The 3rd International Conference on "Computational Mechanics and Virtual Engineering"
COMEC 2009
29 – 30 OCTOBER 2009, Brasov, Romania

CONTRIBUTIONS TO THE STUDY OF THE DYNAMIC STABILITY OF AGRICULTURAL TRACTORS EQUIPPED WITH FRONT LOADER

Simion Popescu[1], Vlad Popescu[2], Ion Cândea[3]
Transilvania University of Brasov/Romania, simipop38@yahoo.com
Transilvania University of Brasov/Romania, vladpop@unitbv.ro
Transilvania University of Brasov/Romania, candmec@ unitbv.ro

Abstract*. In the paper is analysed the longitudinal stability of tractor-front end loader system in the most difficult work situations: descending on a slope, braking in translatory motion and acceleration of the bulk while lifting the load. Based on the equivalent dynamical models of the real tractor-loader systems are elaborated the mathematical models describing the dynamical behaviour of the systems under different working transportation conditions, which deliver the criteria for the dynamic longitudinal and transversally stability. The computer simulation allows the study of the overturning stability of the systems trough application for the constructive tractor-loader models in different working conditions.*
Keywords: *agricultural tractor, front end loader, dynamic model, mathematical model, overturning stability*

1. INTRODUCTION

The loaders mounted on the front end of agricultural wheel tractors are produced by specialized manufacturers and the range of constructive and dimensional variants offered on the market are easily adapted to mounted on and dismounted from most actual types of tractors. Further, the manufacturers offer a wide range of working (loading) organs, which can be mounted, as needed, on the lifting arm of the loader. In general, front loaders are currently used with small and medium power tractors (up to 75kW), ensuring lifting forces (charges) up to 20 kN [2; 4].

Despite the constructive variety of front loaders for tractors, their basic construction is similar (fig. 1). A front loader consists of a lifting arm 1 mounted on the tractor body by means of a support 2. The oscillation of arm 1 in the longitudinal – vertical plane is achieved by the hydraulic cylinders 3 (one on each side). The working organ 4 is mounted at the end of arm 1, the unloading of materials being achieved by its rotation (dumping) driven by hydraulic cylinder 5. In order to ensure the longitudinal stability during working and motion of the tractor – loader combination (system), at the rear of the tractor a counterweight 6 is coupled to the rear three points hitch of the tractor.

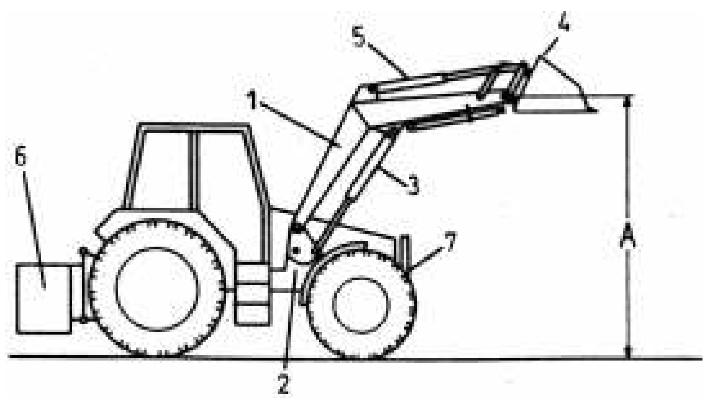

Figure 1. The basic constructive and functional elements of the front end loader mounted on the agricultural tractor

In order to protect the front part of the tractor from possible contacts (collisions) with the bumpers of other vehicles or from materials falling from the working organs, a shield 7 is mounted on the front part of the tractor. Further, constructively a certain unloading distance is ensured between the front part of the tractor and the bucket in its unloading position.

The constructive and functional parameters of front loaders mounted on agricultural wheel tractors have to satisfy the requirements of the working process and of the dynamic stability and have to correspond to the structures of the tractors they are mounted on.

2. MATERIAL AND METHODS

2.1. Longitudinal dynamic stability of tractor- front loader system

During traveling between the loading and the unloading place, the tractor- front loader systems are frequently subjected to braking processes, which under certain circumstances may cause the loss of longitudinal stability, by overturning round the front axle [1;5]. The diagram of figure 2 considers the most difficult situation in regard of stability, when the system is braked with during descending a longitudinal slope, at the same time with the braking of the charge during the lowering process (when inertia force acts upon the charge *Q*).

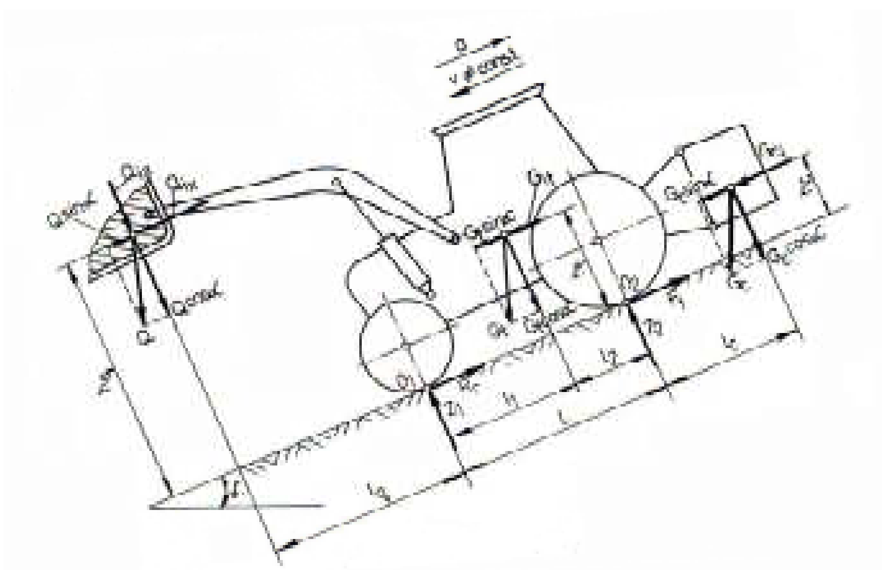

Figure 2. Exterior forces acting upon the tractor-front loader system during downhill braking

The exterior forces acting upon the system consisting of the tractor and the front loader with the filled bucket raised into transport position (height h_m) are as described in figure 1: G_t –own weight of the tractor; Q –total load of the working component (bucket); G_c – weight of the counterweight mounted at the rear of the tractor(including own weight of the loader arm and the weight of the empty working component); Z_1 and Z_2- loads on the front and rear axle, respectively; F_f –total braking force developed on the braked wheels of the tractor (achieved by the adherence of the wheels to the road surface); R_r=f $(Z_1 + Z2)$ - total rolling resistance of the tractor wheels (where *f*. is the rolling resistance coefficients of the tractor wheels The weights G_t, Q and G_c, are located in the mass centres of the respective components and are identified by the coordinates indicated in figure 2 (distance *L* represents the wheel base of the tractor).

Due to the braking of the tractor the force F_f *is* developed on the tractor wheels, and therefore the system is subjected to inertia forces generated by the deceleration *d,* forces which are parallel to the road surface and placed in the mass centres of the tractor body $G_{ti} = d.G_t/g$), counterweight $G_{ci} = d.G_c/g$, and load $Q_{ix} = d.Q/g$, respectively. During braking of charge Q while lowering, corresponding inertia forces $Q_{iz} = Q.a_q$ upon the fork, generated by the deceleration a_q

The total braking force F_f developed on the tractor wheels by the adherence to the road surface depends on the actuating mode of the brakes. In the case of four-wheel drive (4WD) tractors the wheels of both tractor axles are braked (integral braking) and, therefore, the maximum braking force is given by the relationship: $F_{fmax}= \varphi(Z_1+Z_2)$, where φ is the adherence coefficient of the wheels to the road surface. In the case of rear driven tractors (4x2), only the rear axle wheels are braked (with normal load Z_2, so that the maximum braking force has the value: $F_{fmax}= \varphi Z_2$.

Due to the small values of the rolling resistance coefficients *f* compared to those of the adherence coefficient φ, the rolling resistance R_r can be neglected (i.e. $R_r = 0$).From the equation of equilibrium of the system in traveling direction the braking force F_f is expressed by equation:

$$F_f = (G_t + Q + G_c)\left[\sin\alpha + (d/g)\right]. \tag{1}$$

The normal loads Z_1 and Z_2 on the front and rear axle of the tractor follow from the equilibrium equations of the equivalent dynamic model of figure 1, and are given by the following expressions:

$$Z_1 = \frac{\left[G_t l_2 + G_c l_c + Q.(L + l_q)\right]\cos\alpha}{L} + \frac{(G_t h_t + G_c h_c + Q.h_q)\sin\alpha}{L} + \frac{(G_t h_t + G_c h_c + Q.h_q).d + Q.l_q.a_q}{L.g}; \tag{2}$$

$$Z_2 = \frac{\left[G_t l_1 - G_c(L + l_c) - Q.l_q\right]\cos\alpha}{L} - \frac{(G_t h_t + G_c h_c + Q.h_q)\sin\alpha}{L} - \frac{(G_t h_t + G_c h_c + Q.h_q).d + Q.l_q.a_q}{L.g}. \tag{3}$$

The longitudinal overturning stability of the system for various values of the slope tilt angle α and the braking deceleration d is ensured if the tractor is counterweighted by a minimum weight $G_{c.min}$, determined from equation (3) by condition $Z_2 = 0$, wherefrom follows:

$$G_{c.\min} = \frac{G_t\left[h_t(\sin\alpha + d/g) - l_1\cos\alpha\right] + Q.\left[h_q(\sin\alpha + d/g) + l_q\cos\alpha\right]}{(L + l_c)\cos\alpha - h_c(\sin\alpha + d/g)} \tag{4}$$

The value of the maximum deceleration d_{max} for which the longitudinal overturning stability is ensured is determined from equation (3) by condition $Z_2 = 0$. Solving the equation in function of deceleration d yields the expression of the maximum admissible braking deceleration for traveling conditions is expressed by equation:

$$d_{\max} = g\frac{\left[G_t.l_1 + G_c(L + l_c) - Q.l_q\right]\cos\alpha - (G_t h_t + G_c h_c + Qh_q)\sin\alpha}{G_t h_t + G_c h_c + Qh_q}. \tag{5}$$

For maximum intensity braking the maximum deceleration d_{max} depends on the load on the braked wheels and the adherence coefficient to the road φ. For 4 WD tractors braking is carried out on both axles (integral braking), so that the maximum braking force will be: $F_{fmax} = \varphi\ (Z_1+Z_2)$. Considering equations (2) and (3) the maximum achievable value of the deceleration d_{max} is given by equation:

$$d_{max} = g\ (\varphi\cos\alpha - \sin\alpha). \tag{6}$$

For rear driven tractors (2WD tractors) with barking of the rear wheels only, the maximum braking force is $F_{fmax} = \varphi\ Z_2$. Considering equations (2) and (3), the maximum achievable value of the deceleration d_{max} is given by equation:

$$d_{\max} = g.\varphi.\frac{\left[G_t l_1 - G_c(L + l_c) - Q.l_q\right]\cos\alpha - (G_t h_t + G_c h_c + Q.h_q)\sin\alpha - L(G_t + G_c + Q)\sin\alpha}{L(G_t + G_c + Q) + \varphi.(G_t h_t + G_c h_c + Q.h_q)}. \tag{7}$$

3. TRANSVERSALLY DYNAMIC STABILITY OF TRACTOR- FRONT LOADER SYSTEM

Another important aspect of the tractor - front loader system dynamics is the motion of the charged bucket on transversal slopes with the angle β (Fig. 3). From the equilibrium equations of the system the normal reaction (perpendicular on the road surface) in the uphill Z_s is given by the expression:

$$Z_s = \frac{0{,}5E(G_t + Q)\cos\beta - (G_t h_t + Q.h_q)\sin\beta}{E} \tag{8}$$

where B is the track width of the tractor.

The stability to lateral (transversal) turning over of the tractor – front loader system is given by the condition $Z_s > 0$, wherefrom, following transformations the expression of the stability condition to lateral turning over follows:

$$tg\beta \le \frac{0{,}5E.(G_t + Q)}{G_t h_t + Q.h_q}. \tag{9}$$

By means of relation (9) the values of maximum angle β_{max} of the transversal slope on which the tractor may move without overturning on one side, may be determined.

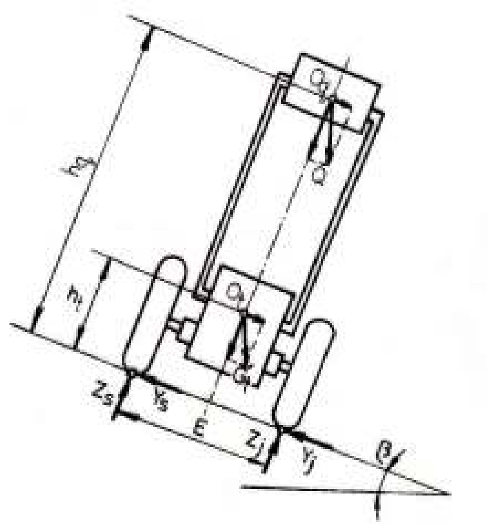

Figure 3. Forces acting on the tractor - loader system when moving on a transversally slope with the angle β

When the turnover is negotiated uphill, it appears the critical situation because the centrifugal force is added to the G_t *sinß* and Q *sinß* components, contributing thus to the decrease of charge Z_s and, as a result, to the reduction of the turnover stability.

To a low lateral adhesion of the road the tractor on a transversally slope with the angle β may slide sideways. The stability against sliding is given by the condition (1;7]:

$$(Y_s + Y_j) \geq (G_t + Q)\sin\beta, \qquad (10)$$

where Y_s and Y_j are the lateral forces developed on the tire-ground patch (usually named the cornering forces).
Considering that $(Z_s + Z_j) = (G_t + Q)\cos\beta$, the maximum force at sliding is given by relation:

$$(Y_s+Y_j)_{max} = \varphi_y\,(Z_i + Z_s), \qquad (11)$$

where φ_y is the friction coefficient for wheels sliding.
Substituting equation (11) into relationship (10) results:

$$\varphi_y(G_t + Q)\cos\beta \geq (G+Q)\sin\beta\,. \qquad (12)$$

Based on relation (12), the condition of the stability of the system against sliding can be written as:

$$tg\beta \leq \varphi_y\,. \qquad (13)$$

Generally, the value of friction coefficient for sliding φ_y is equal to 80 % of the value of the coefficient of traction and braking force φ [7]:$\varphi_y = 0.8.\varphi$.
For a reduced road adhesion coefficient the loss of side-sliding stability is possible to occur before the loss of transversally turnover stability.

4.RESULTS AND DISCUSSION

The computer aided solving (simulation) of equation (3) allows the analysis of the variation of the rear axle loads Z_2 and implicitly the longitudinal stability of the tractor depending on the values of the descended slope tilt angle α and the braking deceleration d, considering the front loader bucket carrying various loads Q and raised to various heights h_q ,plotted for the case of the system consisting of a 2WD wheel tractor U-650 M (made in Romania) [5;6], with wheel base L = 2400 mm, weight G_t = 4000 N and coordinates l_2 = 952 mm; h_t = 1263 mm, equipped with an *IF 65* front end loader (made in Romania) and an counterweight G_c = 8000 N, for traveling with the bucket carrying the rated load (Q =10000 N) raised to transport height h_q = 2120 mm.

The graph from figure 4 shows that for braking deceleration values of d = 2.5 m/s^2, the longitudinal overturning of the system (Z_2 = 0) occurs for tilting angles of the slope of about α = 0,57 rad (α = 33^0). For braking deceleration values of d = 5.0 m/s^2 (achievable by 4WD tractors during braking on flagstone or dry earth roads), the longitudinal overturning of the system occurs at much smaller tilting angles of the slope, of about α = 0,27 rad (α = 16^0)..

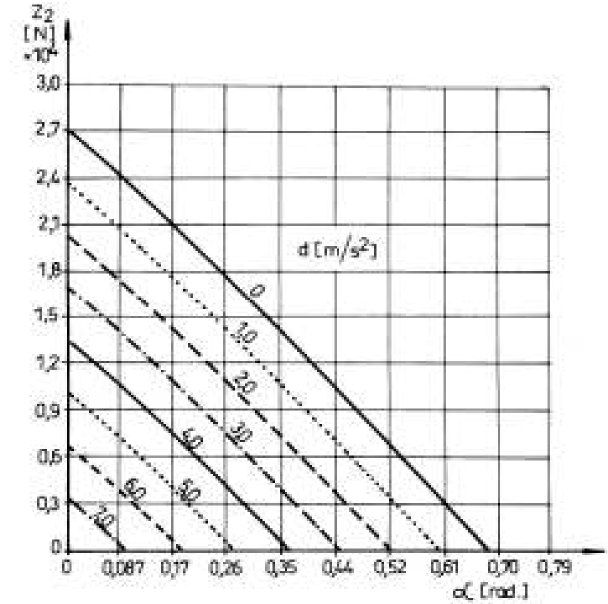

Figure 4. Variation of loads Z_2 on the tractor rear axle during downhill braking in dependence on the slope angle α, for different braking decelerations d

The braking adherence coefficient φ depends on the road type and state and has the values: φ = 0,70…0,75- for dry concrete and asphalt; φ = 0,5…0,7- for wet concrete and asphalt; φ = 0,5…0,6- for flagstone roads; φ = 0,65- for dry earth roads; φ = 0,4…0,5 - for wet earth roads [6;7].

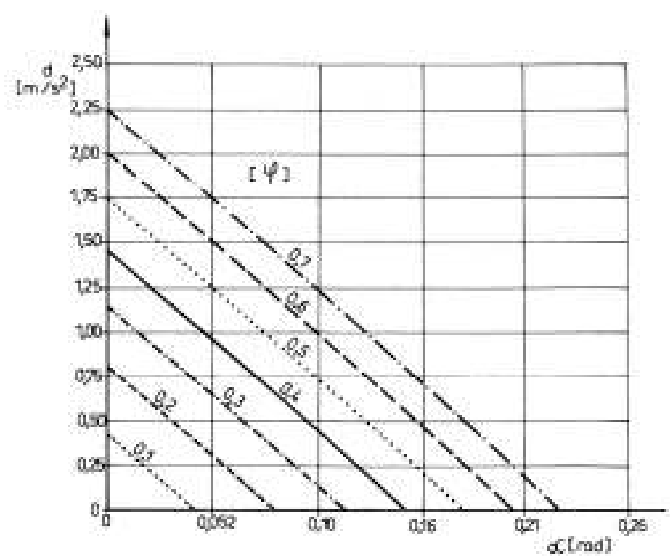

Figure 5. Variation of the maximum braking deceleration d_{max} achievable by the 2WD agricultural tractor U- 650 depending on the slope tilt angle α and different adherence coefficients of the road φ

Figure 5 presents a graph highlighting the variation of the maximum braking deceleration d_{max} depending on the values of the descended slope tilt angles α, for different adherence coefficients of the road φ, taking values between φ = 0...

0.7, during the braking of the system including the 2WD agricultural tractor *U 650 M* and the front loader *IF 65* [5;6]. The graph was obtained by computer simulation of equation (7) for the system traveling with a bucket carrying the rated load (Q = 7000 N) and raised to a medium height (h_q = 2120 mm). It can be noticed that for braking on surfaces of adherence $\varphi = 0,7$ (asphalt, concrete) and traveling on horizontal roads ($\alpha = 0°$) the braking deceleration reaches values of about 2.25 m/s^2. With an increasing tilt of the slope, the maximum braking deceleration d_{max} decreases significantly, reaching 1.25 m/s^2 for angles α = 6...7^0, and tends towards zero for slopes of tilt angle α = 12…15°.

5. CONCLUSIONS

1. The front loaders mounted at the front-end of agricultural wheel tractors are increasingly employed for the mechanization of material loading and unloading operations into/from transport means or other locations on low and medium agricultural farms.
2. The constructive and functional parameters of front loaders mounted on agricultural wheel tractors have to satisfy the requirements of the working process and of the dynamic stability and have to correspond to the structures of the tractors they are mounted on.
4. The braking of the tractor equipped with front loaders during descending a longitudinal slope with the filled bucket in transport position is in relation to the longitudinal stability of the system the most difficult situation of the traveling process.
4. The dynamics of tractor – front end loader systems can be analyzed by mathematical modeling of the equivalent dynamic models of the real systems, taking into account the exterior forces to which they are subjected in various working situations.
5. Based on the equivalent dynamical models of tractor – front end loader systems it can be elaborated the mathematical model describing the dynamical behaviour of the system during the descending on a slope by slowing down (breaking) of the vehicle and acceleration of the fork while lifting the load.
6. The mathematical models tractor –front loader systems allow the analysis of the overturning stability of the systems, for various concrete working and traveling conditions. The mathematical models deliver the criteria for the overturning stability.

REFERENCES

[1]. Czyborra, M., S. Popescu and D. Karamousantes. Particularities of the Dynamics of the Agricultural Tractor equiped with Front Loader. In: III International Research and Technical Conference, MOTROL 2001, Lublin (Poland), Vol.1, pp.86-93. [2]. Froba, N.: 1999 Traktoren – Frontlader. Landtechnik, nr. 3, 2001. pp. 157 – 162.
[2]. Froba, N. Traktoren – Frontlader. Landtechnik, nr. 3, 1999, pp. 157 – 162.
[3]. Karamousantas, D., Ros, V., Gheres, M.I., Fechete, L.V. Optimization of the Driving Mechanism of the Hoe Machines. The 29. International symposium on agricultural engineering, Opatija/Croatia, 06. – 09, February 2001
[4]. Ortiz-Canavate, J.Tecnica de la mecaizacion agraria. Ediciones Mundi – Prensa, Madrid, 1989, pp. 223 – 232.
[5]. Popescu, S.,. Czyborra, M., Padureanu, V.: Research on the Longitudinal Stability of Tractor – Front Loader Systems during Braking. In: Proceedings of the Union of Scientists „Energy Efficiency and Agricultural Engineering", Rousse (Bulgaria), 2004, pp. 439-445.
[6]. Popescu, S. Contributions regarding the experimental Research on the Dynamic Behaviour of the Tractor Front Loader System. In: Bulletin of the Transilvania University of Brasov, Serie A, Vol 11 (46), 2004, pp. 101-106.
[7]. Wang, I. Y.: Theory of Ground Vehicles. John Wiley and Sons, New York, 1993.