6th International Conference
ð Computational Mechanics and Virtual Engineeringð
COMEC 2015
15-16 October 2015, Braşov, Romania

MATHEMATICAL AND DYNAMIC MODELING OF THE COUPLINGS FROM SYSTEM TRACTOR-AGRICULTURAL MACHINE OPERATED BY THE PTO

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Abstract: *The study stresses from transmission mechanism of tractors and agricultural machinery driven trough PTO with sufficient accuracy is achieved by using dynamic and mathematical modeling method thereof. The results are more accurate, how much more are envisaged constructive and functional features the of transmission mechanism assembly tractor - agricultural machine. The paper studies the dynamic and mathematical modeling particularities of the couplings of the transmission mechanism independent PTO, used to drive a significant number of mechanisms and working systems to agricultural machinery. They have analyzed this way PTO own coupling, a flexible intermediate coupling and a safety coupling.*
Keywords*: connecting couplings, PTO tractor, dynamics and mathematical modeling*

1. INTRODUCTION

The operation of farm machinery trough PTO requires the satisfaction of certain conditions relating to the correlation between starting and stopping working bodies of their with starting and stopping the system tractor - agricultural machine as a whole [4]. Dynamic loads that occur in these situations in independent PTO transmission mechanism is manifested in all tractor and agricultural machine transmission, so it is necessary to use couplings that can reduce these requests. Whatever their nature, these couplings are important elements influencing essential dynamic load regime. Currently, agricultural tractors and machinery have become extremely advanced technical equipment in the sense that it possesses capabilities and high reliability. Meanwhile, the prices of machinery rose as much. For these reasons, designers and manufacturers of tractors and agricultural machines using the most efficient methods and solutions research, that which is done in this paper[5].

2. MATERIAL AND METHOD

2.1.Modeling own of the PTO clutch

Characteristic for the construction of independent PTO transmission mechanism is the presence an intermittent coupling with external control, which ensures its autonomy movement shaft, by wich to transmit the movement and the drive torqe by working equipment of agricultural machine. Typically, this coupling takes the form of a friction clutch, usually multi-disk, and optionally coupled or one step planetary gear [2]. In the second case, the gear body is attached to the rear of the transmission housing. The housing *2* (Figure 1.*a*) of the planetary gear receiving the motion from the driving portion of the tractor clutch by means of the shaft *1*. For PTO shaft *8* coupling, the brake *6* of central wheel *5* cuddle while weakening the brake *7* of PTO shaft. In this way crown 2 make to satellites to roll on the central wheel, moving the arm port - satellite *3* and *8* PTO shaft [1].
In this situation it is obvious that the planetary gear fulfills the role of a clutch. Based on this dynamic modeling in both cases (clutch or gearbox) can be made by introducing a *K* frictional coupling equivalent (Figure 1.*b*).
For mathematical modeling of *K* coupling can use either the law of variation of M_{fp} torqe transmitted to the slip friction surfaces, be a law to increase the angular velocity of the coupling party led to the equalizer with the angular velocity of the ruling party.

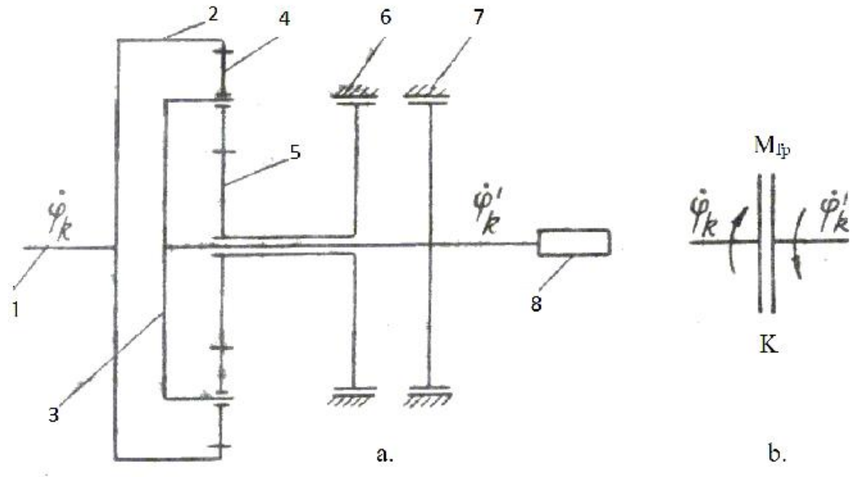

Figure1.The dynamic model of own PTO coupling

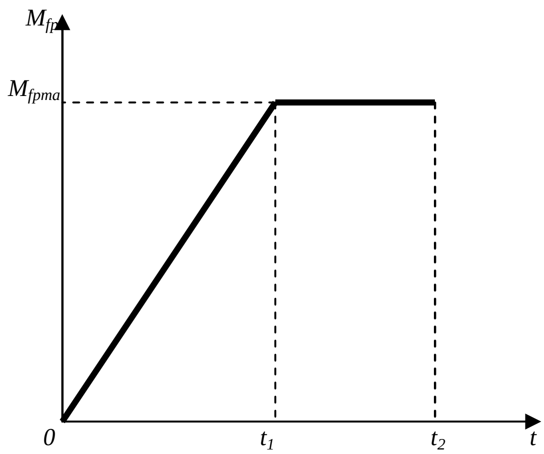

Figure 2.The M_{fp} function variation modeling in the PTO coupling process

By analogy with the main clutches coupling process[1], [3], it can be considered that the M_{fp} *(t)* function increases approximately linearly during coupling (Figure 2), ie the time between the start coupling ($t = 0; M_{pf} = 0$) and when it reaches maximum torque of friction ($t = t_1;M_{fp} = M_{fpmax}$), remaining constant until to further slip (at time t_2).
It follows that the variation of torque friction M_{fp} *(t)* can be modeled mathematically by linear function:

$$M_{fp} = \begin{cases} \dfrac{M_{fp\max}}{t_1} \cdot t & for \;\; 0 < t < t_1; \\ M_{fp\max} & for \;\; t \geq t_1 . \end{cases} \tag{1}$$

To calculate the size M_{fpmax} recommended relationship [1]:

$$M_{fp\max} = \quad \cdot M_n , \tag{2}$$

where*:* M_n is nominal torqe developed by the tractor engine; *ß* - the safety rating of the clutch, with values ß = 1.2 ... 1.5.
The expressions of friction torqe M_{fp} by equation (1) is suitable whilst the moving parts assets of farm machinery driven trough PTO shaft when the moment of resistance applied partly driven reaches the maximum moment of friction that can be developed between the coupling components . Construction features of the coupling PTO and specific way of increasing resistance due to working bodies of agricultural machines by their operation, sometimes do that, throughout the implementation moving and engaging, resisting moment applied partly driven coupling to remain lower than the frictional torque. Consequently, the size M_{fp} can not achieve maximum value at any time, taking intermediate values, that before solving the mathematical model can not be known. This situation can not be used for modeling equation (1) as it would be to artificially introduce a value of torque

friction coupling (M_{fpmax}) higher than that found practical. However, modeling can be done, as stated by introducing a law to increase the angular velocity of the driven part of the coupling.

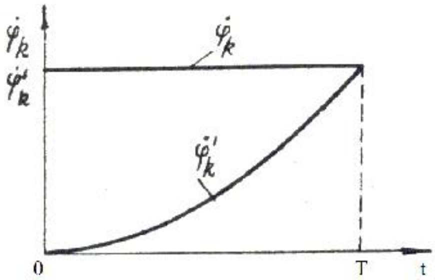

Figure 3. The variation of the angular speed of the driven part of the coupling *K* after a parabolic law

Taking into account that the engines of tractors are fitted with speed governors and assuming that during coupling PTO shaft angular velocity of the party led not change much, [2], [3], [4] proposes a expressing relationship parabolic growth (Figure 3) the angular velocity of the driven part of the coupling *K*, ie:

$$\dot{\varphi}'_k = \dot{\varphi}_k \left(\frac{t}{T}\right)^2 , \tag{3}$$

where: *t* is the current time; *T*- time at which the equalizer and angular velocities $\dot{\varphi}'_k$ and $\dot{\varphi}_k$ ($\dot{\varphi}'_k$ and $\dot{\varphi}_k$ the generalized coordinates of the movement led parties or conductive coupling PTO).

Since it was assumed that the moment of resistance is always lower than the maximum torque of friction that can be developed in the coupling that after the time t_1 that ends acting actuator, the clutch is not slipping, so they can know the value of time T ($T = t_1$).

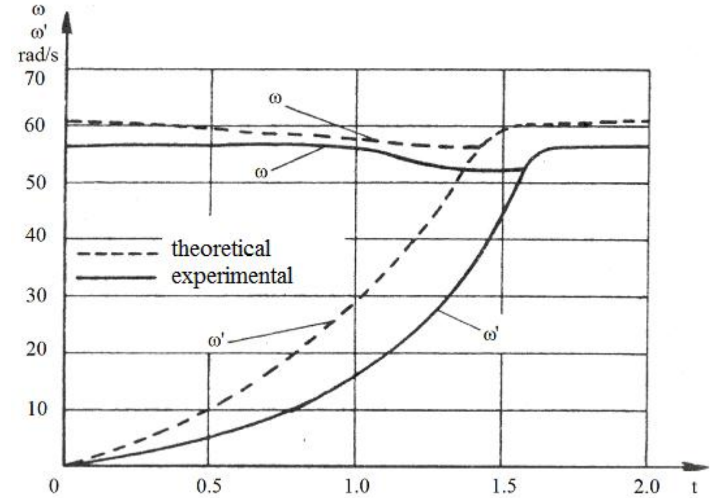

Figure 4. Theoretical and experimental variations of angular velocities of the coupling parts in the coupling process

When using equation (3) in the mathematical model written by applying cineto – still method on a dynamic model pentamasic system consisting of tractor U-650 and mechanically pressed round hay and straw PPF afforded evolution from Figure 4 the by angular velocity ω' of the PTO shaft with respect to the angular speed ω of the driving side of the coupling it to a coupling time of 1.4 *s*.

2.2. Modeling intermediate elastic coupling

To cushion the shock load coupling shaft PTO or experimental research of the influence of the elasticity constant change resilient of the link between the transmission PTO and agricultural machine is useful in this connection interposition of a flexible coupling.

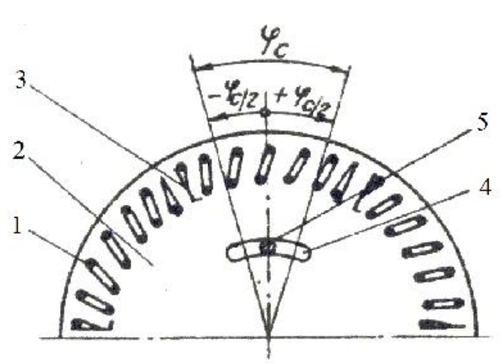

Figure 5. The scheme a flexible intermediate coupling

It is considered an intermediate elastic coupling schematically in Figure 5, provided with six coil spring *1,* placed between two identical discs *2*, each provided with brackets 3 for fastening the spring. One of the discs is provided with two channels, and the other wich two screws *5*, which ensure the mutual guide and prevent removal of the discs during operation.

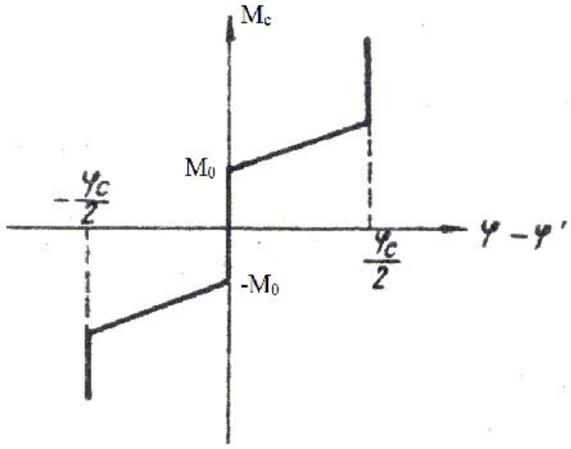

Figure 6. The variation of elastic torque M_c transmitted of the elastic coupling

Mathematical modeling of elastic torque M_c submitted by this coupling can be done by a linear equation set based on the representation of Figure 6, of the form:

$$M_C \begin{cases} \leq -M_0 - k_C \cdot \frac{\varphi_C}{2} & for \quad \varphi - \varphi' = -\frac{\varphi_C}{2}; \\ = -M_0 - k_C \cdot \left(\varphi - \varphi'\right) & for \; -\frac{\varphi_C}{2} < \varphi - \varphi' < 0; \\ \in (-M_0; +M_0) & for \quad \varphi - \varphi' = 0; \\ = M_0 + k_C \cdot \left(\varphi - \varphi'\right) & for \quad 0 < \varphi - \varphi' < \frac{\varphi_C}{2}; \\ \geq M_0 + k_C \cdot \frac{\varphi_C}{2} & for \quad \varphi - \varphi' = \frac{\varphi_C}{2}. \end{cases} \tag{4}$$

wherein: M_c is time that can be transmitted due to initial consecutive springs compression; φ_c - the maximum angle of relative rotation of the two discs of the clutch; φ and φ' - yawing of the ruling, respectively led coupling party; k_c - elastic constant coupling.

2. 3. Modeling safety clutch

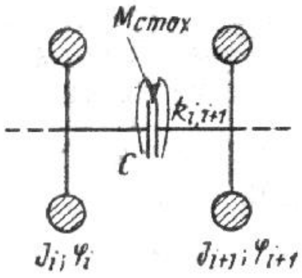

Figure 7 The dynamic model of safety clutch

To protect against overload of important elements of the transmission mechanism of agricultural machinery, frequently ceding limiting couplings are used, interrupting the transmission of movement to achieve a certain amount of elastic torque on request. Neglecting these couplings in dynamic and mathematical transmission mechanisms modeling causing removal of the theoretical research results obtained by experimental research.

Such a coupling can be reproduced in a mathematical model by limiting the elastic torqe on the respectively section of the transmission, by maximum torque value M_{cmax} amount of time that can be transmitted by coupling. Thus, if the clutch *C* is inserted between low weight J_i ..J_{i+1} of a certain dynamic model (Figure 7), can be written:

$$M_{i,i+1} = \begin{cases} k_{i,i+1} \cdot (\ddot{\ }_{i} - \ddot{\ }_{i+1}) & \text{for} \quad k_{i,i+1} \cdot (\ddot{\ }_{i} - \ddot{\ }_{i+1}) < M_{c\max}; \\ M_{c\max} & \text{for} \quad k_{i,i+1} \cdot (\ddot{\ }_{i} - \ddot{\ }_{i+1}) \geq M_{c\max}. \end{cases} \quad (5)$$

where: $M_{i,\ i+1}$ is the elastic torqe in the relationship between the masses J_i, J_{i+1}; $k_{i,\ i+1}$ -equivalent spring constant of this link; φ_i and φ_{i+1} – mass movement J_i, respectively J_{i+1}.
Setting to M_{cmax} to made with specific relationship specific safety plugs gear couplings, with friction surfaces etc.

3. CONCLUSIONS

1. By dynamic and mathematical modeling of PTO drives mechanisms the tractor and agricultural machine driven by this, it is necessary to take into account the proper PTO clutch and possible elastic couplings and safety of the two transmission mechanisms of movement and the elastic torqe belive tractor and agricultural machinery.
2. PTO clutch proper behavior of particular importance that it has to start working bodies operated. By using, where appropriate, the law of variation of friction torque or the law of increasing the angular velocity of the party led during coupling, can highlight the influence of this coupling the regime dynamic stresses in the specific conditions of any mathematical model of the mechanism of transmission PTO power.
3. Mathematical modeling of the coupling and the safety spring (limiting), involves the use of equations to control the evolution of the elastic torqe the links that contain the same coupling in a manner determined by the constructive and functional characteristics thereof.

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