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THE DYNAMICS OF THE MOVING TRANSMISSIONS

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ABSTRACT

The chargers on tire with direction through side-slip are equipments for used loading different materials, having very good manoeuvrability in narrow places due to the possibility of turning around their centre of weight. The turning is produced result of the possibility to reach different speeds movement on the couples of left and right wheels, similarly to the equipments with movement on caterpillars. Due to this possibility, the equipment is able perform turnings with different radius, including turning on the spot.

KEYWORDS: direction, side-slip, dynamics of moving transmissions

1. THE STRESSES APPLIED ON THE MECHANISM OF MOVEMENT IN CASE OF UNIFORM TURNING

The forces and the moments that solicit the displacement mechanism of these equipments during the uniform turning executed on horizontal ground at a small speed are the equipment weight, the charge weight, the backward push of the soil and the resistant moment of turning.

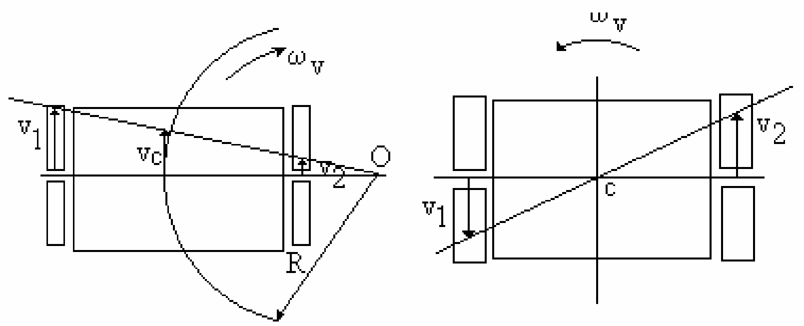

Figure 1

The backward push of the soil appears during movement of the equipment by vertical deformation of the soil and tires (figure 2). Also, during turnings, a lateral movement of the tires appears, leading to a lateral deformation of the tires and soil, as well as the soil cluster and transport, these phenomena leading to the appearance of turning resistant forces. These T_{r1} forces create the moment of turning resistance.

On the ensemble of the equipment displacement mechanism, which executes the turning, operate the forces and the moments presented herein before, as in fig. 3, where:

A - the axle base of the equipment; *B* - the gauge of the equipment; x_G - distance from the deck of the equipment to the barycentre (depending on the shipment of the bucket); *R* - radius of barycentre; *C* - the position to a certain barycentre loading, considering the symmetrical loading against the longitudinal axis of the equipment; F_1, F_2, F_3, F_4 – pulling forces.

In conditions of equilibrium to limit:

$$F_i = \frac{M_m - N_i \cdot s}{r} \tag{1}$$

where $N_i = G_{pi}$.

G_{pi} - the loading on each wheel of the equipment; *r* - static radius of the used tire

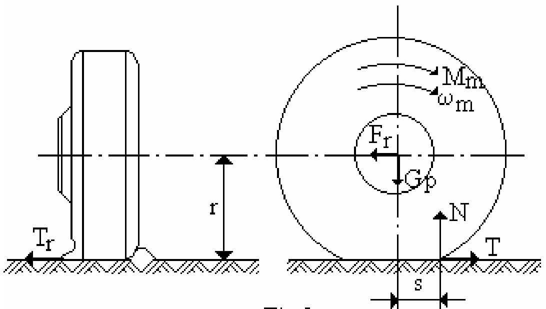

Fig.2

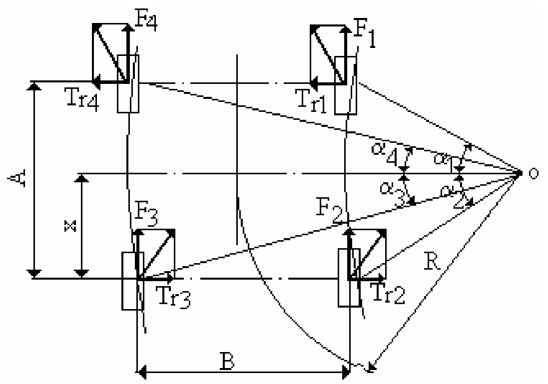

Fig. 3

The sum of the moments of the forces against the centre of the turning is (fig. 3)

$$\sum M_o = 0$$

$$\begin{aligned}\sum M_o &= \left(F_1\cos\alpha_1 + T_{r1}\sin\alpha_1\right)\frac{R-\frac{B}{2}}{\cos\alpha_1} + \\ &+ \left(F_2\cos\alpha_2 + T_{r2}\sin\alpha_2\right)\frac{R-\frac{B}{2}}{\cos\alpha_2} + \\ &+ \left(F_3\cos\alpha + T_{r3}\sin\alpha_3\right)\frac{R-\frac{B}{2}}{\cos\alpha_3} + \\ &+ \left(F_4\cos\alpha_4 + T_{r4}\sin\alpha_4\right)\frac{R-\frac{B}{2}}{\cos\alpha_4}\end{aligned} \qquad (2)$$

When the tire loading is the same, therefore $Gp_1 = Gp_2 = Gp_3 = Gp_4$, the barycentre of the equipment is in:

$$x_G = \frac{A}{2} \qquad (3)$$

In this case

$$\alpha_1 = \alpha_2;\ \alpha_3 = \alpha_4;\ F_1 = F_2;\ F_3 = F_4$$

and

$$T_{r1} = T_{r2}; T_{r3} = T_{r4}$$

therefore, the relation of equilibrium becomes:

$$\begin{aligned}\sum M_o &= 2\left(F_1 - T_{r1}tg\alpha_1\right)\left(R-\frac{B}{2}\right) + \\ &+ 2\left(F_3 - T_{r3}tg\alpha_3\right)\left(R-\frac{B}{2}\right) = 0\end{aligned} \qquad (4)$$

Considering that $T_{r1} = \mu \cdot G_{pi}$, the relation can also be written as follows:

$$\begin{aligned}&2\left(F_1 - \mu\cdot G_p tg\alpha_1\right)\left(R-\frac{B}{2}\right) + \\ &+ 2\left(F_3 - \mu\cdot G_p tg\alpha_3\right)\left(R-\frac{B}{2}\right) = 0\end{aligned} \qquad (5)$$

therefore the global friction coefficient can be determined:

$$\mu = \frac{F_{1,2}(2R-B) + F_{3,4}(2R+B)}{2G_p\cdot A} \qquad (6)$$

In the particular case when R = 0, the turning is performed around the barycentre, as shown in fig.4.

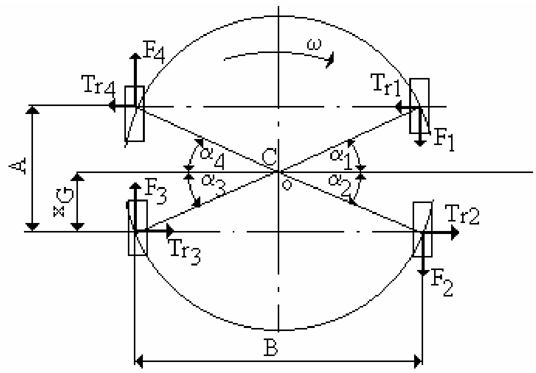

Fig. 4

$$\sum M_o = \sum_{i=1}^{4}\left(F_i\cos\alpha_i - \mu G_{pi}\sin\alpha_i\right)\frac{B}{2\cos\alpha_i} = 0 \qquad (7)$$

For $x_G \le \frac{A}{2}$ and if the loading against the longitudinal axis is symmetrical, results

$G_1 = G_4;\ G_2 = G_3$ and

$F_1 = F_4;\ F_2 = F_3$

The relation becomes

$$\begin{aligned}&2\left[F_{1,4} - \mu G_{p1,4}\cdot\frac{2(A-x_G)}{B}\right]\frac{B}{2} + \\ &+ 2\left[F_{2,3} - \mu G_{p2,3}\cdot\frac{2(A-x_G)}{B}\right]\frac{B}{2} = 0\end{aligned} \qquad (8)$$

resulting the global coefficient:

$$\mu = \frac{(F_{1,4} + F_{2,3})B}{2G_{p1,4}(A - x_G) + 2G_{p2,3} \cdot x_G} \qquad (9)$$

In the conditions of symmetrical loading on tires, it can be considered:

$$x_G = \frac{A}{2};\ F_i = F\ ;\ G_{pi} = \frac{G}{4};\ \alpha_i = \alpha$$

G - the total weight of the equipment (dead load plus charge from bucket), therefore

$$\sum M_o = 4\left(F\cos\alpha - \mu\frac{G}{4}\sin\alpha\right)\frac{B}{2\cos\alpha} = 0$$

or

$$4\left(F - \mu\frac{G}{4}tg\alpha\right)\frac{B}{2} = 0$$

But

$$tg\alpha = \frac{A}{B} \qquad (10)$$

therefore

$$4\left(F - \mu\frac{G}{4}\cdot\frac{A}{B}\right)\cdot\frac{B}{2} = 0$$ hence it results that

$$F = \mu\frac{G}{4}\cdot\frac{A}{B} \qquad (11)$$

It is noticed from the relation that the pulling force that must be developed by each of the driving wheels in order to potentate the turning on the spot depends on:
- the friction between soil and tire;
- the loading on tire;
- the report between the axle base and the gauge

Therefore, the conclusion is that, for these pulling forces which are necessary for the turning, the element which can be acted upon, when designing the equipments of this kinds, is the report *A/B*.

2. EXPERIMENTAL DATA

The work is an attempt to experimentally verify the previously suggested hypotheses regarding the determination of a global coefficient of proof run on the whole equipment for different loadings on motion decks and for different radius of turning. It is intended, also, to establish the power consumption of the equipment for the execution of the turning.

The experimentations performed do not totally solve the problem. It must be mentioned the fact that such gearboxes for equipments were not analysed by the specialty literature because these kind of equipments appeared not so long ago and they cannot be assimilated to the classic gearboxes on caterpillar.

The measurements of the necessary parameters of the suggested experimentations were done merely for the movement on macadam, for a single value of the pressure of the tire (6 bar) and only for some usual radius of turning.

The measured parameters were:
- pressure drops on the 4 hydraulic engines of wheels driving;
- the absorbed flows for movement;
- the loading on the motion decks;
- radius of turning.

3. ANALYSIS OF EXPERIMENTAL DATA

3.1. In case of equal tire loading with a proper charge in the bucket (maintained at a constant height), it was noticed,

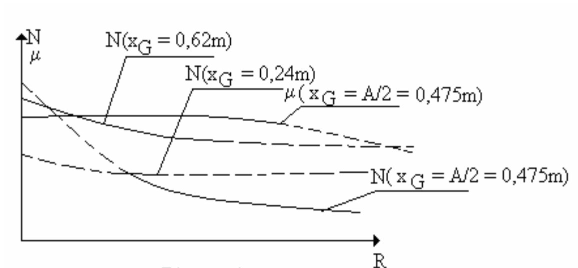

Diagram 1

- when executing the turning around the barycentre (X_G = *A/2*), the necessary moments to the driving wheels and the power absorbed by them are approximately equal, confirming the initial hypothesis;
- it was determined, by using the experimental data, a global coefficient of proof turning, with the help of the suggested formula, the value being variable with the turning radius, as it can be seen in the diagram no.1;
- for the turnings with different radius, the total consumption of power varies, having a descending line, from a maximum value when turning on the spot to a minimum value, when running in-line ($R = \infty$ when $\mu = f$ - coefficient of proof run) as shown in diagram no. 1

3.2. In case of discontinuous loading on the decks, through the variation of the charge from bucket, with the preservation of the symmetry of the loading of the

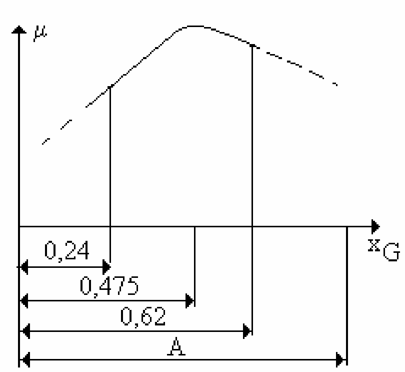

Diagram 2

tire against the longitudinal axis of the equipment, there have been drown the following conclusions:

- the global coefficient of proof turning around barycentre varies with the loading on the decks, as shown in diagram no. 2;
- it is noticed that it has the maximum value when X_G=A/2
- in case of turning with an inactive board, with different braking moments on this board's wheels, for constant revolutions of the wheels of the active board, different radius of turning were obtained. The power consumption at that rate, for different radius of turning varies with the decks loading; this can be seen from diagram no. 1

4. CONCLUSIONS

The experimentations were partially confirmed by the suggested hypotheses, having a global coefficient of proof turning.

For the further activity of designing of such equipments, it is necessary to continue the experimentations for different grounds and to expand the analysis about behaviour of each driving wheel depending on its position when turning and on loading.

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