SIMULATION OF MECHANICAL PARAMETERS OF SPRAYER BOOM

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Abstract. The sprayer boom is a large and complex structure, used to support the spray nozzles. It is important to control and minimize the vibration of the structure on the vertical and horizontal planes, in order to insure the uniformity of pulverization over the field. Owing to the large spray boom widths, flexible behaviour of spray booms becomes important. Even small deformations of the structure can cause, by the large spray boom width, considerable displacements at the boom tips. However, there is not a clear correlation between the boom width and the amount of the spray boom motion. Another important factor is whether it concerns a trailed or a mounted sprayer. As spray boom motions play a dominant role on the spray distribution pattern, spray boom stability is important. Theoretical studies, simulations and field experiments have indicated that due to the spray boom vibrations, spray deposit distribution varies between 0 and 80 %. In this paper the static and dynamic study of a sprayer boom structure of about 24 m length is presented. Starting from the CAD model of the sprayer boom structure, a standard mesh procedure as a preprocessing step of a finite element analysis has been followed. Mainly shells, a reduced number of solid elements and rigid connection elements were used. The purpose of the static linear analysis was to determine the state of tension and deformation that develop in the boom during its exploitation. The geometric model made for the linear static analysis using the finite element method comprises several steps and highlights the mechanical response of the structure. An important parameter of the dynamic behaviour is the boom vibration amplitude. This parameter was observed in the finite element analysis of the boom structure considered at a natural scale. Other similar parameters, such as the resonant frequencies, have been observed.

Keywords: boom, simulation, finite element analysis.

Introduction

In the last decades, the width of boom sprayers has increased drastically. Nowadays, sprayers having a width of 45m are commercially available. Owing to the large spray boom widths, flexible behaviour of spray booms becomes important. Even small deformations of the structure can cause, by the large spray boom width, considerable displacements at the boom tips [1].

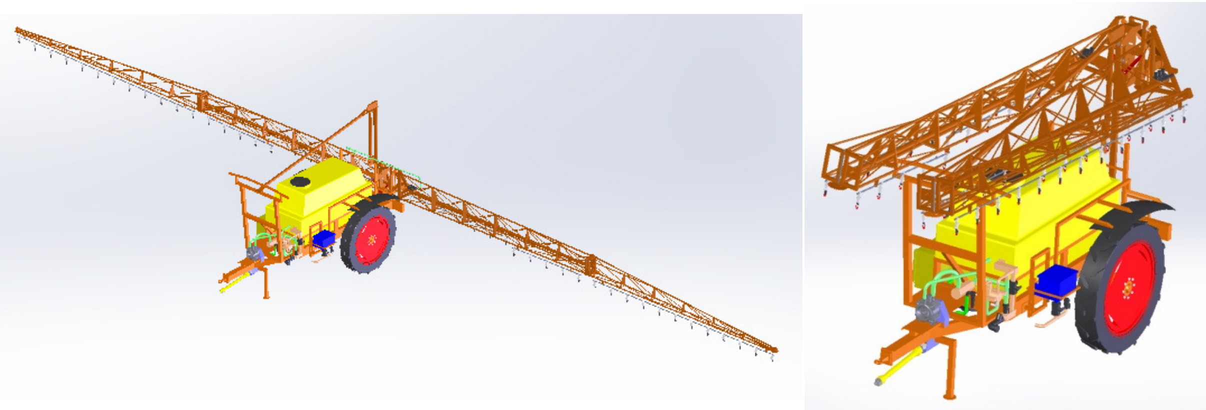
Theoretical studies, simulations and field experiments have indicated that due to the spray boom vibrations, spray deposit distribution varies between 0 and 80 % [2; 3]. Structure dynamics is a very broad discipline, which uses a huge arsenal of theoretical and experimental methods to solve a fundamental problem of structures: the dynamic response to variable tasks over time [4- 8]. Vibrations and especially vibration in resonance modes are problems that occur frequently in large structures. Because large structures with large numbers of components cannot be optimally engineered for resonant regimes, it is often done to modify the structures or improve them using the modal analysis of these structures [9].

In the paper [10], the dynamic study of a sprayer boom structure of about 12m length on each side was presented. Initially, the real boom has been optimized in terms of minimizing the vertical vibration, considering the dynamic model of the whole sprayer mechanism excited from the ground, when it is following a standard bumpy path.

In this paper the static and dynamic study of a sprayer boom structure of about 24 m length is presented. The purpose of the static and dynamic linear analysis was to determine the state of tension and deformation that develops in the boom during its exploitation. The linear static analysis using the finite element method comprises several steps and highlights the mechanical response of the structure. An important parameter of the dynamic behavior was the boom vibration amplitude. This parameter was observed in the finite element analysis of the boom structure considered at a natural scale. Other similar parameters, such as the resonant frequencies, have been calculated.

Materials and methods

The analyzed structure is a boom of a trailed spraying machine (Figure 1), with a working width of 24 m, used to apply phytosanitary treatments and fertilization in field crops. The finite element method was used, working in the SolidWorks 2013 Simulation package [11]. Solid Works Simulation contains real-time design, testing, and simulation tools (the product is subjected to the same conditions it will endure in real life), leading to an accelerated design process, increased design quality and productivity, at the same time lowering the cost of testing prototypes before proceeding with the manufacturing process.



a) working position b) folded boom for transport

Fig. 1. **Trailed boom sprayer – 3D model**

The first step of this study was the static linear analysis. The purpose of static linear analysis was to determine the state of tension and deformation that develops in the boom of the spraying machine during its exploitation. The geometric model made for the linear static analysis using the finite element method comprises several steps, which will be presented below, and highlights the mechanical response of the structure. Due to the symmetry of the geometric model, only half of it was considered for the static analysis (Figure 2). From the point of view of the location of the component elements of the investigated structure, it was considered that they are positioned according to the real case of its operation in the field.

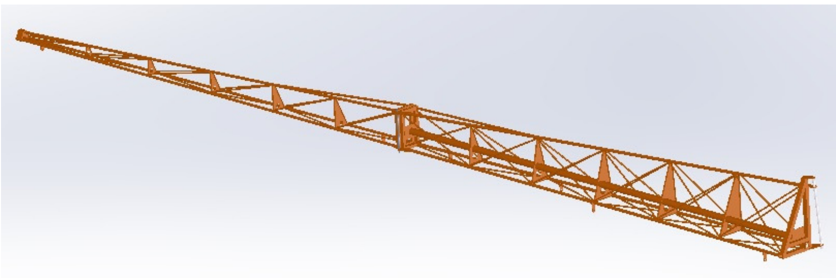
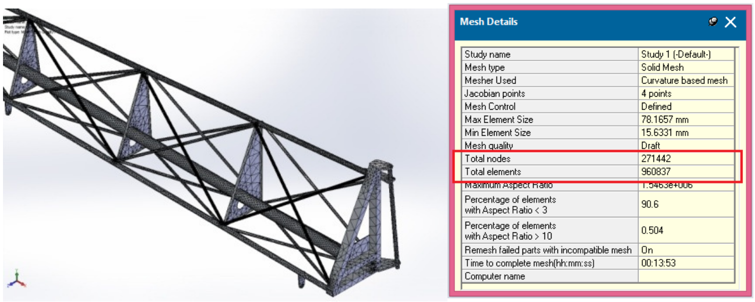


Fig. 2. **Geometric model of analyzed structure** (half boom, 12 m width)

The discretization of the boom structure was made by modifying certain parameters so that the finite element network obtained from the meshing process is a continuous network to be adapted to the geometrically developed model without obtaining distorted areas that could have led to incorrect results. The finite element network obtained after the meshing process is shown in Figure 3.

Analyzing Figure 3, it can be seen that the dimensions of the mesh network elements are relatively small. This was desirable in the meshing process to make the model optimal and to make a mathematical model as accurate as possible. Thus, the condition that the smallest dimension of an element be equal to the smaller geometric dimension corresponding to a component of the assembly was imposed. The finite element network resulting from the meshing process consists of 960837 elements and 271442 nodes. The types of finite elements used in the meshing process are three-dimensional Solid type elements.

The material of the boom structure is S235JR steel, with hypothetical linear, elastic, homogeneous and isotropic behaviour. The characteristics of the S235JR steel that were used for the analyzed structure, taken from the SolidWorks material database are the following: elastic modulus 2.10000003 + 011 N·m^{-2}, Poisson's ratio 0.28; mass density 7800 kg·m^{-3}, tensile strength 360000 kN·m^{-2} and yield strength 235000 kN·m^{-2}.



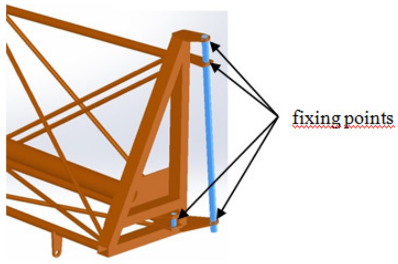


Fig. 3. **Meshline appearance and meshing parameters** Fig. 4. **Boom structure fixing points**

The next step was to establish the fixing points and apply the external loads. The boom structure was fixed at its points of attachment to the central folding / unfolding device (Figure 4). The loading of the structure is pure gravitational (its own weight), the gravitational acceleration having the value - 9.81 m·s^{-2}, on the vertical direction.

Results and discussion

Figure 5 presents the main results of **linear static analysis**, namely: von Misses equivalent stress, relative static displacement (URES), strain (ESTRN), and factor of safety (FOS) distribution. It is observed that the maximum equivalent stress (2.29·10^{8} N·m^{-2}) is less than the yield strength (2.35·10^{8} N·m^{-2}) of the structure material. Under its own weight, the maximum displacement of the structure is approximately 7 mm, corresponding to the boom end. Factor of safety plot shows min FOS is 1.02.

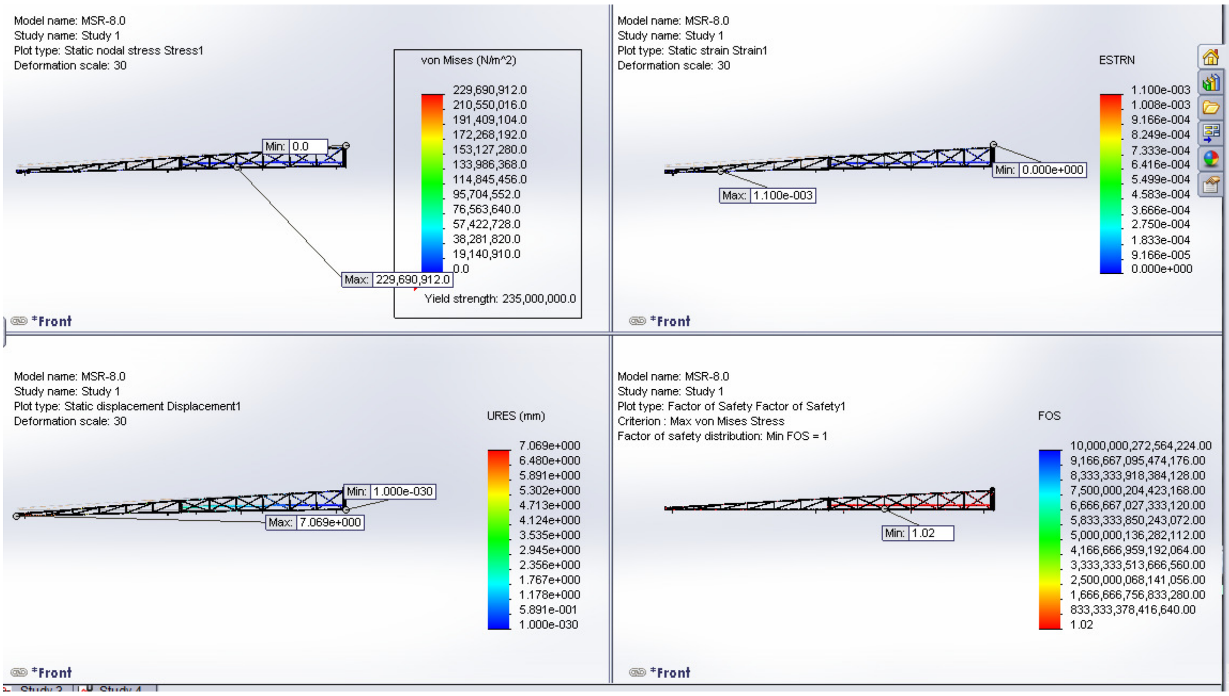


Fig. 5. **Results of static linear analysis**

The **dynamic frequency response analysis** performed in this study had as its main objective the determination of its own frequencies in all the constituent elements of the projected structure. The result of calculating frequencies or modal analysis is essentially in the list of a number of its own frequencies (in Hz) (or pulses in rad·s^{-1}), in the order starting from the lowest (fundamental frequency). There are also given the relative displacements on the directions and the resultant in the structure, for each vibration mode, separately. Also, colour maps of the field of relative displacements, component or resultant displacements were given. In Table 1 from the Solidworks program report, with which, using the Simulation module, frequency analysis was performed, a list of the first five frequencies corresponding to five vibration modes is presented.

Table 1

The first five frequencies corresponding to five vibration modes

Vibration mode	Frequency, Hz	Pulses, rad·s^{-1}
1	1.958	12.303
2	4.2053	26.422
3	6.2908	39.526
4	13.364	83.968
5	17.698	111.2

As it is known, the main result of the modal analysis is the set of the calculated frequencies. In principle, it may be required to calculate a bigger number of its own frequencies. In fact, just the first few are useful. The most important is the fundamental frequency, which has the lowest value of the calculated ones. Most of the times, the list of the own frequencies is used to avoid resonant work regimes and, in general, resonance phenomena that may occur in various circumstances. In Fig. 6-10 the own frequencies and amplitude maps are given on the deformed shape of the structure in the vibration modes corresponding to each frequency.

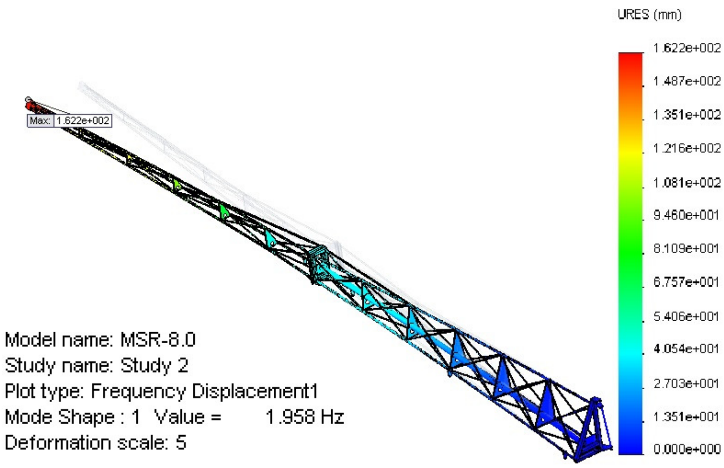


Fig. 6. **Displacement map on the deformed shape of the boom for vibration mode 1, 1.9580 Hz**

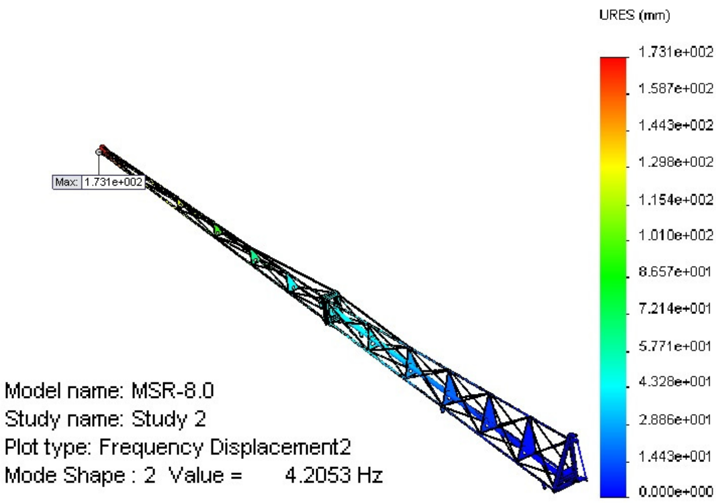


Fig. 7. **Displacement map on the deformed shape of the boom for vibration mode 2, 4.2053 Hz**

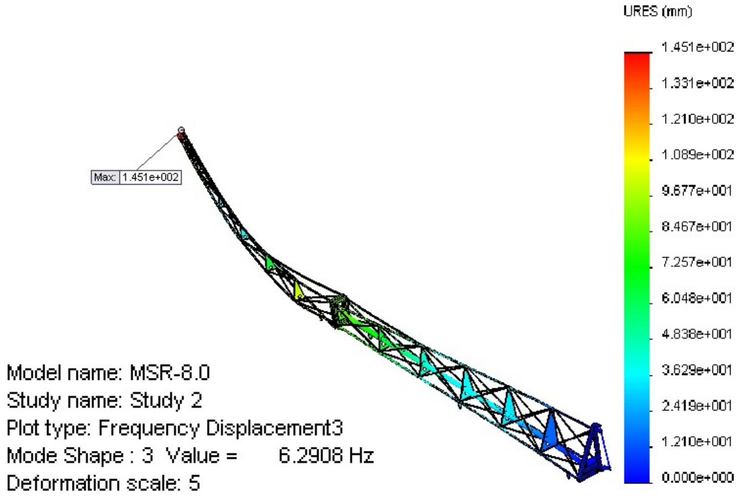


Fig. 8. **Displacement map on the deformed shape of the boom for vibration mode 3, 6.2908Hz**

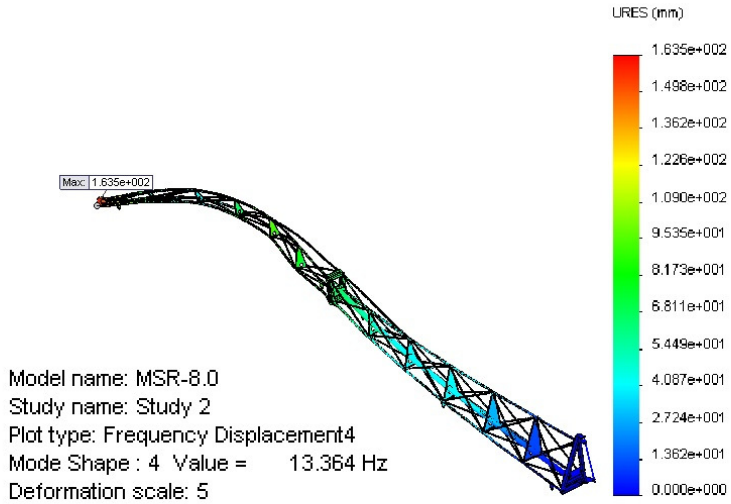


Fig. 9. **Displacement map on the deformed shape of the boom for vibration mode 4, 13.3640 Hz**

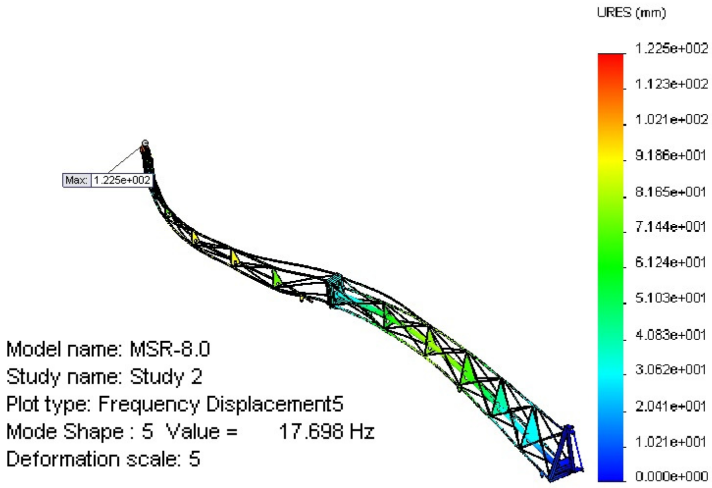


Fig. 10. **Displacement map on the deformed shape of the boom for vibration mode 5, 17.6980 Hz**

For the boom structure analyzed in this paper, we limited the number of frequencies calculated to 5. We considered that this way we cover all the basic frequencies that can occur in the working process of the spraying machine for different working speeds. The highest frequencies we have considered were those that originate, usually from the uneven field (Figure 11).

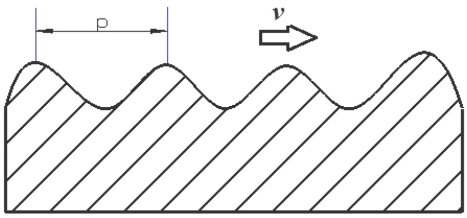


Fig. 11. **Appearance of uneven terrain**

Assuming that these bumps follow an approximately constant interval noted with *p*, we can calculate their frequency with the formula (1):

$$f_i = \frac{v_i}{\overline{p}}, \text{ s}^{-1} \tag{1}$$

where v_i – working speed of the tractor - sprayer aggregate, m·s^{-1};
$\overline{p}$ – average value of the distances between two successive field misalignments, m.
(in this study, we considered $\overline{p}$ = 0.3 m, to simplify the calculus).

The diagram in Figure 12 shows the variation in the critical operating speed, depending on the resonance frequency for each vibration mode in Table 2. During working with the sprayer, the critical speed values shown in the diagram in Figure 12, or values close to them must be avoided. The reasons for avoiding the working speeds corresponding to the resonance frequencies are related to the serious damage to the boom and to unevenness of spraying on the ground.

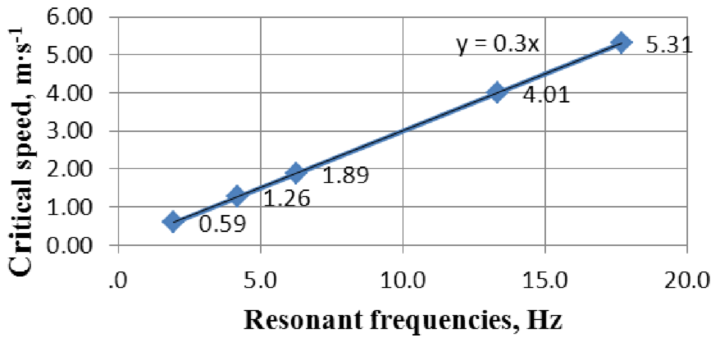


Fig. 12. **Variation of critical speed, depending on resonance frequency**

It is known from the practice and from the literature [2; 3], that the working speed of a spraying machine is in the range of 1.6 ÷ 3.3 m·s^{-1} (6 ÷ 12 km·h^{-1}). According to the algorithm presented above, the resonance frequencies will be between 5.3 Hz and 11 Hz. This range of resonant frequencies ranges between the vibration modes 2 (4.2053 Hz) and 4 (13.364 Hz), presented in Fig. 6-10, demonstrating the utility of the virtual simulation of the boom sprayer.

Conclusions

Following this structural study on the 24 m sprayer boom, several important conclusions can be drawn for further investigations. There was no risk of the structure material failure, because the value of von Misses maximum equivalent stress (2.29·10^{8} N·m^{-2}) was less than the yield strength (2.35·10^{8} N·m^{-2}). Also, the maximum relative displacement values (7 mm), corresponding to the boom end, guarantee that deviations from the working parameters dictated by agrotechnical requirements (e.g. working speed) are negligible. The value of the safety coefficient (1.02), in relation to its usual values in the practice of designing and manufacturing boom sprayers, shows that there is an important potential for optimizing this type of boom sprayers.

The analysis of the critical speed dependence on the resonance frequencies that can occur in the boom sprayer is particularly useful for designers and users of such machines. The critical speed values (0.59; 1.26; 1.89; 4.01; 5.31 m·s^{-1}), or values close to them must be avoided.

The usefulness of this analysis is particularly evident in the test phase and even in the first stages of operation, when the working regime of a boom sprayer should be improved.

Acknowledgements

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