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ENVIRONMENTAL IMPACT OF AGRICULTURAL AGGREGATES

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Abstract: *Having established a theoretical method for the study of the particularities of transmissions, the paper brings a contribution to the better knowledge of concrete aspects of tractor and agricultural implements operation, and allows for a rational selection of the computation regime of the power drive transmission, as well as of the optimum manufacturing and operation parameters and environmental impact. Following these research significant similarities emerged of the theoretical and experimental results, thus proving the adequacy of the theoretical research method.*

Key words: *agricultural aggregates,* dynamics*, environment.*

1. Introduction

The general trend in the manufacturing of both tractors and agricultural machines is the permanent stimulation of productivity growth of the tractor-implement aggregate. The solutions prevailing in current tractor manufacturing are the 4 wheel drive and machines coupled at the front, thus improving the use of engine power and aggregate dynamics.

The recent intensification of agriculture, and the prospects of future intensification, will have major detrimental impacts on the nonagricultural terrestrial and aquatic ecosystems of the world. Based on a simple linear extension of past trends, the anticipated next doubling of global food production would be associated with approximately 3-fold increases in nitrogen and phosphorus fertilization rates, a doubling of the irrigated land area, and an 18% increase in cropland.

These projected changes would have dramatic impacts on the diversity, composition, and functioning of the remaining natural ecosystems of the world, and on their ability to provide society with a variety of essential ecosystem services. Because of aerial redistribution of various forms of nitrogen, agricultural intensification also would wither many natural terrestrial ecosystems and contributes to atmospheric accumulation of greenhouse gases.

These detrimental environmental impacts of agriculture can be minimized only if there is much more efficient use and recycling of nitrogen and phosphorus in agro-ecosystems.

The agricultural achievements of the past 35 years have been impressive. Grain production, mainly from wheat, rice, and maize, has increased at a rate greater than human population. This has decreased the number of malnourished people even as the earth's human population doubled to

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5.8 billion. Although the estimates vary widely, world population is projected to increase about 75% before leveling off at about 10 billion. In combination with increasing demand for meat in developing countries and the use of grains as livestock feed, this increased population density should cause world demand for grain production to more than double. A new long-term, multidisciplinary research program is needed to develop agricultural methods that can feed a growing world and still preserve the vital services provided to humanity by the world's natural ecosystems.

Current agricultural practices involve deliberately maintaining ecosystems in a highly simplified, disturbed, and nutrient-rich state. To maximize crop yields, crop plant varieties are carefully selected to match local growing conditions. Limiting factors, especially water, mineral nitrogen, and mineral phosphate, are supplied in excess, and pests are actively controlled. These three features of modern agriculture - control of crops and their genetics, of soil fertility via chemical fertilization and irrigation, and of pests (weeds, insects, and pathogens) via chemical pesticides - are the hallmarks of the green revolution.

The quality of agricultural machines and technologies has also a considerable impact on soil evolution, and can improve the further crop production [1].

2. Dynamic Model for the Transmission

For frontal attached implements the use of 4WD tractors is recommended, for the following reasons:

a) their whole operational weight is adherent, thus inducing a growth of the traction force and a more complete use of engine power;

b) smaller fuel consumption in relation to traction power;

c) improved traction efficiency;

d) better visibility of the worked area [3].

In the case of classical aggregates (4WD tractor and implement at the rear) the load on the front axle is reduced (particularly for carried implements), what leads to increased slipping, reduced stability and manipulation capacity of the tractor. In order to compensate this phenomenon, the front axle is typically counterbalanced with additional masses, but thus the resistance to motion of the tractor increases and implicitly traction force diminishes.

Front coupling of agricultural machines on the tractor can remove these disadvantages. Front attachment of implements is also justified by aspects related to the working technique.

Machines yielding good results when coupled at the front of the tractor are: fertilizer scattering machines, sprinkling machines, cultivators, agricultural cutters, harrows, sugar beet root neck strippers, mowers, various equipment for harvesting (by cutting or tearing), ploughs etc.

The symbols used by the dynamic model for the transmission of the aggregate consisting of a U833DT 4WD tractor with frontal carried implements driven by the front power drive, are:

J_i - inertia moments reduced to the clutch shaft (1 - engine; 2 - driven part of the clutch; 3 - rear axle transmission; 4 - front axle transmission; 5 - tractor mass; 6 - transmission of the front power drive; 7 - agricultural machine);

$k_{i,i+1}$ - reduced elastic constant, corresponding to the transmission elastic elements (global);

φ_I - angular displacement of the rotating part;

M_m - driving torque applied to mass J_1;

M_a - friction moment of the main clutch;

$M_{\varphi s}$ - torque transmitted by the rear driven wheels by adherence to the soil;

$M_{\varphi f}$ - torque transmitted by the front driven wheels, by adherence to the soil;

M_{fs} - torque resistant to the motion of the aggregate with carried machines (rear axle);

M_{ff} - torque resistant to the motion of the aggregate with carried machines (front axle);
M_r - torque resistant to the motion of carried agricultural machines;
M_R - (global) torque resistant to motion;
M_{rf} - torque resistant to the advancing in the field of the frontal carried implement [2].

The mathematical relations were written for both transitory and stable operation. For the first stage (transitory operation) the system of differential equations describing the behaviour of the dynamic model is the following [4]:

$$\begin{aligned}
&J_1 \cdot \varphi''_1 = M_n - M_a(t) - k_{16}(\varphi_1 - \varphi_6) - c_{16}(\omega_1 - \omega_6),\\
&J_2 \cdot \varphi''_2 = M_a(t) - k_{23}(\varphi_2 - \varphi_3) - c_{23}(\omega_2 - \omega_3) - k_{24}(\varphi_2 - \varphi_4) - c_{24}(\omega_2 - \omega_4),\\
&J_3 \cdot \varphi''_3 = k_{23}(\varphi_2 - \varphi_3) + c_{23}(\omega_2 - \omega_3) - k_{35}(\varphi_3 - \varphi_5) - c_{35}(\omega_3 - \omega_5),\\
&J_4 \cdot \varphi''_4 = k_{24}(\varphi_2 - \varphi_4) + c_{24}(\omega_2 - \omega_4) - k_{45}(\varphi_4 - \varphi_5) - c_{45}(\omega_4 - \omega_5),\\
&J_5 \cdot \varphi''_5 = M\phi_1 + M\phi_2 - M_r - M_R,\\
&J_6 \cdot \varphi''_6 = k_{16}(\varphi_1 - \varphi_6) + c_{16}(\omega_1 - \omega_6) - k_{67}(\varphi_6 - \varphi_7) - c_{67}(\omega_6 - \omega_7),\\
&J_7 \cdot \varphi''_7 = k_{67}(\varphi_6 - \varphi_7) + c_{67}(\omega_6 - \omega_7) - M_r c,
\end{aligned} \tag{1}$$

where:
φ_I - is the angular displacement of element i [rad];
ω_I - angular velocity of element i [rad];
φ_I'' - angular acceleration of element i [rad].

For the second stage (stable operation) the mathematical relations describing the behaviour of the dynamic model are [4]:

$$\begin{aligned}
&(J_1 + J_2) \cdot \varphi''_1 = M_m - k_{23}(\varphi_2 - \varphi_3) - c_{23}(\omega_2 - \omega_3) - k_{24}(\varphi_2 - \varphi_4) - c_{24}(\omega_2 - \omega_4),\\
&J_3 \cdot \varphi''_3 = k_{23}(\varphi_2 - \varphi_3) + c_{23}(\omega_2 - \omega_3) - k_{35}(\varphi_3 - \varphi_5) - c_{35}(\omega_3 - \omega_5),\\
&J_4 \cdot \varphi''_4 = k_{24}(\varphi_2 - \varphi_4) + c_{24}(\omega_2 - \omega_4 - k_{45}(\varphi_4 - \varphi_5) - c_{45}(\omega_4 - \omega_5),\\
&J_5 \cdot \varphi''_5 = M\phi_1 + M\phi_2 - M_r - M_7,\\
&J_6 \cdot \varphi''_6 = M_m(t) - k_{16}(\varphi_1 - \varphi_6) - c_{16}(\omega_1 - \omega_6),\\
&J_7 \cdot \varphi''_7 = k_{16}(\varphi_1 - \varphi_6) + c_{16}(\omega_1 - \omega_6) - k_{67}(\varphi_6 - \varphi_7) - c_{67}(\omega_6 - \omega_7).
\end{aligned} \tag{2}$$

Solving of these equations implies writing linking equations and boundary conditions that determine the behavior of the equivalent model in a manner quite close to the real system. These equations are [4]:

$$\begin{aligned}
&M_a = M_a(t),\\
&M\phi_1 = M\phi_1(k_{35}, \varphi_3, \phi_5, \omega_3, \omega_5, M\phi M_1),\\
&M\phi_2 = M\phi_2(k_{45}, \varphi_4, \varphi_5, \omega_4, \omega_5, M\phi M_2),\\
&\phi_{m1} = \phi_{m1}(M\phi_1, M\phi M_1),\\
&\phi_{m2} = \phi_{m2}(N\phi_2, M\phi M_2),\\
&\delta_1 = \delta_1(M\phi_1, M\phi M_1, \delta_{1\max}),\\
&\delta_2 = \delta_2(M\phi_2, M\phi M_2, \delta_{2\max}),\\
&\eta_{\delta 1} = \eta_{\delta 1}(\delta_1),\\
&\eta_{\delta 2} = \eta_{\delta 2}(\delta_2).
\end{aligned} \tag{3}$$

3. Computer-Aided Data Processing

The integration of the system time is considered an independent variable, while the angular displacements φ_I and angular velocities ω_I of the inertial masses J_i are considered dependent variables. The computer aided solving of the system of differential equations written in the previous subchapters is achieved in MathCAD 6.0 PLUS, using the Runge-Kutta general integration method.

The time-related variation of the various computed quantities was represented graphically. Figure 1 shows the variation of the angular velocities for the main components of the transmission of a U 833 DT tractor equipped with a front mower driven by the front power drive.

The plotted quantities are the following:

y_1 - is the angular velocity of the driving shaft [rad/s];

y_2 - the angular velocity of the clutch driven shaft [rad/s];

y_3 - angular velocity of the rear driven wheels [rad/s];

y_4 - angular velocity of the front driven wheels [rad/s];

y_6 - angular velocity of the front power drive shaft [rad/s];

$f(z)$ - speed impulses of the front power drive shaft;

$f1(z1)$ - speed impulses of the rear wheel axle;

x - time [s].

Figure 2 presents the moments (torques) transmitted by the shafts.

The plotted quantities are the following [3]:

M_m - is the driving torques [Nm];

M_a - torque transmitted by the clutch [Nm];

M_{1e} - torque transmitted by the axles of the rear driving wheels [Nm];

M_{3e} - torque transmitted by the axles of the front driving wheels [Nm];

M_{5e} - torque transmitted by the shaft of the front power drive [Nm];

M_rF - torque resistant to mowing [Nm] [1], [4]:

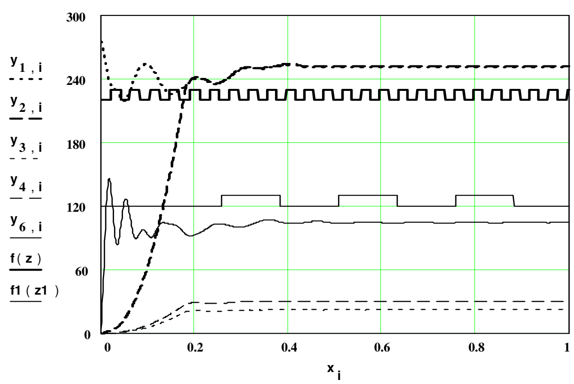

Fig. 1. *Variation of the angular velocities for the main components of the transmission*

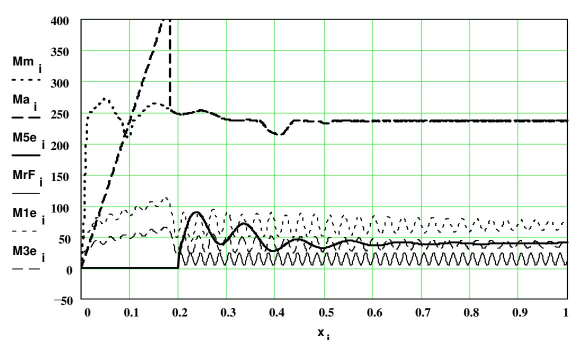

Fig. 2. *Variation of the moments (torques) transmitted by the shafts*

This torques are reduced to the clutch shaft and represented proportionally.

The front power drive clutch was coupled during halting, where after the aggregate was started. From this moment the evolution of the parameters was monitored graphically. It can be noticed, that the variation of the torques transmitted by the driving shaft [Nm] and the tractor clutch shaft (M_a) takes place between optimum limits. The field is entered following total coupling of the clutch, when a variable resistant force occurs, expressed by the resistant moment M_rF. The evolution of the torque transmitted by the front power drive shaft (M_{5e}) is adequate and stabilizes soon after engaging into load.

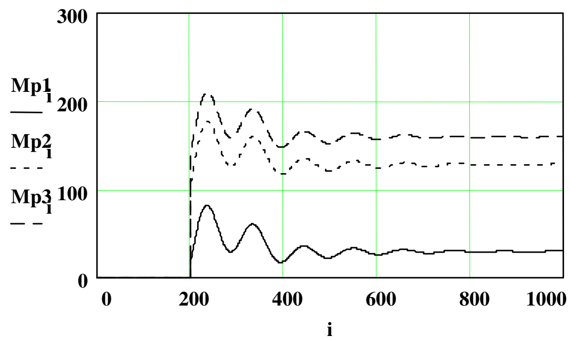

Fig. 3. *Variation of the torques transmitted by the front power drive shaft in the case of driving, only at the front, of various types of agricultural machines*

Figure 3 shows the variation graphs of the torques transmitted by the front power drive shaft in the case of driving, only at the front, of various types of agricultural machines: front mower with an alternating straight motion cutting device (M_{p1}), rotating front mower (M_{p2}) and agricultural cutter (M_{p3}).

When the work is with a carried mower driven by the front power drive, endowed with an alternating straight motion cutting device, with a cutting and counter-cutting blade, the reduced resistant moment is not great and can reach values of 4...10 Nm. The force resistant to advancing into the field was considered variable according to a sinus curve. The greatest slipping occurs in the real axle ($\delta_{f1} = 0.023$), corresponding to a cinematic non-concordance coefficient of $k = 0.994$, considered better than just acceptable.

Loading the power drive with an increased reduced moment (20-25 Nm), (front power drive of a rotating mower with four working sections, with cutting blades on drums), the front axle will be additionally stressed, slipping will increase, reaching a sensibly equal value to that of the rear axle ($k = 1.003$).

The study of the transmission with an even greater load of the power drive was achieved by simulation of the operation of the aggregate consisting of a tractor equipped with a front cutter and driven by the power drive. For considerably greater reduced moments (50-60 Nm) the slipping has increased to values like: $\delta_{s3} = 0.083$ and $\delta_{f3} = 0.0962$ that corresponds to a cinematic non-concordance coefficient of $k = 1.015$.

Under the same working circumstances the behavior of the power drive and the aggregate was studied, during use of both power drives, by driving front and rear carried implements, the employed notations being similar to the ones of the previous case. Following these research significant similarities emerged of the theoretical and experimental results, proving on one hand the adequacy of the theoretical research method.

4. Conclusions

The theoretical analysis and experimental study allow for conclusions useful for the consequent research of the dynamic stresses in the power drive and aggregate transmissions, the most important ones being presented further on.

1. The theoretical study of the stresses on the aggregate power drive and transmission calls for an extension of the concept of transmission, including the assembly of all parts linking the tractor engine to the active parts of the implement, that transmit working resistances. The transmission represents a dissipative vibrating system, consisting of inertia masses linked between themselves by elastic parts and friction couplings. Variable disturbing factors act upon this system, inducing in the elements of the transmission torsion vibrations and considerable dynamic stress will occur.

2. Based on the set up dynamic models, considering the working particularities of the studied aggregates and simulating by adequate relations the disturbing factors of the real system, the mathematical models of the working process can be established.

3. In all analysed cases the fact was highlighted, that using the power drive in mobile aggregates induces variable stresses not only in the power drive transmission, but also in the transmission to the driving wheels. For this reason the dynamics of the entire aggregate has to be studied.

4. Using machines attached only frontal is advantageous for small power consumption's, when the resistance to advancing does not exceed 4.8 kN.

5. For operations loading the power drives with large resistant moments, working with aggregates with implements attached both at the front and the rear is recommended, allowing for an improved dynamics of the aggregate, as well as for reducing the amplitude of the variable stresses.

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