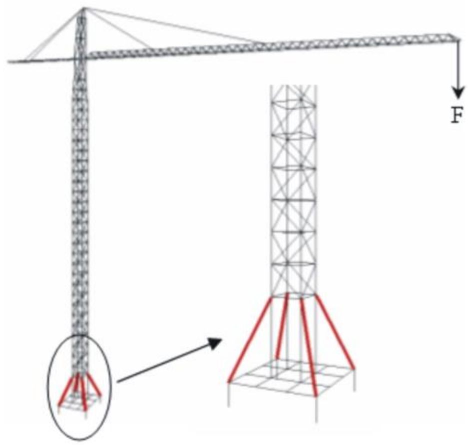

Fig. 4 *Tower crane with active truss elements*

In the test example, the crane is exposed to an external impact excitation in duration of 0.3 s acting at the tip of the crane jib (F in Fig. 4). The vertical displacement of the very same point is observed as a representative response of the structure. The internal structural damping is assumed to be relatively small. The obtained results are summarized in Fig. 5.

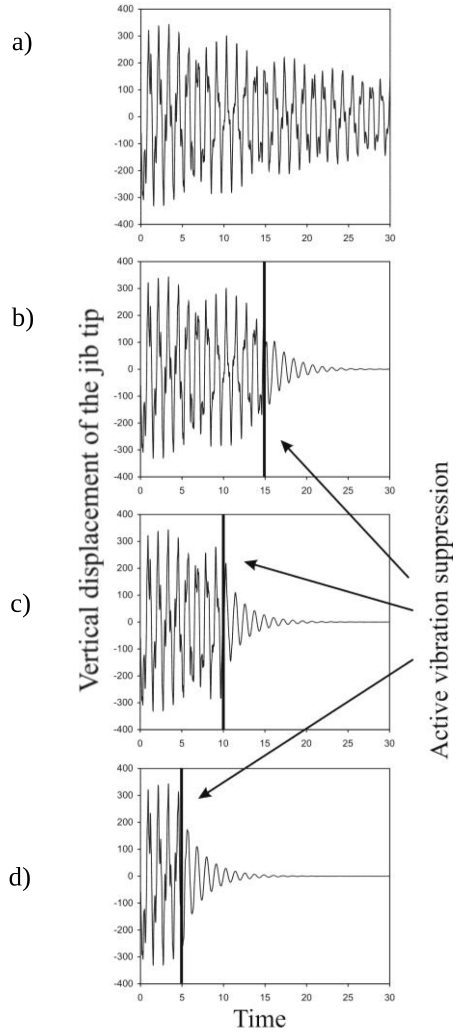

Fig. 5 *Vertical deflection of the crane's jib tip: a) no active damping; and with active damping started after: b) 15 s; c) 10 s; d) 5 s*

Fig. 5a gives the jib tip deflection without the active control applied and the decrease in amplitudes is the consequence of internal damping only. On the other hand, Figs. 5b, 5c and 5d show the response when the active control is turned on 15 s, 10 s and 5 s after the excitation, respectively. The effect of active control is obvious.

4 CONCLUSIONS

Innovative solutions are needed in the field of transport to push the leading edge of technology. Innovations in design solutions are performed to meet these objectives. And adequate modeling is a prerequisite that enables efficient simulations and therewith assessment of the proposed design solutions.

With the aim of very efficient modeling, the idea of combined linear - geometrically nonlinear modeling is proposed. It comes particularly to the fore in structural dynamics, where modal superposition technique can be additionally applied for the linear part of the model. Natural mode shapes are used in the tower crane example. In practical applications, however, different possibilities for the choice of modes should be considered, such as application of Craig-Bamption method of component mode synthesis [3]. This would offer higher modeling flexibility as variable boundary conditions can be considered during the simulation.

Over the past few decades, the idea of adaptive structures has progressed from almost an engineering curiosity to a new generation of high-performance structural and mechanical systems with integrated sensing, actuating and control capabilities. Their application becomes wider every day covering many areas of engineering, transportation being among the most important ones. The presented example is only illustrative. Many aspects are idealized in the tower crane example with the objective to demonstrate the basic idea. Despite of that, it shows the possible advantages of adaptive structures, but one should keep in mind that some tuning is required with real structures in order to make everything work as designed.

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