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CONSIDERATIONS REGARDING THE STEERING AND LINEAR DRIVE MECHANISM WITH DOUBLE DIFFERENTIAL FOR TRACKS MACHINES

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Abstract: *This paper focuses on the design and modeling of a steering mechanical drive system for tracked machines that have a double conical differential mechanism. The next objective in the continuation of the paper is oriented towards the mathematical modeling of the proposed mechanical system that will contain the kinematics and dynamics of the double differential depending on the movements made by such a machine. The double conical differential is a complex mechanism that, used in excavator movement applications, allows the tracks to rotate independently of each other. Thus, the two tracks of the excavator will have the possibility to cover different distances when cornering or when traveling on rough terrain with irregularities.*

Key words: *crawler excavator, earthmoving machine, undercarriage, track drive, steering mechanism, double differential kinematics, double differential dynamics.*

1. INTRODUCTION

Excavators are part of the family of earthmoving machines, that deserving important sectors of the mining industry, agriculture, and road infrastructure [1]. These types of earthmoving machines are perfect for applications and work on construction sites, with impressive capabilities for digging, lifting, breaking, and moving various materials. The end-effectors that can be attached to an excavator are diversified, for example: buckets of varied sizes and types, wood grippers, hydraulic hammers and many more accessories for drilling, piling, pipes manipulations, crushing concrete, etc. The tasks that must perform are very versatile and difficult to accomplish which involve a multitude of challenges [2]. These excavators usually dig below the support level of the base machine by moving the bucket towards the base machine. The diagram and component parts of the reverse bucket excavator are given in figure 1. On the rotating platform there is also the diesel engine, the pump set, the hydraulic equipment, the cab with the control

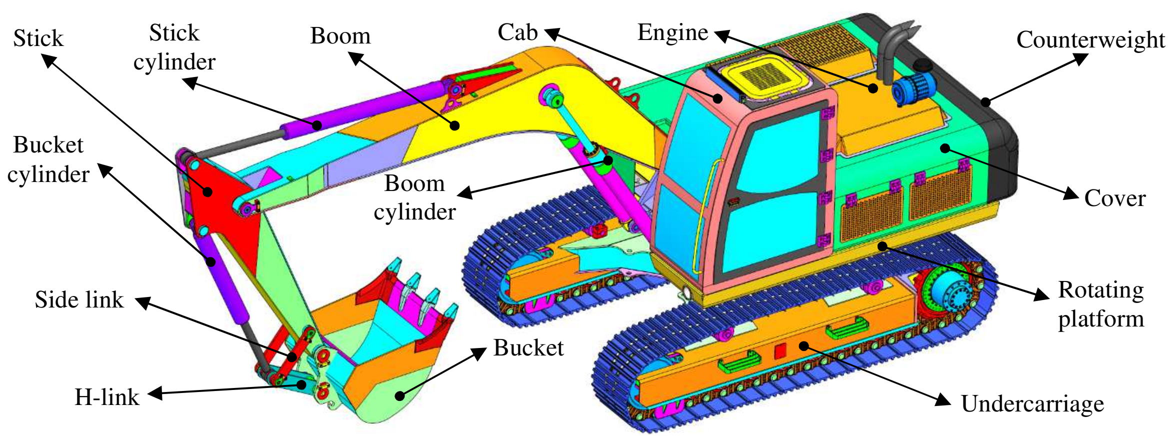

Fig. 1. Basic parts of hydraulic crawler excavator

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and command system, the work equipment consisting of boom, stick, hydraulic cylinders, and bucket.

The paper focuses on undercarriage and especially on the steering drive mechanism and its design. The undercarriage of an earthmoving machine is a main component of its construction and can be divided into two main categories depending on the travel system: with driving wheels or driving tracks [3], as show in figure 2.

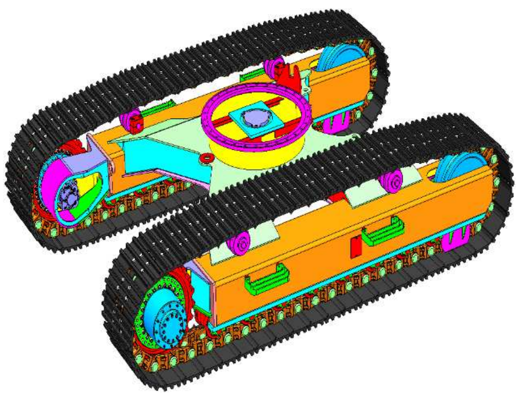

Fig. 2. Excavator undercarriage with steel track

The functions of undercarriage must support the weight of the entire excavator as well as, to move forward and backward but, also to turn in different directions. The main components of an excavator undercarriage with steel track drive are the following: sprocket, carrier rollers, track rollers, track shoes, track links, idler, track adjuster-tensioner and hydraulic drive with planetary reducer, as show in figure 3.

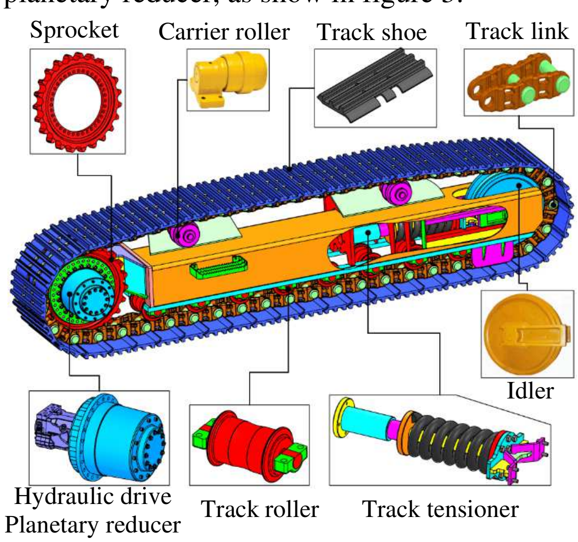

Fig. 3. Excavator undercarriage components

The track is a closed belt that supports the excavator on the ground and moves it. For heavy excavators, the tracks are made of steel, and they are made of links articulated with bolts on which the track shoes will be mounted.

The main advantage of crawler excavators is the uniform distribution of weight during operation, with low contact pressure with the tracks-ground which will result in high traction and stability. So, excavators are effective means of increasing labor productivity by increasing the degree of mechanization.

2. STEERING AND LINEAR DRIVE MECHANISM WITH DOUBLE DIFFERENTIAL

From constructive point of view, the tracks are driven by a hydraulic motor and a planetary gearbox or by a mechanical mechanism.

The author's concept approaches the mechanical variant of moving the excavator and for its construction it will use: two input bevel gears, a double differential, two main sliding bearings, two drum braking systems, two drums, two flexibles coupling, two output drive shafts, two rolling bearing units, and two sprockets for final drive, as figure 4.

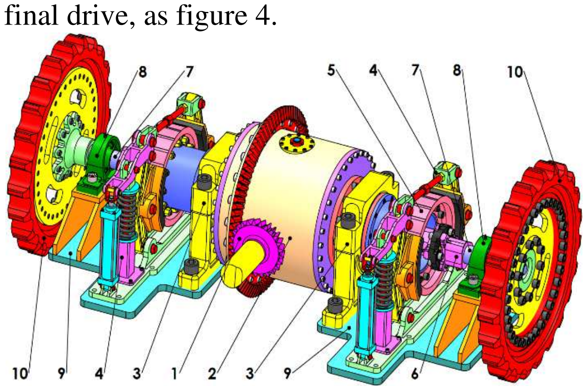

Fig. 4. CAD model of the excavator steering system with double differential

1- input bevel gears; 2- double differential; 3- main sliding bearings; 4- drum braking system; 5- drum; 6- flexible coupling; 7- drive shaft; 8- bearing unit; 9- chassis support; 10- sprocket drive wheels.

Further will be described the operation of the mechanical system that drives the excavator. As can be seen from figure 5, the main mechanism in the transmission system is based on a double conical differential, consisting of bevel gear (1) and satellite gears box (2). In the inside of satellites gearbox is the differential group consisting of satellite carrier (3) and the outer planetary gears pinion (6), (7) as well as the inner planetary gears pinion (8), (9). The outer planetary gears pinion is secured to the brake drums (10), (11) and the inner ones are secured with the shafts (12), (13) that are transmitting the movement to the sprocket drive wheels. If is applied the brake on the drum (11), the planetary gear pinion (7) it slows down its movement and at the same time through the satellites (4), (5) the speed of the shaft (13) is reduced and obviously of the corresponding tracks. The same is done to change the speed of the shaft (12) by applying this time the brake on the drum (10).

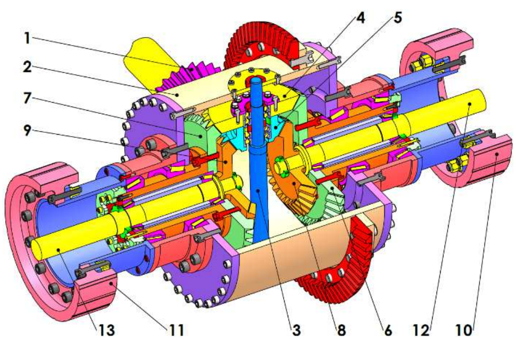

Fig. 5. Detail sectional view of the double conical differential

1- bevel gear; 2- satellites gearbox; 3- satellite carrier; 4- big satellite gear; 5- small satellite gear; 6,7- outer planetary gears pinion; 8,9- inner planetary gear pinion; 10,11- brake drums; 12,13- drive shafts.

3. DOUBLE DIFFERENTIAL KINEMATICS

To present and deduce the kinematics of the double differential, there are applied the following considerations underlying the determination of the angular velocities of the wheels that set the two tracks in motion. Thus, the following notations are made on figure 6:

- ω_{C2} - the angular velocity of satellites gearbox (2);
- $\omega_{4,5}$ - the angular velocity of the group of satellites (4) and (5);
- ω_{12} - the angular velocity of drive shaft and sprocket drive wheels (12);
- ω_{13} - the angular velocity of drive shaft and sprocket drive wheels (13);
- r_4 - radius of pitch circle of the big planetary gear pinion (4);
- r_5 - radius of pitch circle of the small planetary gear pinion (5);
- $r_{6,7}$ - radius of pitch circles of the outer planetary gear pinion (6), (7);
- $r_{8,9}$ - radius of pitch circles of the inner planetary gear pinion (8), (9).

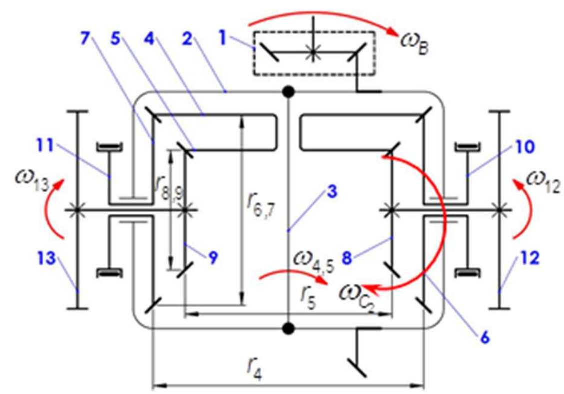

Fig. 6. Kinematic diagram of the steering and linear drive mechanism with double differential

1- bevel gear; 2- satellites gearbox; 3- satellite carrier; 4- big satellite gear; 5- small satellite gear; 6,7- outer planetary gear pinion; 8,9- inner planetary gear pinion; 10,11- brake drums; 12,13- output shaft with sprocket drive wheels.

In case of a right turn, by braking the drum (10), the common angular velocity of the outer planetary gear pinion (6) and the drum (10), considering the angular velocity from the engine ω_B, it can be written [4]:

$$\omega_{C2} \cdot r_{6,7} = \omega_B \cdot r_{6,7} + \omega_{4,5} \cdot r_4 \qquad (1)$$

For the external differential, consisting of elements (6), (4) and (7), it can be written:

$$\omega_{4,5} = \frac{\omega_{13} - \omega_{12}}{2} \cdot \frac{r_{8,9}}{r_5} \qquad (2)$$

But, if we consider the inner differential formed by the elements (8), (5) and (9), then:

$$\omega_{12} = 2 \cdot \omega_{C2} - \omega_{13}; \omega_{13} = 2 \cdot \omega_{C2} - \omega_{12} \qquad (3)$$

If $\omega_{4,5}$ is eliminated from expressions (1) and (2), it is obtained:

$$\omega_{13} = \omega_{12} + 2 \cdot \frac{r_{6,7}}{r_{8,9}} \cdot \frac{r_5}{r_4} \cdot (\omega_{C2} - \omega_B) = \omega_{12} + \frac{2}{i_{I_d}} \cdot (\omega_{C2} - \omega_B) \quad (4)$$

Where, $i_{I_d} = \frac{r_{8,9}}{r_{6,7}} \cdot \frac{r_4}{r_5}$ represents the differential constant, a constructive characteristic, different from zero [5].

Substituting ω_{12} and ω_{13} from (3) and (4), the expressions for the angular velocities of shaft with sprocket drive wheels (12) and (13) are obtained:

$$\begin{aligned} \omega_{12} &= \omega_{C_2} \cdot \left(1 - \frac{1}{i_{I_d}} \cdot \frac{\omega_{C_2} - \omega_B}{\omega_{C_2}}\right) \\ \omega_{13} &= \omega_{C_2} \cdot \left(1 + \frac{1}{i_{I_d}} \cdot \frac{\omega_{C_2} - \omega_B}{\omega_{C_2}}\right) \end{aligned} \quad (5)$$

To have small radius cornering it is necessary to immobilize the drum (10). So, if we consider $\omega_B = 0$, in the previous expressions, it results:

$$\begin{aligned} \omega_{12} &= \omega_{C_2} \cdot \left(1 - \frac{1}{i_{I_d}}\right) \\ \omega_{13} &= \omega_{C_2} \cdot \left(1 + \frac{1}{i_{I_d}}\right) \end{aligned} \quad (6)$$

Which leads to expression:

$$\frac{\omega_{13}}{\omega_{12}} = \frac{1 + i_{I_d}}{i_{I_d} - 1} \quad (7)$$

Considering expression (7) and expression (6), its ca be written:

$$\frac{\omega_{13}}{\omega_{12}} = \frac{1 + \frac{1}{i_{I_d}} \cdot \frac{\omega_{C_2} - \omega_B}{\omega_{C_2}}}{1 - \frac{1}{i_{I_d}} \cdot \frac{\omega_{C_2} - \omega_B}{\omega_{C_2}}} \quad (8)$$

When moving the excavator on a curve with radius *R*, see figure 7, each track will go through an arc of a circle, according to [5]:

$$\begin{aligned} S_s &= (R + 0,5 \cdot B) \cdot \alpha = r \cdot \omega_s \cdot t \\ S_d &= (R - 0,5 \cdot B) \cdot \alpha = r \cdot \omega_d \cdot t \end{aligned} \quad (1.9)$$

Where, *r* is the radius of sprocket drive wheels, and ω_s, ω_d are the angular velocities of the left and right tracks.

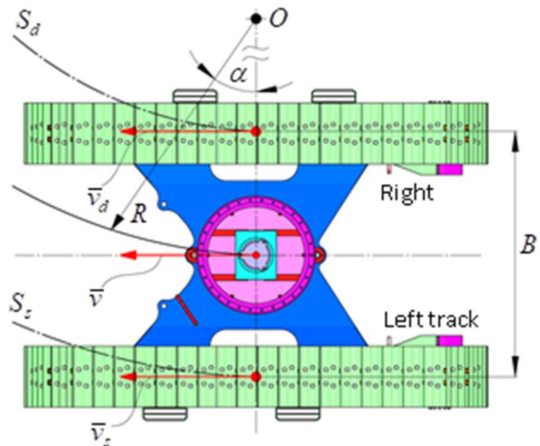

Fig. 7. Moving the excavator on a curve

In keeping with figure 7,and making the ratio between the two expressions (9), results:

$$\frac{\omega_s}{\omega_d} = \frac{R + 0,5 \cdot B}{R - 0,5 \cdot B} \quad (10)$$

Considering the expression (10), in which $\omega_s = \omega_{13}$, $\omega_d = \omega_{12}$, expression (8) can be rewritten as following:

$$\frac{\omega_{13}}{\omega_{12}} = \frac{1 + \frac{1}{i_{I_d}} \cdot \frac{\omega_{C_2} - \omega_B}{\omega_{C_2}}}{1 - \frac{1}{i_{I_d}} \cdot \frac{\omega_{C_2} - \omega_B}{\omega_{C_2}}} = \frac{\omega_s}{\omega_d} = \frac{R + 0,5 \cdot B}{R - 0,5 \cdot B} \quad (11)$$

Considering the expression (11), the minimum turning radius $R_{\min}$, is obtained when $\omega_B = 0$, as following:

$$RI_{d_{min}} \quad (12)$$

The angular velocity ω_B of the input shaft, depending on the turning radius, and will change from (11) to the following expression:

$$\omega_B = \omega_{C_2} \cdot \left(1 - i_{I_d} \cdot \frac{0,5 \cdot B}{R}\right) \quad (13)$$

4. DOUBLE DIFFERENTIAL DYNAMICS

For the dynamic study of the double conical differential presented in figure 6, it will be considered only the satellite gearbox as well as the satellite gears block [5]. According to figure 8, the effect of the half-shafts is replaced by the reactions R_d an R_s, on the teeth gear of the inner satellite (5).

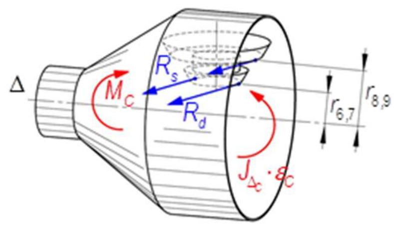

Fig. 8. Satellite gearbox and satellite gears block

Thus, in keeping with previous assumptions, and according to figure 8, the equation of dynamic equilibrium for the satellite box is:

$$M_C = (R_d + R_s) \cdot r_{8,9} + R_{dt} \cdot r_{6,7} + J_{\Delta_C} \cdot \varepsilon_C \qquad (14)$$

Where, $J_{\Delta_C} \cdot \varepsilon_C$ is the moment of resistance due to the forces of inertia acting on the cassette [6]. Next, it will be we studied the group made up of the brake drum and the planetary gear on the right, in solidarity with it, whose sketch can be found in figure 9.

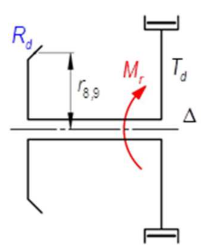

Fig. 9. Drum brake and right planetary gear

Starting from the premise that the movement of the drum is uniform, it results that the braking moment is [6]:

$$M_r = R_{dt} \cdot r_{6,7} \qquad (15)$$

Thus, considering each shaft separately, the equations of dynamic equilibrium of the two planetary half-shafts are:

$$\begin{aligned} R_s \cdot r_9 &= M_{ws} + J_{\Delta s} \cdot \varepsilon_s \\ R_d \cdot r_8 &= M_{wd} + J_{\Delta d} \cdot \varepsilon_d \end{aligned} \qquad (16)$$

Where the indices *s* and *d* refer to the left or right side, and M_{ws}, M_{wd} are the resistant moments of the semi-planetary shafts. Considering the group of satellite gears isolated, and removing the box, according to the figure 10, its effect is replaced by force F_C. The force with which the cassette acts on the axis of the satellites, and the effect of the planetary semi-shafts will be eliminated by the reactions R_d and R_s, and the effect of the drum braked by R_{dt}.

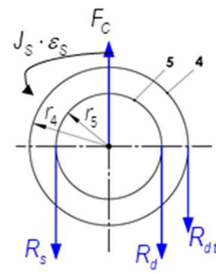

Fig. 10. Isolated satellite gears group

The equation of dynamic equilibrium corresponding to the group of satellites is:

$$(R_d - R_s) \cdot r_5 + R_{dt} \cdot r_4 - J_S \cdot \varepsilon_S = 0 \qquad (17)$$

or

$$(R_d - R_s) \cdot r_5 + M_r \cdot \frac{r_4}{r_{6,7}} - J_S \cdot \varepsilon_S = 0 \qquad (18)$$

Where, $J_S \cdot \varepsilon_S$ is the resistant momentum due to satellite inertia. In constant working process it can be considered $\varepsilon_C = \varepsilon_s = \varepsilon_d = \varepsilon_S = 0$, thus resulting from (17) the following expression:

$$\begin{aligned} M_C = R_d \cdot r_{8,9} + R_s \cdot r_{8,9} + R_{dt} \cdot r_{8,9} \\ \equiv M_{wd} + M_{ws} + M_r \end{aligned} \qquad (19)$$

and the expression (18) will become:

$$M_r = -(R_d - R_s) \cdot r_5 \cdot \frac{r_{6,7}}{r_4} \qquad (20)$$

The expression (19) represents the dynamic equation of the differential, which shows that the torque developed by the motor at the differential box must overcome the forward track resistance and the braking torque M_r. According to some mathematical transformations, the expression (20) becomes:

$$M_r = (M_{ws} - M_{wd}) \cdot \frac{1}{i_{I_d}} \quad (21)$$

and shows that the braking torque is equal to the difference in the torque applied to the planetary shafts, having the ratio: $1/i_{I_d}$.

7. CONCLUSION

The paper aims to present a constructive solution for moving excavators on tracks. Thus, a mechanical solution was designed and analyzed in which a double differential mechanism was used with bevel gears and hydraulically operated and controlled drum brakes. In terms of the performance of excavators in which the tracks are hydraulically operated, they are very reliable and can transmit high moments compared to the method approached in the work. Considering all these disadvantages and the advantage in the paper, we highlighted the kinematics and dynamics of the drive mechanism with double differential.

The angular velocities of the differential output shafts, noted in the paper with ω_{12} and ω_{13}, were established based on kinematic diagrams. It was also set according to the reduction ratio i_{I_d}, the angular velocity ω_{13}. The dynamic study includes the equation of the dynamic equilibrium, as well, the equation of the motor moment that will drive the mechanism, which must be greater than M_r.

8. REFERENCES

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CONSIDERAȚII PRIVIND MECANISMUL DE DEPLASARE DIRECȚIONALĂ ȘI LINIARĂ CU DIFERENȚIAL DUBLU PENTRU MAȘINI PE ȘENILE

Rezumat: *Această lucrare se focusează pe proiectarea şi modelarea unui sistem mecanic de realizare a virajelor destinat maşinilor pe şenile care va avea in componenta sa un mecanism de direcţie cu diferenţial dublu conic. Următorul obiectiv vizat în continuarea lucrării este orientat către modelarea matematică a sistemului mecanic propus care va conţine cinematica şi dinamica diferenţialului dublu în funcţie de mişcările pe care le realizează o astfel de maşină. Diferențialul dublu conic este un mecanism complex care utilizat în aplicații de mișcare a excavatoarelor, permite rotirea independentă a şenilelor, una faţă de alta. Astfel cele două şenile ale excavatorului vor avea posibilitatea să parcurgă distanțe diferite la viraje sau la deplasarea pe teren accidentat cu neregularități.*

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