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**MODAL ANALYSIS OF LOADER CRANE WITH VARIABLE CONFIGURATION**

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<https://creativecommons.org/licenses/by/4.0/>**Key words:** modal analysis, loader crane, FEM analysis.

**Abstract:** Paper presents a method of simplifications used over numerical studies on dynamic properties of a hydraulic loader crane. Simplification concerns hydraulic actuators and allows for a significant reduction in the time of building model and calculations. Modal analysis of a loader crane using the finite element method, for both complex and simplified models, was carried out. The modal analysis computation times for both models were compared. Next, the results obtained on the basis of simplified model were compared with the results of experimental studies.

**Analiza modalna żurawia przeładunkowego o zmiennej konfiguracji****Słowa kluczowe:** analiza modalna, żuraw przeładunkowy, analiza MES.

**Streszczenie:** W pracy przedstawiono metodę uproszczeń stosowanych w badaniach numerycznych nad właściwościami dynamicznymi żurawia hydraulicznego. Uproszczenie dotyczy siłowników hydraulicznych i pozwala na znaczne skrócenie czasu budowy i obliczeń modelu. Przeprowadzono analizę modalną żurawia za pomocą metody elementów skończonych zarówno dla pełnych, jak i uproszczonych modeli. Porównano czasy obliczeń analizy modalnej dla obu rodzajów modeli. Następnie porównano wyniki uzyskane na podstawie uproszczonego modelu z wynikami badań eksperymentalnych.

**Introduction**

Loader cranes are devices of wide application in various areas of transport. From the point of view of their structure, they can be classified as manipulators. The growing demand and simultaneously increasing functional requirements have led to constant improvement of their construction. As a result, they have become more complex and sophisticated. In the case of such structures, accurate prediction of problems related to time varying loads at the design stage is of significant importance.

One of the methods to predict the dynamic behaviour of such structures is the finite element method. However, the fact that dynamic properties of loader cranes change with their configuration, a new model for each configuration should be constructed. For models of such complexity, it is ineffective in terms

of calculations time. Therefore, modelling methods that allow a significant reduction of model order while maintaining the high accuracy are sought.

In this paper, a simplified model of a loader crane hydraulic actuator is proposed. Modal analysis of a loader crane using the finite element method using full and simplified actuator models was carried out. All analyses were conducted in Midas NFX. This was the basis for the influence of simplifications comparing accuracy and model order. In the first part, analysis of a simple model of loader crane with only two rotational joints was conducted. The computational times for the structure with the actuator modelled in a classical way and for the structure with the substitution in place of actuator model were compared. Based on these results, it was concluded that used simplifications has a negligible effect on the results. Therefore, in the second part of the work, a model of crane was built that consisted of six translational joints. For this structure, all of the actuators

were modelled in a simplified way. The obtained results were then compared with the results of experimental studies.

## 1. Research problem

Loader cranes are a very popular subject of scientific research in many aspects. In paper [1], the authors compiled a lot of works related to the study of the dynamics and control of cranes. They specified three types of these mechanisms: gantry, rotary, and boom cranes. Truck cranes generally consist of booms with a rotary basis. The other mentioned aspect is the way of modelling their load; it could be modelled as a lumped mass or a distributed mass. The difference in both approaches in terms of mathematics is presented in paper [1].

While modelling a truck crane, all subsystems influencing its operation should be considered. Paper [2] presents a universal mathematical model that considers complex interaction between the elements of dynamic system. There were specified five main subsystems that determine the truck loader working conditions: supporting structure, base surface, basic machine (basis), loader crane construction, and actuating elements. The identification of the elastic support system has been presented in [3]. That consisted in the determination of spring constants replacing the flexible carrier system. The values of these elastic constants were determined based on the solution of the problem of optimization and experimental modal analysis. The other works present the issue of interactions between mechanical and hydraulic system of loader crane [4] or interaction with operator [5] and the environment [6, 7].

Loader cranes, during operation, are exposed to very high loads related to dynamic forces. These forces of various origins could cause a loss of stability or a decrease in accuracy at the end-point. The former was studied both in terms of structural [8,9] and dynamic stability [10, 11]. The other research on dynamic properties of loader cranes concerned tip over stability of mobile boom cranes [12]. The latter could be caused by stopping the movement of flexible manipulator that makes residual vibrations occur. Numerical and experimental studies on vibration control of single link flexible manipulators with payloads were presented in [13].

Although there are many works addressed to issue of loader cranes, dynamic analysis of structure is one of the most valuable. There are papers which consider analytical modelling of dynamic properties such as mode shapes and natural frequencies. The most popular method of numerical modelling of dynamic properties is the finite element method. Free vibration analysis of truck cranes and studies on changes in the frequency of vibration for different radii of the crane and its load

were presented in [14]. Paper [15] presents the results of numerical and experimental analysis of mode shapes and eigenvalues of the telescopic platform with 5 booms.

There are papers which present the results of experimental modal analyses of cranes. In [16], the authors received eigenvalues and mode shapes of a full-scale crane, and in [17] for smaller, laboratory one. Due to multiple flexible links, there are many translational contacts, which change in various configurations, so creating a full model of crane is very time consuming and it take a long time to solve. Authors in paper [18] present an efficient model by using the assumed modes method with consistent kinematics and suggest that using a low-order dynamic model is sufficient.

The description of the variable-configuration loader crane dynamic properties can be represented by a set of models corresponding to the distinguished configuration options associated with the characteristic points of the workspace. There can be a significant number of such distinguished variants; therefore, the dynamic model of the crane should be simplified as much as possible to reduce the building and computational time.

In the presented paper, a method of loader crane model reduction by replacing hydraulic actuators by substitute elements is presented. This simplification enables shortening both the time of building the model and solving. Calculations were carried out on Hiab XS 111 HI DUO, which was presented in Fig. 1.

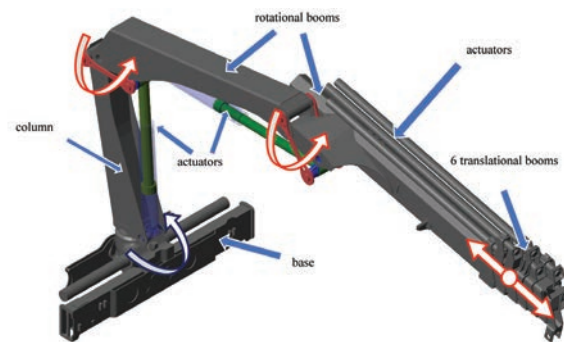


Fig. 1. Hiab XS 111 HI DUO structure

## 2. Modelling hydraulic actuators with FEM

The finite element method is the most common approach of modelling dynamic properties of loader cranes. It assumes the replacement of all solid parts and liquids with adequate finite elements. Structural elements like booms or columns can be modelled relatively easily using solid or beam finite elements. However, due to phenomena occurring in hydraulic oil, modelling hydraulic cylinders is a rather problematic task.

Loader cranes usually contain several hydraulic cylinders that control the movement of its individual components. In the conventional modelling approach,

the geometry of hydraulic cylinder (including hydraulic oil located in it) should be divided into a deliberate mesh of finite elements with appropriate properties and parameter values. Such an approach results in the necessity of building an actuator model with a significant number of degrees of freedom; moreover, each change in configuration, forces model re-discretization.

Therefore, a simplified model of the actuator was proposed, which does not require re-discretization but only changes in the values of the parameters that describe it. Simplifications consist in replacing the actuator model with an equivalent stiffness ROD

element and inertia elements, which depend on the actuator configuration. The concept of model simplifications is presented in Fig. 2.

The area of fixing the cylinder and piston to the construction was modelled with rigid elements, while the rod element and the mass element were stretched over their whole length. The rigid elements have been fixed in the way to allow the arms to rotate, while the parameters of the mass elements were based on the weight distribution related to the mutual configuration of individual hydraulic cylinder elements.

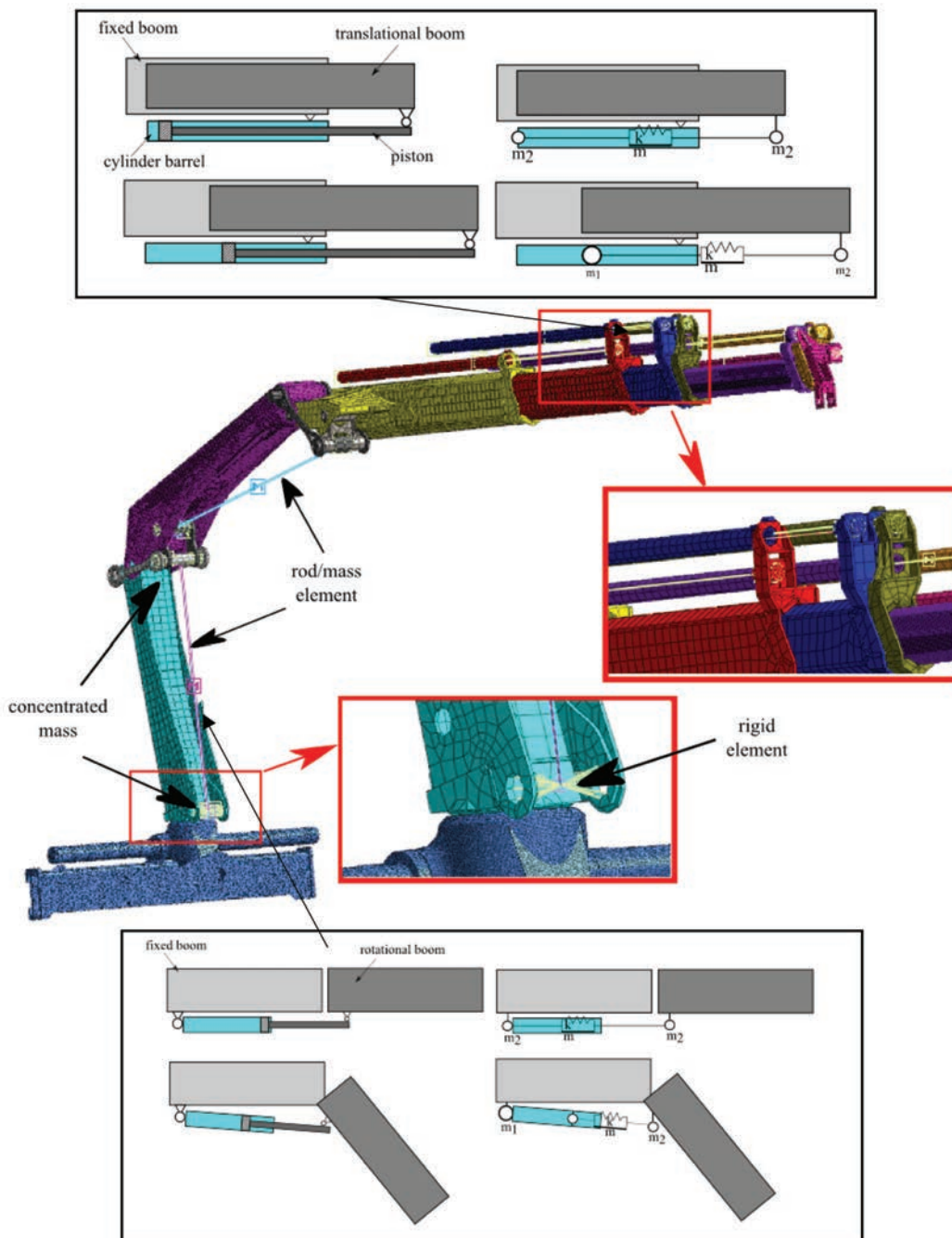


Fig. 2. The idea of actuator model simplifications

To determine the equivalent stiffness of the ROD element, the main factors affecting the stiffness of a hydraulic cylinder like the stiffness of the hydraulic oil  $k_o$ , the piston rod  $k_R$ , and cylinder  $k_C$  were considered. Due to the fact that, in the analysed crane structure, hoses were not flexible, its stiffness was omitted. Therefore, the equivalent stiffness the  $k$  of the ROD elements can be expressed as follows:

$$\frac{1}{k} = \frac{1}{k_o} + \frac{1}{k_R} + \frac{1}{k_C} \quad (1)$$

where:  $k_o$  – stiffness of the hydraulic oil;  $k_R$  – stiffness of the piston;  $k_C$  – stiffness of the cylinder.

According to the literature [19], the stiffness of the hydraulic oil contained in the cylinder (5) is composed of the stiffness of the oil in the chamber from the side of the piston and in the chamber from the side of the head, and it can be expressed as follows:

$$k_o = B \left( \frac{A_1}{V_1 + V_{L1}} + \frac{A_2}{V_2 + V_{L2}} \right) \quad (2)$$

where  $B$  – elastic modulus of the oil,  $A_{1,2}$  – the effective area the head chamber (1) and the rod chamber (2);  $V_{1,2}$  – the effective volumes,  $V_{L1,L2}$  – fluid line volumes.

The basic parameter of hydraulic oil is its elastic modulus  $B$  described by Equation (3). For the most commonly used hydraulic fluids and at a normal temperature (around 20°C), the elastic modulus  $B$  is close to the value of  $B_1 = 1500$  MPa [20]. However, it is important that the value of this parameter depends on its properties (e.g., aeration) and working conditions. Its value decreases with the increase of the temperature  $a_{\Delta t}$ , and in the case of air admixture in oil, but slightly increases with the pressure increase  $a_{\Delta p}$ , and it can be expressed as follows:

$$\frac{1}{B} = \frac{1}{B_1(1 + a_{\Delta p} + a_{\Delta t})} + \frac{\varepsilon}{P} \quad (3)$$

where:  $\varepsilon$  – is an oil aeration coefficient and  $P$  is the absolute pressure increased by pressure changes in chambers.

The stiffness of the piston rod results from the fact that it is a cylindrical bar, so its axial rigidity can be calculated from following formula:

$$k_R = \frac{EA_R}{L_R} \quad (4)$$

where:  $E$  – Young's modulus of piston rod material,  $A_r$ ,  $L_r$  – cross-sectional area and length of a piston rod respectively. The stiffness of a cylinder is understood

as both the stiffness associated with its extension and deformation, and it can be expressed as follows:

$$k_c = \frac{E_B A_c}{2L_c} \cdot \frac{1}{\lambda_c + \nu_b} \quad (5)$$

where:  $E$  – Young's modulus for the cylinder barrel,  $A_c$ ,  $L_c$  – cross-sectional area and length of the cylinder barrel, respectively,  $\gamma_c$  – expansion coefficient,  $\nu_b$  – is the Poisson ratio for cylinder barrel.

In order to pre-check the correctness of the suggested simplification, computational analyses were carried out for a simple three-boom model with two actuators (Fig. 3).

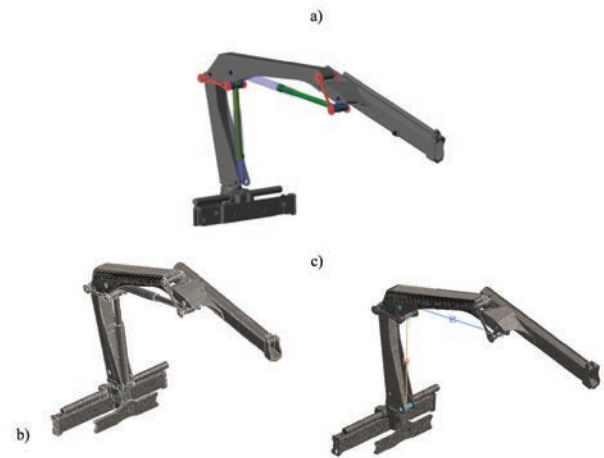


Fig. 3. Geometry (a) and discretized model of a simple tree-boom b) full model, c) simplified model

The tests were carried out for two variants. In one of them, the actuators were modelled using the classical approach. In the second, the actuators were replaced by presented substitute elements, choosing the values of their parameters, respectively. For such prepared models, a modal analysis was carried out, which resulted in the natural frequency of the crane for both variants. The values of the first five frequencies are presented in Table 1.

Table 1. Natural frequencies comparison between full and simplified model

| Full FEM model | Simplified FEM model | Relative error |
|----------------|----------------------|----------------|
| 5.8 Hz         | 5.4 Hz               | 6%             |
| 7.9 Hz         | 8.2 Hz               | 4%             |
| 19.7 Hz        | 19.3 Hz              | 2%             |
| 20.8 Hz        | 21.8 Hz              | 4%             |
| 47.0 Hz        | 45.7 Hz              | 3%             |

Differences between the obtained eigenvalues for both variants differ by no more than 6%. Therefore, it can be concluded that applying the suggested simplification is justified.

Therefore, in this convention, a model was created for the entire crane, and then the obtained results were compared with the results from experimental research. For entire crane, there were conducted analyses in two configurations. In both configurations, two rotational booms were in the same angle, but in the first configuration, designated ‘min’, all translational booms were retracted, and, in the second – ‘max’ configuration – chosen booms were partially extracted (Fig. 4). Incomplete booms extraction was caused by the space limitations during experimental research.

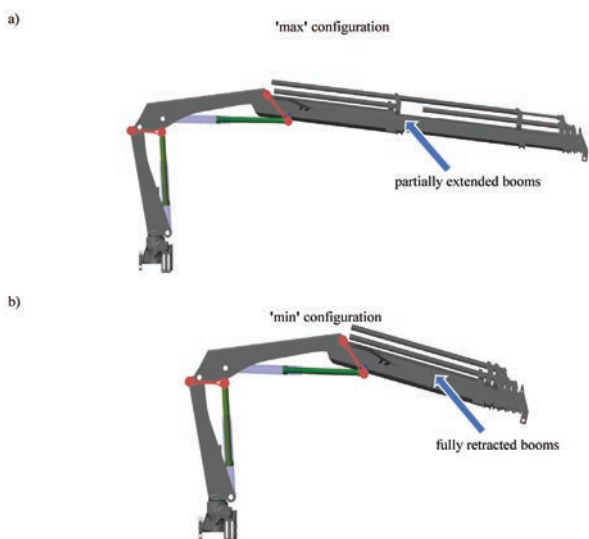


Fig. 4. Research object configurations a) extracted ‘max’ and b) retracted ‘min’

#### 4. Model Validation

To validate developed model, an experimental modal analysis in form of impact test was conducted for analysed variants. The experimental setup is presented schematically in the Fig. 5. Excitation was realized using a modal hammer with a 1.5 kg head mass. To obtain spatial mode shapes, the structure was excited at the end of the telescopic boom in three orthogonal directions. The response of the structure was measured using PCB 393A03 accelerometers, due to their high sensitivity level and accurate signal representation at the low frequency range. The experiment was performed using Siemens Testlab software and Scadas III hardware and included data processing, monitoring power, spectral density, and coherence functions. The detailed procedure was presented in [16].

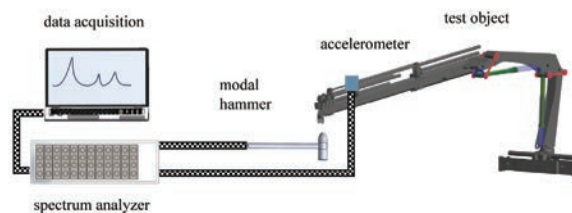


Fig. 5. Experimental setup

On the basis of determined frequency response functions, the modal model was built using a Polymax algorithm. The estimation process was supported by the stabilization diagram interpretation. Obtained values of natural frequencies are presented and compared by  $\delta$  ratio with FEM model results in Table 2.

Table 2. Comparison of natural frequencies obtained from model and experiment for both variants

| Mode number | Variant „MAX” |              |          | Variant „MIN” |              |          |
|-------------|---------------|--------------|----------|---------------|--------------|----------|
|             | Experiment    | FEM analysis | $\delta$ | Experiment    | FEM analysis | $\delta$ |
| 1           | 0.92 Hz       | 0,87 Hz      | 4%       | 1.93 Hz       | 1.98 Hz      | 3%       |
| 2           | 1,41 Hz       | 1,34 Hz      | 7%       | 3.09 Hz       | 3.42 Hz      | 11%      |
| 3           | 4.87 Hz       | 5.21 Hz      | 1%       | 7.11 Hz       | 6.97 Hz      | 2%       |
| 4           | 7.26 Hz       | 7,20 Hz      | 1%       | 9.58 Hz       | 9.97 Hz      | 4%       |
| 5           | 13,67 Hz      | 14,26 Hz     | 4%       | 13.29 Hz      | 12.66 Hz     | 5%       |

$$\delta = \frac{|f_{iFEM} - f_{iEXP}|}{f_{iEXP}} \cdot 100\% \quad (6)$$

where  $f_{iFEM}$  is sequent frequency from FEM analysis, and  $f_{iEXP}$  is sequent frequency from experimental research. Comparisons of the first mode shapes are presented in Fig. 6.

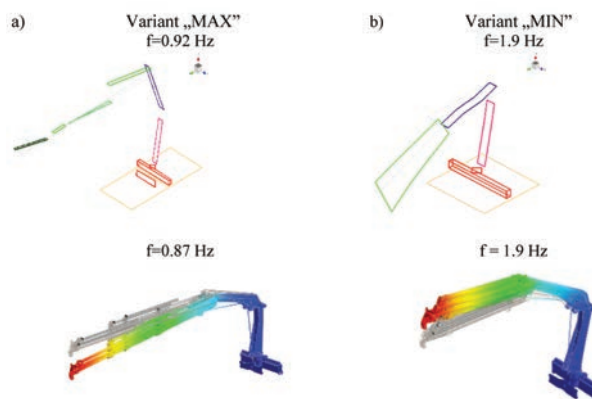


Fig. 6. Comparison of first mode shapes for both variants: a) “MAX” variant, b) “MIN” variant

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## Conclusions

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The dynamics of loader cranes can be represented by a set of configurable models. In the classical FEM method, for each configuration, re-discretization is required. The presented method of modelling allows conducting analyses in various configurations on the same mesh. The use of simplified elements in place of hydraulic cylinders allows one to shorten the time of the model building, as well as reduce its dimensionality and thus the time of calculations. Searching for such simplifications in the case of reconfigurable structures, for which a new model needs to be built for each configuration, is particularly reasonable. Owing to the shortening of the calculation time, it is possible to perform a computational analysis of a larger number of configurations in order to obtain the most complete description of the dynamic properties of the considered crane. Based on the results presented in the paper, it can be concluded that, despite the simplifications used, a high model compliance with the experimental results was obtained. The greatest difference in frequency values between experimental and numerical one is 7%, and the mean value of disagreement is 4%. However, it should be taken into account that a very important step is an appropriate selection of the equivalent parameters of the mass and stiffness elements of an actuator.

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