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Modelling, simulation and optimization of agricultural sprayer boom horizontal motion behaviour

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The study reported here presents a method offering realistic depiction of horizontal motion behaviour of agricultural sprayer booms and for reducing the extent of the simulation model involved. Additionally, various solutions of reducing boom vibrations are presented and compared with the help of simulation. Thereby is demonstrated the extent to which boom motion behaviour on booms with large working widths can be improved through passive vibration absorbers and an active vibration isolation.

Keywords
Plant protection, sprayer boom, vibration damping, simulation

In crop spraying there has developed a clear trend towards more area performance and more individual, precise, and thereby cost-efficient and environmentally friendly, operations. Greater area coverage is enabled through, among other measures, increasing volume capacity of sprayer tanks, higher application speeds, faster adjustment capability to cope with changing field conditions, and increased working widths in the form of boom length. However, increasing boom scale and mass means more stress on mechanical components, which also alters motion behaviour of boom towards lower natural frequencies so that greater vibration amplitudes may occur (Schranz 2014).

Sprayer boom movement relative to the base implement in direction of travel, caused by permanent vibration impulse and boom material elasticity, leads to uneven application of spray.

Regarding further increases in boom widths and particularly spray application improvement, it is necessary to further optimise motion behaviour of sprayer booms towards minimizing application rate of active ingredients and thus increasing economic efficiency and environmental friendliness of the spaying equipment.

Developing a simulation model

Simulation with Matlab/Simulink software was used in comparing various solutions towards optimising sprayer boom motion behaviour. A model was developed with the aim of depicting as precisely as possible sprayer boom dynamics while keeping calculation time as short as possible. Thereby, the theory of modal transformation and reduction was applied, producing a model of a linear substitution system with limited degrees of freedom.

A sprayer boom represents a continuous oscillator, the characteristics of which can be described alternatively as a more-mass-oscillator with finite degrees of freedom through the linear, coupled equation system (Herold 2003).

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$$M\ddot{x} + D\dot{x} + Kx = F \qquad \text{(Eq. 1)}$$

The mass matrix *M* and the stiffness matrix *K* depict, hereby, the mechanical characteristics, the damping matrix *D* accounts for the damping within the structure through proportional dependence to the mass and stiffness matrices. The joints between the individual boom elements are accepted in ideal case as rigid, so that only the structure of the components influences the damping behaviour of the sprayer boom.

$$D = \alpha M + \beta K \qquad \text{(Eq. 2)}$$

To reduce model complexity, the complex geometry of the individual boom elements was abstracted through beam elements with six degrees of freedom and symmetrical cross sections constant over the beam length – describing the mass, the stiffness and dynamic behaviour of the boom elements. The stiffness and mass matrices were applied for every individual element, and their combination led to the total stiffness and mass matrices for the boom. The geometrical measurements of the beam cross sections, as well as the young's modulus, were determined for every individual boom element iteratively. For this, the first ten eigenfrequencies and eigenforms of an element are calculated, determined and compared with the here-described model as well as a finite element.

Towards further reduction, the combined differential equation system is transformed with the help of modal transformation into a system of linear decoupled equations in the modal area. Modal transformation in this case means the left and right sided multiplication of the movement equation with the modal matrix. Hereby, the first step of the general eigenvalue problem

$$[K - \omega_i^2 M]u_i = 0 \quad i = 1 \dots n \qquad \text{(Eq. 3)}$$

is solved. From this follows the modal matrix *U*, the columns of which are constructed from the eigenvectors u_i, as well as the eigenvalue matrix ω with the eigenvalue ω_i^2 on the main diagonals.

The subsequent coordinate transformation

$$x = Uq \qquad \text{(Eq. 4)}$$

and the left-hand multiplication of the general movement equation with U^T led onto a system of *n* decoupled differential equations with, in each case, a modal degree of freedom.

$$U^T M U \ddot{q} + U^T D U \dot{q} + U^T K U q = U^T F \qquad \text{(Eq. 5)}$$

Through the modal transformation, emerges a system of decoupled differential equations. However, the model still includes *n* degrees of freedom, whereby the modal transformation for systems with a great number of degrees of freedom is not practicable on its own.

To minimise the numbers of degrees of freedom, and thus model complexity, the model is modally reduced whereby all eigenvalues above a threshold frequency ω_{max} are not taken account of in the model. This action represents an approximate solution because only the super positions of all eigenmodes describe the dynamic behaviour completely. Guideline for a good approximate solution is where the threshold frequency ω_{max} corresponds to 1.5 times the investigated frequency Ω_{max} (Ihlenburg 2013). Usually, Ω_{max} is the largest relevant frequency within the stimulation spectrum.

Model validation

For model validation, a trial was conducted in the lower frequency range with a test stand especially developed for vibration analysis of agriculture sprayer booms. Thereby, a field sprayer in direction of travel is stimulated by a frequency sweep with a constant displacement and in a frequency range from 0 to 3 Hz. This was a stationary test whereby boom movement is determined through measuring displacement on different points within the boom. The trial was reproduced in the simulation whereby the measured excitation function was adopted and the displacements at the measuring points compared. For final validation of the model, the damping factors α and β were adapted.

The following figure represents the comparison of the displacements at the end of the boom as standardized and shows the envelope curves of measured and calculated vibrations.

Figure 1 presents the comparison of the displacements at the end of the boom as standardised, and shows the envelope curves of the measured and calculated vibrations: hereby achieving a very good agreement between measurement and simulation. Comparison of the maximum vibration amplitude shows a maximum deviation of 1.4 % of the calculated to the measured amplitude. Additionally, a frequency analysis in the investigated lower frequency range showed the maximum deviation of the eigenfrequency as 0.6 %. However, it was also clear that the actual sprayer boom showed a non-symmetric motion behaviour, which is not shown in the model depicted here. This can be explained through the folding joints between the boom elements, which were not considered in the calculation model, the geometric construction of which caused a non-linear motion behaviour of the sprayer boom.

In total, the simulation model, although representing an approximate solution, was sufficiently precise in depicting the dynamic behaviour of the sprayer boom so that various solutions could be investigated towards optimising motion behaviour.

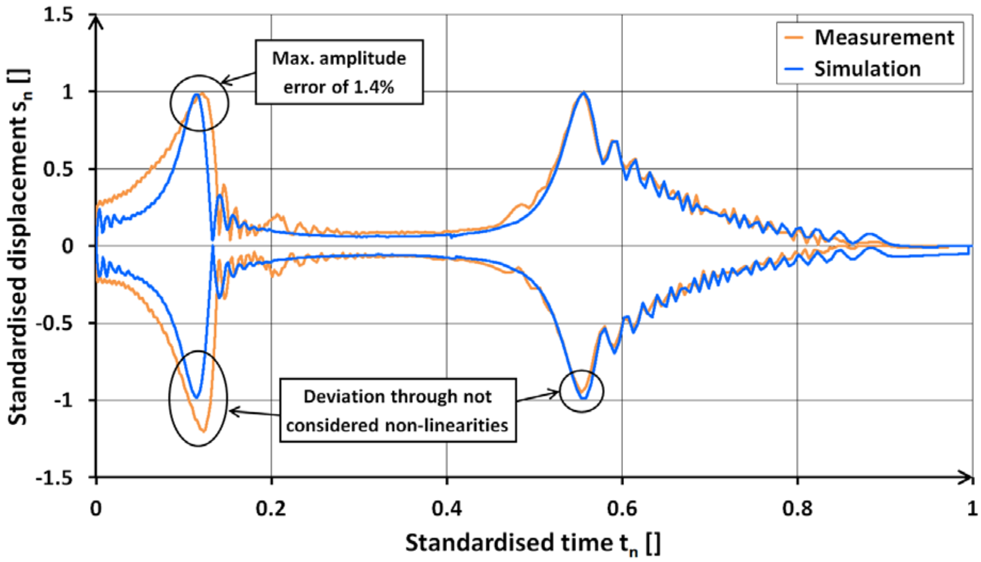

Figure 1: Standardised comparison of the envelop curves of displacement at the end of the boom

Investigation of passive vibration absorbers

If large resonance amplitudes are present at a certain excitation frequency, passive vibration absorbers can be useful for reducing vibration amplitudes. In this case, spring-mass-systems were additionally positioned on the construction part to be damped, and the eigenfrequency of the spring-mass-system is coordinated according to the absorbing frequency (Buck 2005).

To optimise the motion behaviour of the agriculture sprayer boom in this study, two absorbers were applied per boom. These were positioned at the end of both outer boom elements as shown in figure 2.

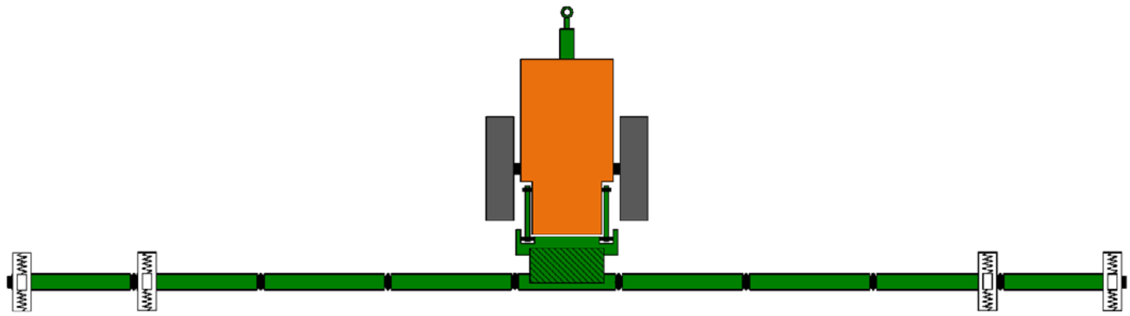

Figure 2: Crop sprayer with passive vibration absorbers at the end of the outer boom elements.

The absorbers have a degree of freedom in direction of travel and are coordinated according to the bending eigenfrequency of the spray boom whereby the outermost of each two absorbers is set-up for the first pronounced bending eigenfrequency in direction of travel, and the inner one for the second.

In this investigation, the frequency sweep described in the previous section was simulated for validating the model.

The result of the test is presented as standardised in the following figure and shows the envelope curves of the vibration occurring at the end of the sprayer boom in comparison to the starting situation without absorbers.

Figure 3 shows the typical pronounced movements of so-called side bands where passive absorption is applied. Thereby are meant the much smaller amplitudes by slightly lower and higher frequencies compared with the eigenfrequency of the system up to that point. At the outer end of the sprayer boom, the maximum vibration amplitude is reduced through the application of the absorbers by approx. 40 %. The graphic also shows, however, that sprayer boom vibrations cannot be completely avoided.

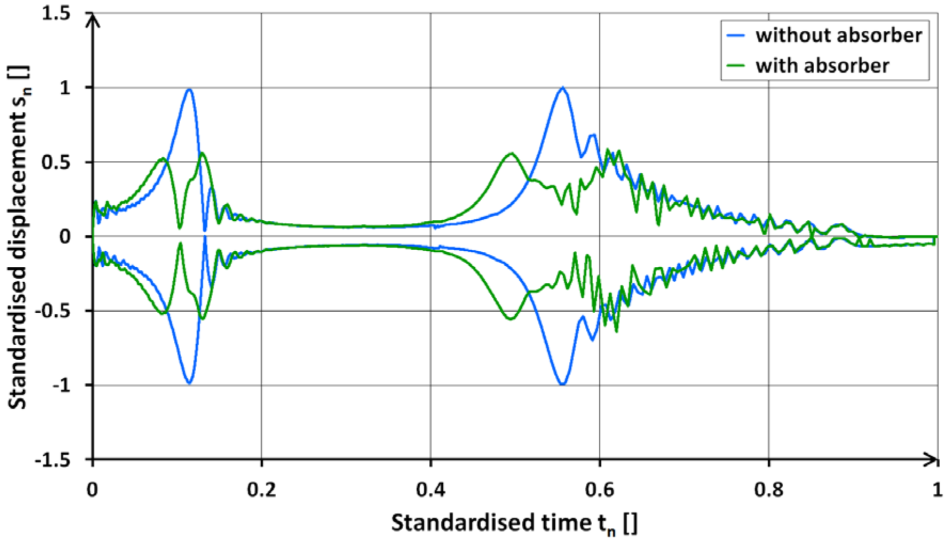

Figure 3: Standardised comparison of the envelop curves of displacement at the end of the boom

Investigation of an active vibration control system

The investigation of passive vibration absorbers showed that passive methods of reducing vibration amplitudes on the boom were effective. However, it was also clear that the vibrations were not able to be completely eliminated.

This led at this stage to the introduction of an active system for minimising vibration amplitudes representing a fully active system whereby the boom at the central point of attachment became a degree of freedom in direction of travel and an actuator effective in direction of travel is positioned between sprayer and boom.

In that this applies to a sprayer boom in a figurative sense and involves two one-sided tensioned beams showing hardly any damping properties of their own, it is very difficult to achieve active reduction of boom vibrations in the area of the respective attachment points when these are already taking place. The strategy applied for active damping therefore features an actuator placed between basic sprayer and boom that actively counters the sprayer movements thus avoiding transfer of movements onto the boom. At this point no detailed description of the algorithms involved is given. Used again for testing the system is the above-described frequency sweep as comparison load case. The following figure shows first results established under "ideal conditions" in the form of the produced vibrations at the outer ends of the boom standardised in comparison with the starting condition.

Figure 4 shows that the system resulted in only very limited movements being transferred to the boom and that these barely led to vibrations.

However, the frequency sweep investigated here does not take into account the dynamic behaviour of the sprayer and represents only a fault load condition. Therefore, investigation of further load change conditions such as road transport, braking, or field operations with the respective load changing condition disconnected is necessary for more precise assessment of the potential of an active vibration control system.

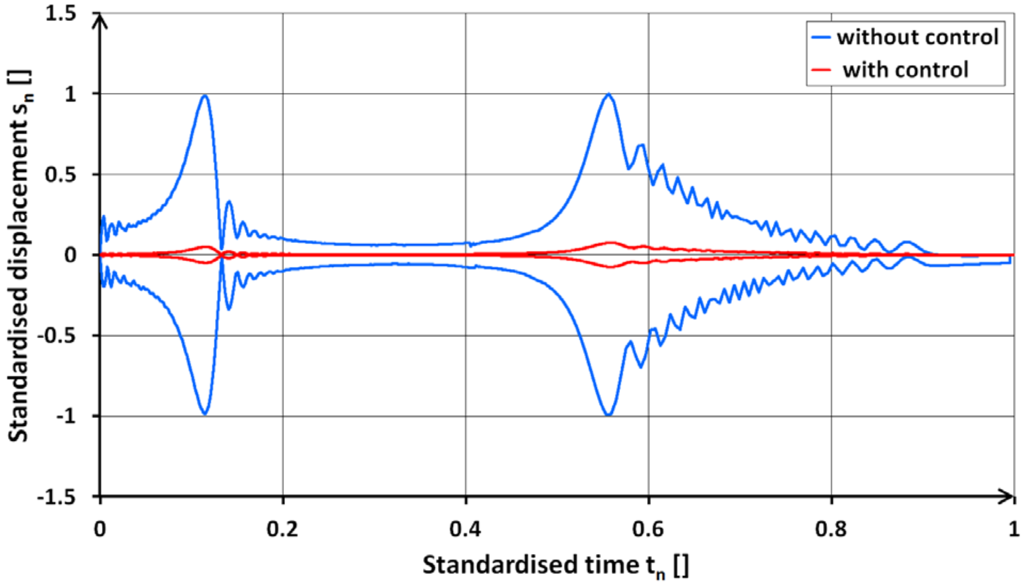

Figure 4: Standardised comparison of envelope curves of displacement at the end of the boom

Conclusions

In the first part of this report modal transformation and reduction is presented as a method for showing horizontal motion behaviour of agricultural sprayer booms and for the reduction of simulation model size.

Comparison of calculated results with those of a validation trial shows that the abstracted calculation model using Matlab/Simulink depicted the real system sufficiently accurately so that it could be applied for comparing various solutions towards optimising motion behaviour.

The second part of this report features comparison of two solutions towards reducing boom vibrations. First investigated are passive absorbers positioned on the outer elements of the boom and adapted for the first two bending eigenfrequencies of the sprayer boom. Simulation of the absorbers showed that the vibration amplitude at end of sprayer boom could be reduced by up to 40 %, but not completely eliminated. An active vibration control system was also presented. This followed the strategy of not conducting sprayer movements into the attached boom by applying actuator-counteraction of sprayer body movements. First calculation results show an optimal decoupling of the vibration disturbances and therefore the great potential of this solution. For this reason, continuation of the project will feature detailed investigations into application of the solutions presented here, with preparation of a multibody simulation model of the sprayer boom including depiction of the real geometry and the elasticity of the boom elements as well as various load cases reflecting real operational conditions.

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