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TRANSMISSION OF VIBRATIONS THROUGH LINKAGE SYSTEMS OF AGRICULTURAL MACHINES

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ABSTRACT

Vibrations produced by agricultural machines are transmitted to the operator seat, inducing comfort and safety problems. The dynamics of the complex mechanical system, composed by the tractor with the operating machine, must be considered to predict the vibrations transmitted to the operator seat. In this paper, models of the tractor, of the typical linkage systems and of mounted or trailed operating machines are proposed. The substructuring approach is exploited to obtain a model of the complete mechanical system by coupling the models of the component subsystems. Transmissibility and inertance frequency response functions are computed to evaluate the effect of the linkage system dynamics on the vibrations of the operator seat.

Key words: Dynamic substructuring, Vibrations, Transmissibility, Agricultural machinery, Agricultural tractors

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1. INTRODUCTION

Vibrations produced by the agricultural machines are transmitted to the seat of the operator [1], inducing comfort and safety problems [2] [3]. Furthermore they are always associated with annoying sound emission [4]. The main vibrations sources are the soil irregularities and the operation of the implement connected to the tractor [5]. In both cases the dynamics of the complex mechanical system, composed by the tractor with the operating machine, must be considered to provide a reliable prediction of the vibrations transmitted to the operator seat.

In this paper, models of the tractor, of the typical linkage systems and of mounted or trailed operating machines are proposed. The substructuring approach [6] is exploited to obtain a model of the complete mechanical system by coupling the models of the component subsystems. The linkage systems are modeled by finite elements methods and reduced by the Craig-Bampton method [7]. The reduced order models are used to perform a frequency-based coupling of the substructures [8]. Transmissibility and inertance frequency response functions are computed in order to evaluate the effects on the vertical vibrations of the operator seat, of the soil irregularities, and of the vibrations generated by the operating machine.

2. THEORETICAL BACKGROUND

The analysis of the dynamic behavior of complex mechanical systems can be performed using the dynamic substructuring. One of the main advantage in the use of this technique is an easier investigation of the effects of local modification on the global dynamics. Moreover, these techniques allow to use a different type of model for each component substructure (i.e. lumped parameters models, FE models, experimental modal models).

2.1. Dynamic Substructuring

For a system made of n coupled subsystems, the equation of motion of each subsystem (r) may be expressed as:

$$\mathrm{M}^{(r)}\ddot{u}^{(r)} + \mathrm{C}^{(r)}\dot{u}^{(r)} + \mathrm{K}^{(r)}u^{(r)} = f^{(r)} + g^{(r)} \qquad (1)$$

where:

- $\mathrm{M}^{(r)}$, $\mathrm{C}^{(r)}$ and $\mathrm{K}^{(r)}$ are the mass damping and stiffness matrices;
- $u^{(r)}$ is the vector of displacements;
- $f^{(r)}$ is the vector of external forces;
- $g^{(r)}$ is the vector of internal constraint forces (i.e. the connecting forces with other subsystems).

The equations of motion (1) of all the n subsystems to be coupled can be gathered in a block diagonal format as:

$$\mathrm{M}\,\ddot{u} + \mathrm{C}\,\dot{u} + \mathrm{K}\,u = f + g \qquad (2)$$

where $\mathrm{M} = \mathrm{diag}(\mathrm{M}^{(1)}, \dots, \mathrm{M}^{(n)})$ and so on.

The compatibility conditions require any pair of matching DoFs (e.g. DoF l on subsystem (r) and DoF m on subsystem (s), to have the same displacement ($u_l^{(r)} - u_m^{(s)} = 0$). All these conditions can be expressed in a matrix format as:

$$\mathrm{B}\,u = 0 \qquad (3)$$

where each row of B defines the compatibility of a pair of matching DoFs. Note that the matrix B is generally a signed Boolean matrix. The equilibrium conditions require the sum of internal constraint forces, arising from the compatibility conditions, to be zero (.e.g. $g_l^{(r)} + g_m^{(s)} = 0$): this holds for any pair of matching DoFs. Moreover, for internal DoFs k on subsystem q, that are not connecting DoFs, it must be $g_k^{(q)} = 0$. All these conditions can be expressed in a matrix format as:

$$\mathrm{L}^T g = 0 \tag{4}$$

where the matrix L is a Boolean localization matrix and its number of column is equal to the difference between the total number of DoFs of the uncoupled structure and the number of pairs of matching DoFs.

Equations (2), (3) and (4) define the three field formulation, i.e. the total system of equations describing the coupling among substructures:

$$\begin{cases} \mathrm{M}\,\ddot{u} + \mathrm{C}\,\dot{u} + \mathrm{K}\,u = f + g \\ \mathrm{B}\,u = 0 \\ \mathrm{L}^T g = 0 \end{cases} \tag{5}$$

The direct substructuring problem (coupling) can be solved using either primal or dual assembly. Using primal assembly, a unique set of interface DoFs is selected and the internal constraint forces are eliminated by automatically satisfying the equilibrium. This is obtained by stating that:

$$u = \mathrm{L}q \tag{6}$$

where q is a set of DoFs, including a unique set of matching DoFs and non-interface DoFs. Since (6) states that the complete set of DoFs u of uncoupled subsystems is obtained from the unique set of DoFs q, the compatibility condition holds for any set q, i.e.

$$\mathrm{B}u = \mathrm{BL}q = 0 \qquad \forall q \tag{7}$$

Therefore, L is the null space of B:

$$\mathrm{L} = \mathrm{null(B)} \tag{8}$$

The choice of a unique set of DoFs q satisfies the compatibility condition in (5) and the system of equations becomes:

$$\begin{cases} \mathrm{M\,L}\,\ddot{q} + \mathrm{C\,L}\,\dot{q} + \mathrm{K\,L}\,q = f + g \\ \mathrm{L}^T g = 0 \end{cases} \tag{9}$$

The equation of motion of the assembled system is finally obtained by pre-multiplying the first equation by L^T and noting that $\mathrm{L}^T g = 0$:

$$\mathrm{L}^T\,\mathrm{M\,L}\,\ddot{q} + \mathrm{L}^T\,\mathrm{C\,L}\,\dot{q} + \mathrm{L}^T\,\mathrm{K\,L}\,q = \mathrm{L}^T\,f \tag{10}$$

i.e.

$$\widetilde{\mathrm{M}}\,\ddot{q} + \tilde{\mathrm{C}}\,\dot{q} + \widetilde{\mathrm{K}}\,q = \tilde{f} \tag{11}$$

with an obvious meaning of symbols.

The equation of motion of the coupled system can be expressed in the frequency domain by letting $\tilde{f}(t) = \tilde{F}(\omega)e^{j\omega t}$ and $q(t) = Q(\omega)e^{j\omega t}$:

$$\left[-\omega^2\widetilde{\mathrm{M}} + j\omega\tilde{\mathrm{C}} + \widetilde{\mathrm{K}}\right]Q(\omega) = \tilde{F}(\omega) \tag{12}$$

hence:

$$Q(\omega) = \left[-\omega^2\tilde{\mathrm{M}} + j\omega\tilde{\mathrm{C}} + \tilde{\mathrm{K}}\right]^{-1}\tilde{F}(\omega) = \mathrm{H}(j\omega)\,\tilde{F}(\omega) \tag{13}$$

Where $\mathrm{H}(j\omega)$ is the receptance matrix of the assembled system.

2.2. Modal Reduction, the Craig-Bampton Method

The modal reduction represent a very useful approach especially in combination with substructuring. In fact, the number of DoFs of each substructure (r) can be significantly reduced by retaining only a subset of the physical DoFs that includes the connecting DoFs of substructure (r) with other substructures.

Therefore, the full set of DoFs $u^{(r)}$of a substructure (r) can be partitioned as:

$$u^{(r)} = \begin{Bmatrix} u_b \\ u_i \end{Bmatrix} \tag{14}$$

i.e. boundary DoFs (subscript b) and interior DoFs (subscript i), where the superscript (r) is dropped for the sake of simplicity. Boundary degrees of freedom are physical DoFs that are retained in the reduction process and must include the connecting DoFs of the r-th substructure to the other substructures. According to[7], the displacements field $u^{(r)}$ can be expressed as a linear combination of static deformation modes Φ^c and a truncated set of m ($m \ll i$) fixed interface modes Φ:

$$\Phi^c = \begin{bmatrix} \mathrm{I}_b \\ \Phi_i^c \end{bmatrix} \qquad \Phi = \begin{bmatrix} 0_b \\ \Phi_i \end{bmatrix} \tag{15}$$

Therefore,

$$u^{(r)} = \begin{Bmatrix} u_b \\ u_i \end{Bmatrix} = \begin{bmatrix} \mathrm{I}_b & 0_b \\ \Phi_i^c & \Phi_i \end{bmatrix} \begin{Bmatrix} u_b \\ p_m \end{Bmatrix} = \Gamma^{(r)} p^{(r)} \tag{16}$$

where p_m is the truncated set of modal coordinates, $\Gamma^{(r)}$ is the transformation matrix and $p^{(r)}$ is the reduced set of generalized coordinates.

Finally, the equation of motion of the substructure (r) can be expressed using the reduced set of coordinate $p^{(r)}$ by substituting (16) in (1) and pre-multiplying all the terms by the transposed transformation matrix ${\Gamma^{(r)}}^T$, thus obtaining:

$$\hat{\mathrm{M}}^{(r)}\ddot{p}^{(r)} + \hat{\mathrm{C}}^{(r)}\dot{p}^{(r)} + \hat{\mathrm{K}}^{(r)}p^{(r)} = \hat{f}^{(r)} + \hat{g}^{(r)} \tag{17}$$

Reduced order models of substructures can be used in substructuring and the solution expressed in the generalized coordinates $p^{(r)}$ can be expanded to the full set of physical DoFs $u^{(r)}$ by means of the transformation in (16).

2.3. Displacement and Force Transmissibility

The transmissibility of a vibrating system is the nondimensional ratio between forces or displacements at different locations . It is a very useful concept since it allows to understand how a vibrating system (e.g. a suspension) amplifies or attenuates the vibrations.

Although the transmissibility is quite simple to define for a single DoF suspension system, in which the ratio between forces and displacements leads to the same formulation, the definition of the transmissibility for a multi DoFs system is not trivial and in general the ratio between forces or displacements does not define the same quantity. In this paper, dealing with a multi DoF system, the displacement transmissibility $T_{ij}^d = x_i/x_j$ (with $F_i = 0$, no constraint acting on DoF i) and the force transmissibility $T_{ij}^F = F_i/F_j$ (with $x_i = 0$, only the DoF i is blocked and F_i represents the generated constraint reaction) are used.

3. SUBSTRUCTURE MODELS

The system considered in this paper is composed by an agricultural tractor and an operating machine connected together through a linkage system. Therefore, the agricultural tractor, the linkage system (three points linkage or hitch) and the machinery are considered as substructures to be coupled using substructuring and in this section their models are presented.

3.1. Lumped Parameter Model of the Agricultural Tractor

The tractor is modeled as a lumped parameter system in the longitudinal plane as shown in Figure 1. The two masses m_1 and m_3 represent respectively the seat with the operator and the front axle, and the rigid body, characterized by a mass m_2 and a moment of inertia I_2, represents the tractor frame. The mass m_1 (operator seat) is connected to the rigid body by a spring with stiffness k_1 and a damper with a damping coefficient c_1.

Since agricultural tractor are generally not provided of rear suspension, the spring with stiffness k_2 and the damper with a damping coefficient c_2, representing the rear wheels, are directly connected to the rigid body. The mass m_3 (front axle) is connected to the rigid body by a suspension with stiffness k_3 and damping coefficient c_3 and the front wheels are characterized by a stiffness k_4 and damping coefficient c_4.

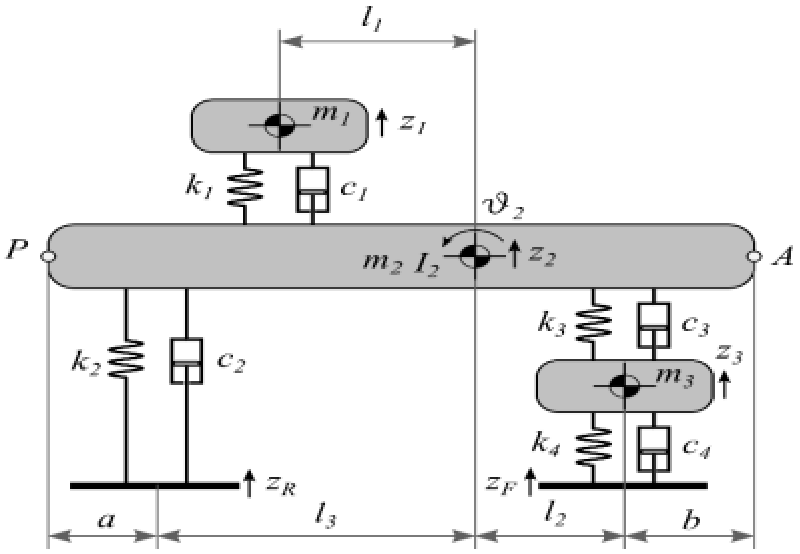

Figure 1 Lumped parameter model of the agricultural tractor

Therefore, the tractor model has 4 DoFs, i.e. the vertical displacements z_1 and z_3 of masses m_1 and m_3, the vertical displacement z_2 and the in plane rotation ϑ_2 of the rigid body. Moreover, in order to account for the unevenness of the soil, the vertical displacement z_F and z_R of the contact points of the front and rear wheels with the ground are considered. The connection points with the rear and front linkage systems are indicated with A and P respectively.

3.2. Reduced Order Model of the Rear Mounted Machine

Rear mounted machinery are generally connected to the tractor through the rear three-point linkage. The rear mounted operating machine and the rear three point linkage are considered as a single substructure. The model of the rear three point linkage considered in this paper is shown in the left part of Figure 2 and it is composed by the two lower links (components 1), the upper link (component 2), the lift rods (components 3) and the two struts (components 5). The linkage system connects the implement (component 4) to the tractor frame in the points P_1 to P_7.

The linkage is modeled by finite element and a rigid body is considered to account for inertia properties of the rear mounted machine. The resulting model is subsequently reduced using the Craig-Bampton modal reduction for the sake of dynamic substructuring. The selected boundary nodes are shown in the right part of Figure 2 where P represents the point to be used in the coupling with the tractor model, E and O represent respectively the connecting point and the CoG of the rear mounted machine.

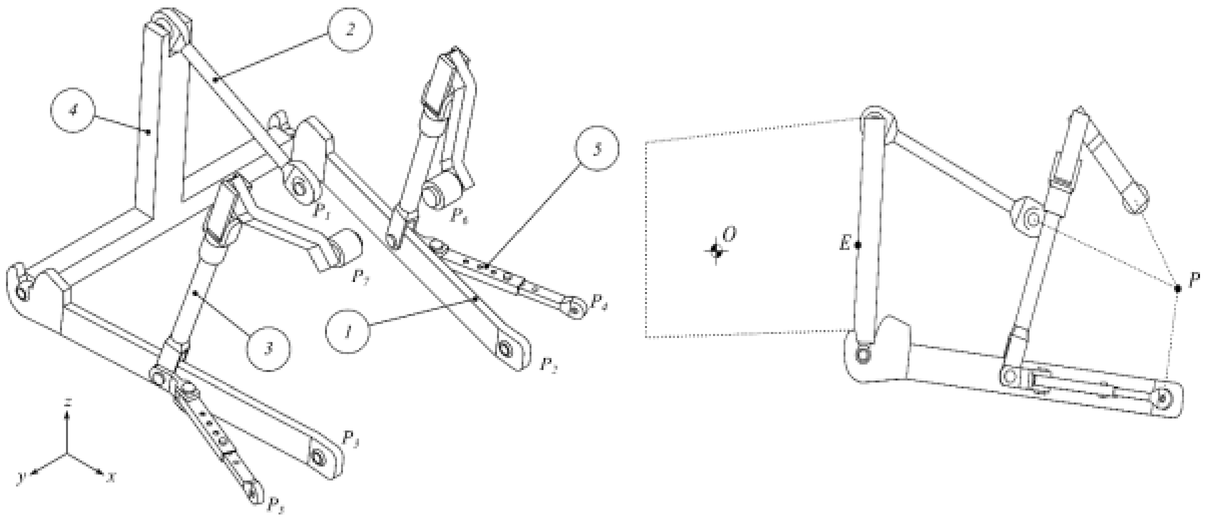

Figure 2 Rear three-point linkage and substructure model of the rear mounted machine.

3.3. Reduced Order Model of the Front Mounted Machine

Front mounted machinery are generally connected to the tractor through the front three point linkage. The front mounted operating machine and the front three-point linkage are considered as a single substructure. The model of the front three point linkage, considered in this paper, is shown in the left part of Figure 3 and similarly to the rear linkage it is composed by the two lower links (components 1), the upper link (component 2), the lift cylinders (components 3). The linkage system connects the implement (component 4) to the tractor frame.

Also in this case, the linkage is modeled by finite element and a rigid body is considered to account for inertia properties of the front mounted machine. The resulting model is subsequently reduced using the Craig Bampton modal reduction and the selected boundary nodes are shown in the right part of Figure 3, where A represents the point to be used in the coupling to the tractor, C and Q represent respectively the connecting point and the CoG of the front mounted machine.

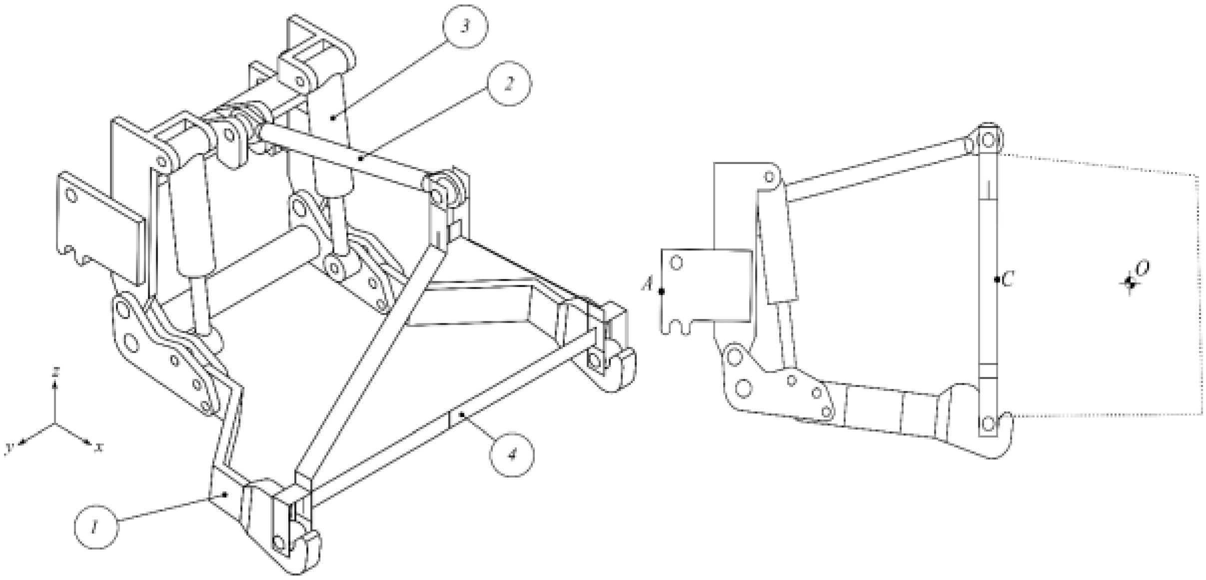

Figure 3 Front three-point linkage and substructure model of the front mounted machine.

3.4. Reduced Order Model of the Trailed Machine

Trailed machinery are generally connected to the tractor through a hitch. In this case the trailed machine and the towing hitch are considered as a single substructure. In the left part of Figure 4, the model of the trailed machine frame and of the coupling device used in the attachment of non-balanced trailers to agricultural towing vehicles, are shown.

The frame and the hitch are modeled by finite element and subsequently reduced using the Craig-Bampton modal reduction. The selected boundary nodes are shown in the right part of Figure 4 where P is the connecting point to the tractor frame, D is the trace of the axle of the towed machinery. Moreover, the wheels of the trailer are modeled by adding to the reduced FE model a spring with stiffness k_5 and a damper with damping constant c_5, connecting the point D with the ground. In order to account for the unevenness of the soil, the vertical displacement z_T of the contact point between the wheels and the ground is considered.

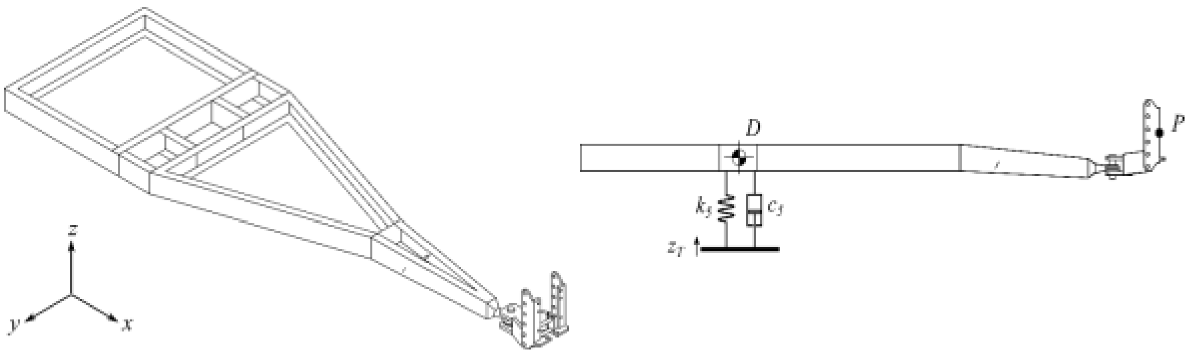

Figure 4 Hitch connection with a trailed machine frame and substructure of the trailed machine.

4. RESULTS

The present work aims to investigate the effects of operating machines on the vibration transmitted from the agricultural machine to the operator seat. In this section, the substructure models are used to investigate the transmissibility of the component substructures at first and subsequently they are coupled, using the substructuring approach, to investigate the transmission of vibrations from the operating machines to the operator seat.

The tractor model shown in Figure 1 is characterized by an operator mass, m_1= 88 kg, a frame mass and moment of inertia, m_2=7655 kg and I_2=15938 kg/m^3 respectively, a front axle mass m_3=900 kg; the stiffnesses of the connecting springs among the lumped masses are k_1=16 kN/m, k_2=894 kN/m, k_3=640 kN/m and k_4=748 kN/m. The horizontal distances between the CoG of the tractor frame and the operator seat is l_1=0.28 m, the distances between the two axle and the CoG of the tractor frame are l_2=1.68 m and l_3=1.16 m, the horizontal distance between the connecting points A and P, and front and rear axle is $a = b =$ 0.25 m, Moreover, for the trailed machine model shown in Figure 4, the stiffness and damping coefficient of the wheels are k_5 = 2e5 N/m and c_5 = 2347 Ns/m respectively.

4.1. Transmissibility Analysis of the Agricultural Tractor

The displacement transmissibilities from the wheels to the tractor frame and to the seat are shown in Figure 5. They give a clear and comprehensive information about the dynamic behavior of the agricultural tractor when travelling over uneven ground surfaces. The transmissibility results show a significative amplification of the displacement of the two wheels for some frequency range. In particular, at the frequency of about 2 Hz, the amplification on the seat of the rear wheel displacement is higher than that of the front wheel. This is probably related to the presence of the suspension on the front axle. Conversely, at around 1.5 Hz the front axle suspension seems to increase the transmissibility.

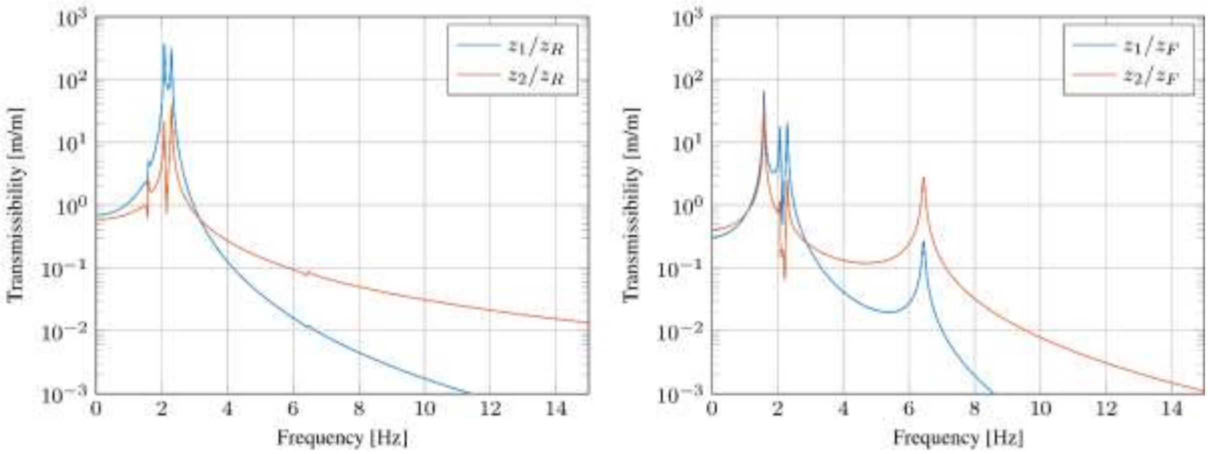

Figure 5 Displacement transmissibilities between the tractor's wheels .and the CoG of the tractor frame (z_2) or the operator's seat (z_1).

4.2. Transmissibility of the Linkage Systems

The force transmissibilities of the linkage systems provide information about the transmission to the tractor of the operating machine vibrations. For the front and rear three points linkages, it is defined between the mounted machine CoG and the connection points to the tractor frame, while for the trailed machine the transmissibility is defined between the point D and the connection point to the tractor frame.

Figure 6 shows the force transmissibilities between a force acting on O along either horizontal or vertical direction and the linking points P_i of the linkage system to the tractor (see Figure 2). Only the transmissibilities to the points P_1, P_2, P_4 and P_6 are represented because of the symmetry of the system with respect to the longitudinal plane. The force transmissibilities highlight that amplification occurs for both horizontal and vertical forces at frequencies higher than 30 Hz. However, in the range of interest for the operator safety (i.e. lower than 10 Hz) only the forces acting in vertical direction are amplified.

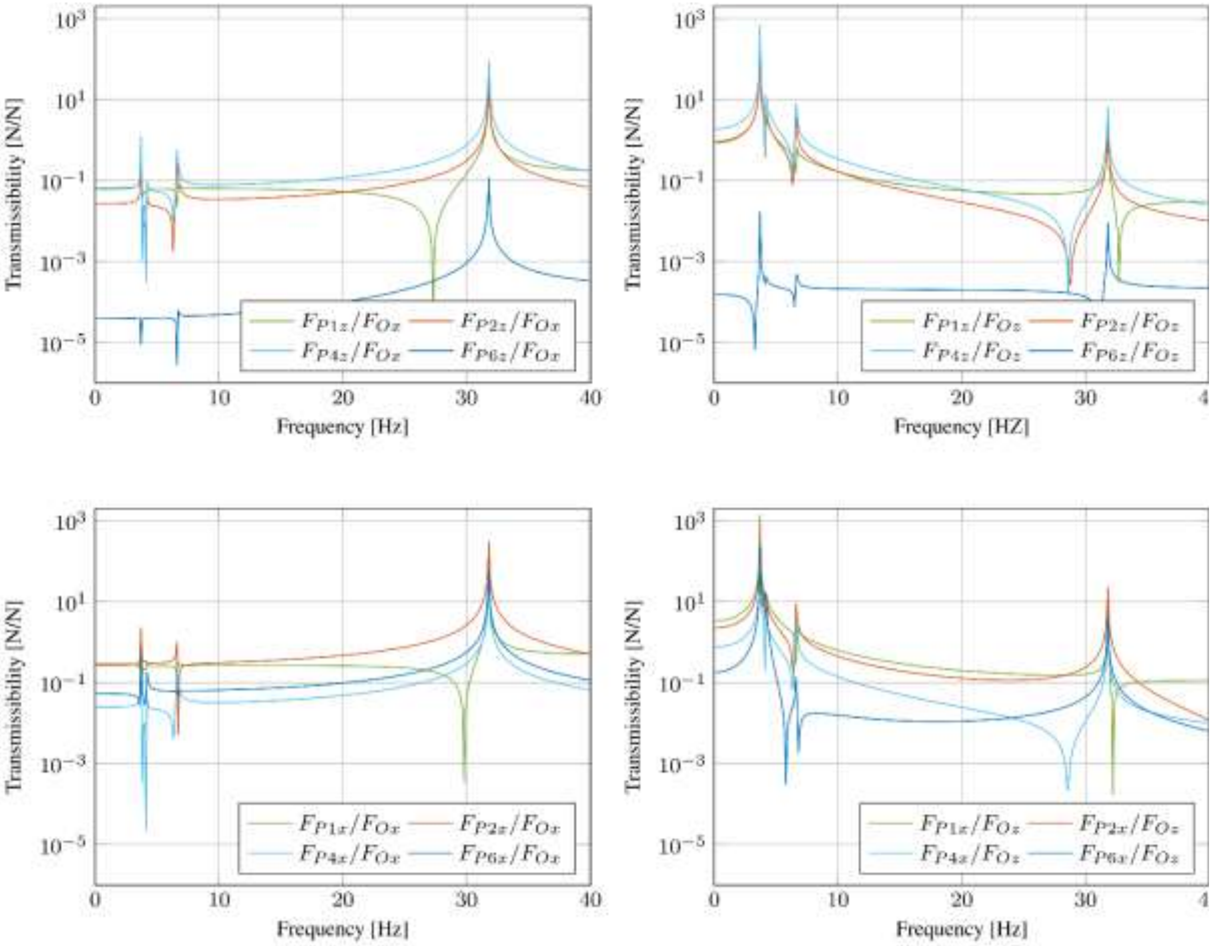

Figure 6 Force transmissibilities between the CoG of the mounted machine O and the linking points P_i of the linkage.

Figure 7 shows the force transmissibilities between a force acting on point C along either horizontal or vertical direction, and the connecting point A of the front three point linkage system to the tractor frame. The force transmissibilities show that forces acting on C along the vertical direction are amplified at the connection point with the tractor A in both horizontal and vertical directions, while forces acting on C along the horizontal direction are amplified in A only in the vertical direction at the connection point with the tractor.

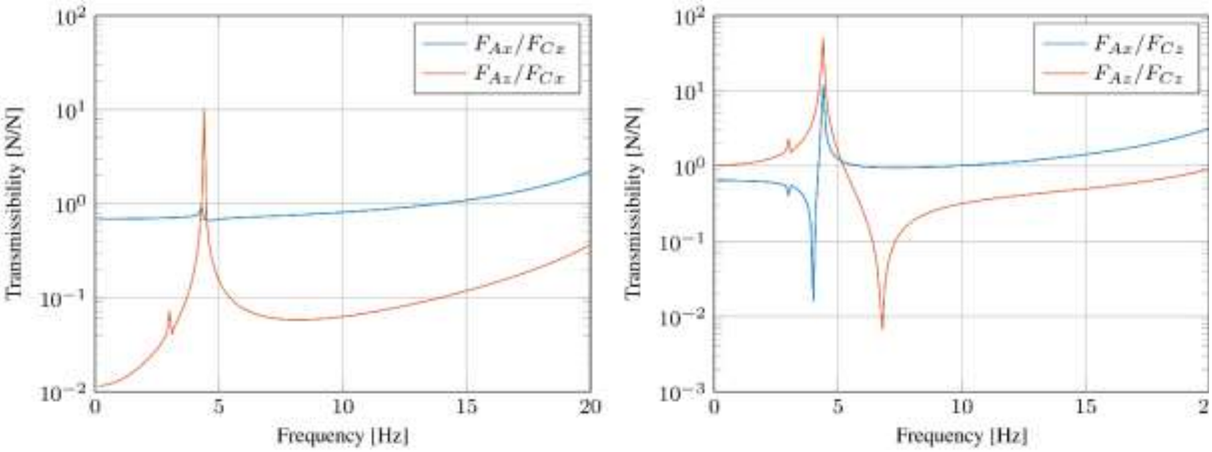

Figure 7 Force transmissibilities between the point C on the implement of the front mounted machine and the point A of the linkage.

Figure 8 shows the force transmissibilities between a force acting on point D, along either horizontal or vertical direction, and the connecting point P of the hitch to the tractor frame. The force transmissibilities show that vertical forces are amplified at the connection point with the tractor in both horizontal and vertical directions, while the horizontal forces are generally attenuated in the vertical direction but they are transmitted without attenuation in the horizontal direction, with a relevant amplification above 15 Hz.

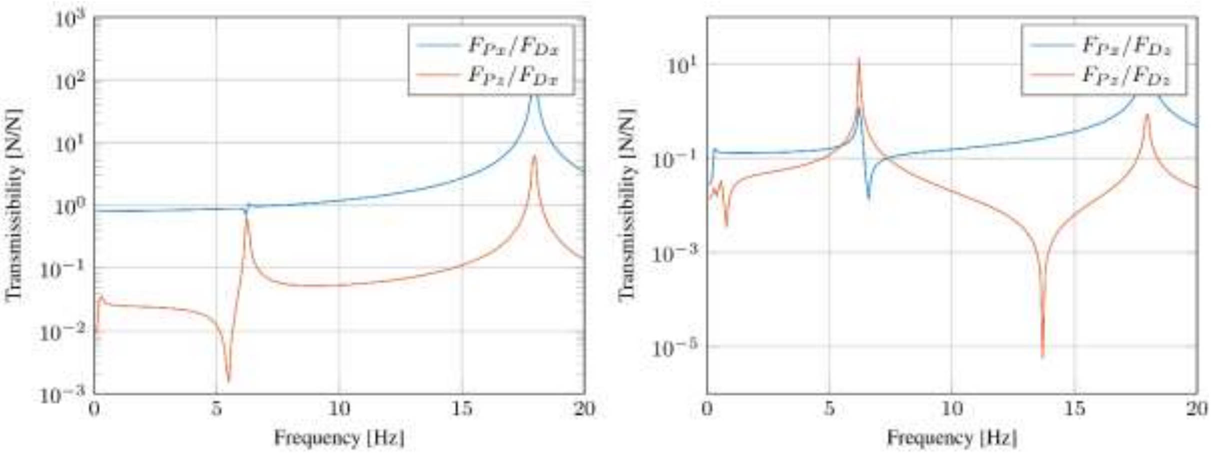

Figure 8 Force transmissibilities between the point D of the trailed machine and the point P.

4.3. Substructuring Analysis of the Tractor and Machinery Set

The reduced order models of the front and rear three points linkages, obtained using 10 fixed interface modes, are here coupled to the lumped model of the tractor in order to evaluate the transmission of vibrations from the operating machine to the operator seat.

Figure 9 shows the frequency response functions of the system obtained by coupling the tractor model with the rear mounted machine trough the three points linkage. The inertance (acceleration/force) between either the vertical or rotational DoFs of the point O on the machinery and the vertical DoFs z_1 or z_2 on the tractor are shown. The results highlight the presence of 6 modes below 15 Hz: some of these resonances were observed in the transmissibilities of the component substructures.

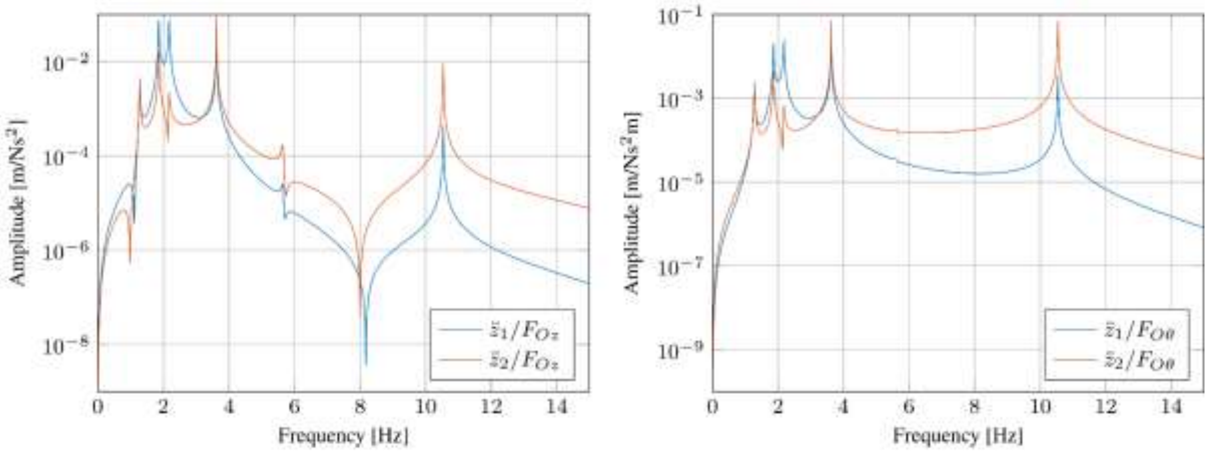

Figure 9 FRF of the coupled system composed by the agricultural tractor with a rear mounted machine connected through a three points linkage.

Figure 10 shows the frequency response functions of the system obtained by coupling the tractor model with the front mounted machine through the three points linkage. The inertance (acceleration/force) between either the vertical or rotational DoFs of the point Q on the machine and vertical DoFs z_1 or z_2 on the tractor are shown. The results highlight the presence of 6 modes below 15 Hz: some of these resonances were observed in the transmissibilities of the component substructures.

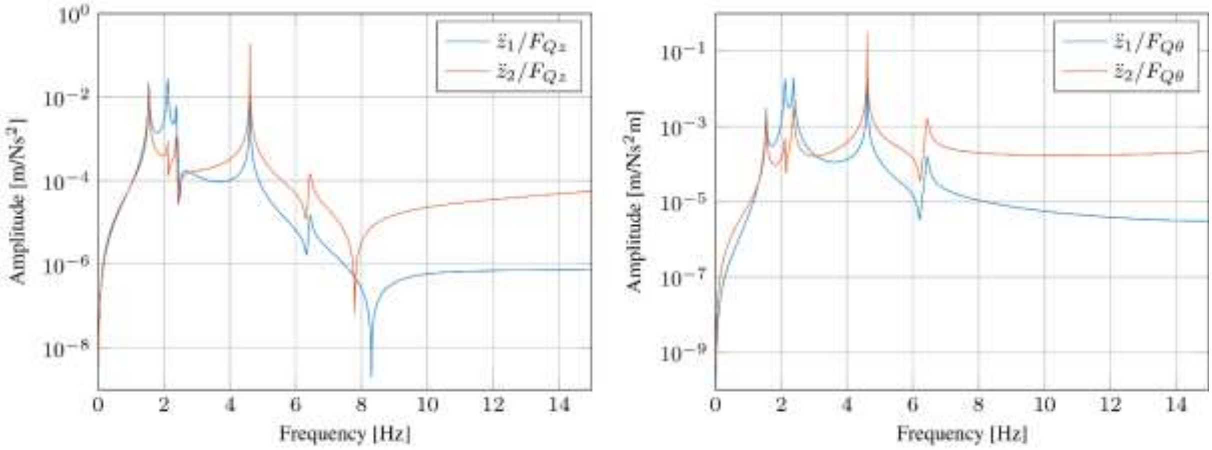

Figure 10 FRF of the coupled system composed by the agricultural tractor with a mounted machine on the front three points linkage.

5. CONCLUSIONS

In this paper, the vibration transmission from the agricultural machinery to the operator seat is investigated. In particular, in order to evaluate the effects of the soil unevenness and of the implement, the complex mechanical system, composed by the tractor and the connected operating machine is considered. Models of the tractor, of the front and rear mounted machine including the three point linkage and of the trailed machine including the hitch are provided. The analysis of the transmissibility of the linkage systems highlight how the dynamics of the linkage affects the transmission from the operating machine to the tractor. Substructuring approach in frequency domain is used to couple the model of the tractor with the reduced order model of the operating machine, including the implement and the linkage system. The coupled system is used to investigate the vertical response of the operator seat to the excitation provided by the operating machine. Results highlight the influence of the linkage dynamics in the vibrations transmitted by the operating machine to the operator seat.

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