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KINEMATIC LINKAGES IN THE HINGED UNDERCARRIAGE OF A MOBILE WORKING MACHINE

Summary. The main purpose of this article is to present a possible description of kinematic characteristics concerning the direction of travel of a selected mobile working machine in motion, namely, a hinged loader. This description is presented in the form of equations, which define the circumferential velocity as well as the angular velocity of the vehicle wheels during a ride involving a curve. In this way, it is also possible to describe a function of the axle differential.

Keywords: hinged loader, kinematic linkages, direction of travel, riding a curve, travel speed

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1. INTRODUCTION

Similar to other engineering entities, optimization of the design process and operation of mobile working machines requires the application of computer simulations based on dynamic models, which are created in relation to real objects.

The predominant characteristics of the dynamic model, which has to simulate the real machine, can be determined after the performed analysis of the concerned kinematic linkages in the given object. The detailed kinematic analysis is often constrained due to real software and hardware possibilities or limitations. Therefore, the overarching complexity of the individual design components of the real machine, including the undercarriage, working equipment and driving system, cannot be integrated into the created model as a whole [6,7].

Taking into consideration the above-mentioned conditions, it is also useful to describe the axle differential function during the vehicle ride involving a curve by means of kinematic linkages that support wheel rolling without slippage.

2. DRIVING IN THE DESIRED DIRECTION OF TRAVEL

Much of the worldwide production of loaders involves the wheeled undercarriage with the articulated framework, such that the both parts of the undercarriage are jointed by a couple of hinges [1,3].

The position of the vertical hinge enables mutual turning of the framework parts in the two-sided angular range of 35° to 45°, i.e., it is used to ride a curve.

The horizontal hinge is designed in order to eliminate bending and torsional moments, which are responsible for loading the framework of vehicle.

As both axles have the same wheel track and the same tyres, the rolling resistance is reduced.

The mechanism for driving in the desired direction of travel is specific to the hinged undercarriages, which are different from undercarriages of the other mobile working machines.

The following are known as driving systems that work with the desired direction of travel [5]:

- Manual driving systems
- Driving systems with a servo unit
- Machine driving (with an external source of power)

There are many variants in the construction of driving mechanisms, which are designed in line with the requirements concerning the precision of the vehicle positioning as well as the forces that result in loading the undercarriage [8,9]. Individual driving systems are described in [2] and [5].

The simplest principle consists of the application of two double-acting linear hydraulic motors (hydraulic cylinders) combined with a vertical hinge. The axles are fixed steadily to the individual machine sections.

In the case of direct (linear) travel, the hydraulic cylinder lengths are the same, while any change in the hydraulic cylinder lengths causes a mutual turning of both machine sections around the vertical hinge VK (Fig. 1). A crosswise interconnection of the working volumes between the hydraulic cylinders ensures the same reaction from the driving system when turning to the left or the right.

The mutual position of the machine sections is controlled by the feedback SV (Fig. 1). The feedback can be arranged using:

- mechanical linkage, which is applied to a leverage mechanism
- hydraulic linkage, which is mostly equipped with a measuring hydraulic generator

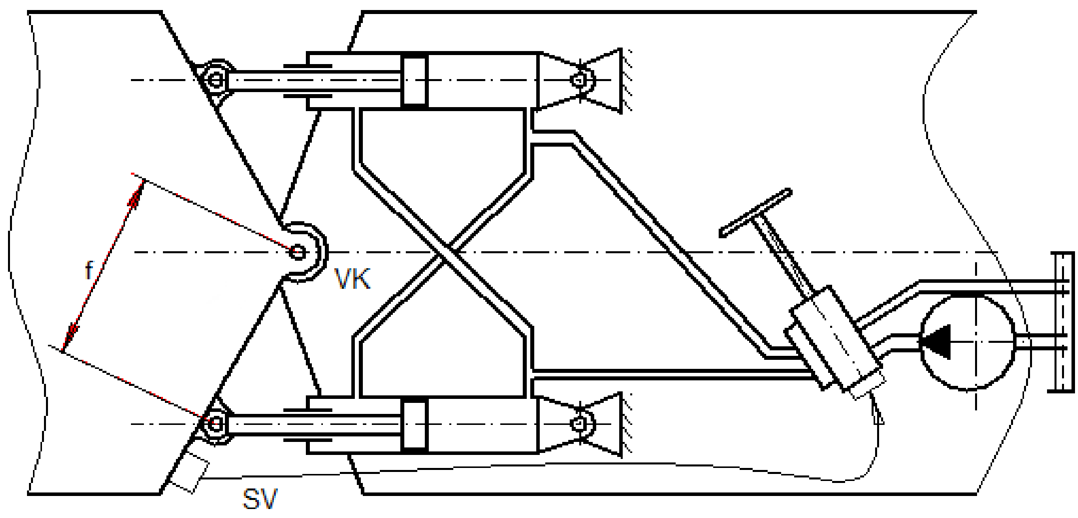

Fig. 1 Scheme of the driving mechanism arrangement

3. KINEMATIC LINKAGES OF TRAVEL WHEELS

Fig. 2 schematically illustrates the driving mechanism, which is designed to turn the hinged loader by means of linear hydraulic motors (hydraulic cylinders). The linear shifting of the pistons in the hydraulic cylinders (piston retracting or pushing out) causes a mutual turning of the front and rear parts of the wheel loader around the vertical hinge VK.

The scheme in Fig. 2 corresponds to the arrangement in Fig. 1. Meanwhile, Fig. 2 is in effect a simplified form of Fig. 1.

Curved travel involves the motion of the vehicle on a curved trajectory around the instantaneous centre of the wheel loader rotation O1. The point O1 is a cross-point of the prolonged axles, according to Fig. 3.

Thus, vehicle turning is caused by a mutual turning of the front and rear parts of the wheel loader around the vertical hinge VK.

In a case involving the constant travelling speed v_p, the circumferential speed of the wheels on both sides of the vehicle will be different during curved travel.

If the circumferential speed of the wheel, which is situated on the internal side of the curve, is signed v_{pP}, then the circumferential speed of the wheel rotating on the external side of the curve is signed v_{pL}. It is evident that $v_{pL} \rangle v_{pP}$.

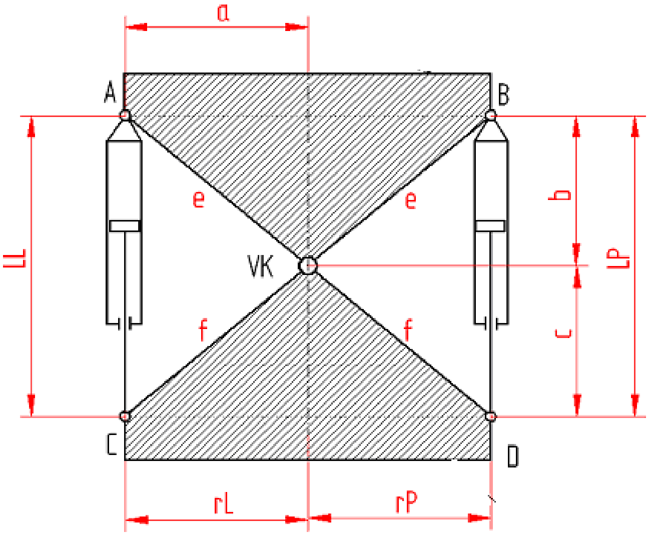

Fig. 2 Scheme of geometric dimensions of a driving mechanism that supports the desired direction of travel

A, B – positions of turning points for cylinders in hydraulic motors; C, D – positions of turning points for piston pins of hydraulic motors; LL, LP – length of the left and the right hydraulic motor; VK – vertical hinge

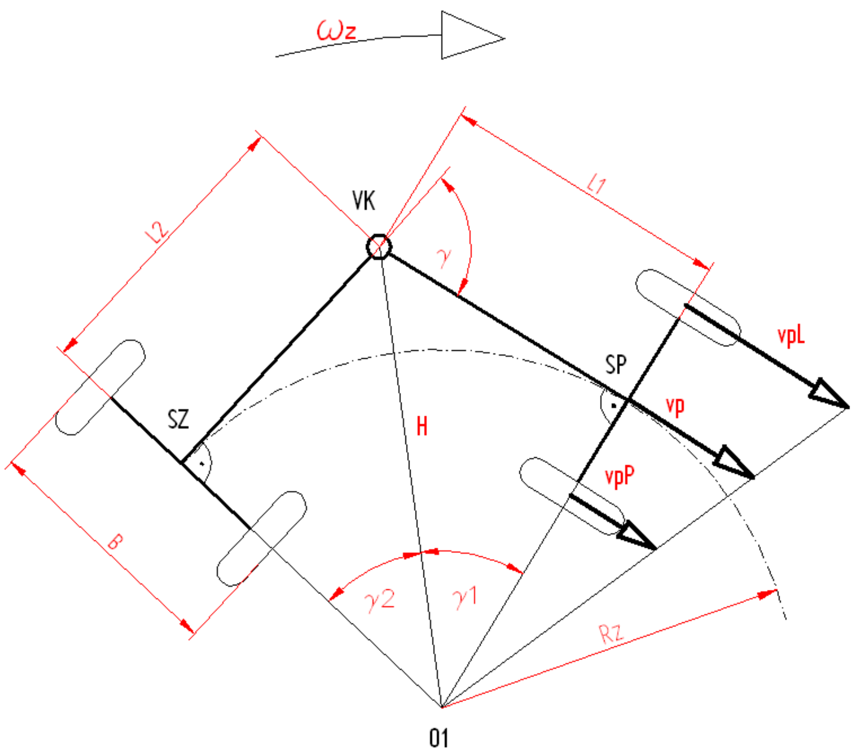

Fig. 3 Curved travel, horizontal projection
O1 – instantaneous centre of the wheel loader rotation; SP – centre of the front axle; SZ – centre of the rear axle; VK – vertical hinge; L1, L2 – distances of axles from the vertical hinge; B – wheel track; R_z – instantaneous turning radius; ω_z – angular speed relating to the centre of rotation

The travel speed of the loader can be expressed as follows:

$$v_P = \omega_z R_z , \tag{1}$$

where ω_z is the angular speed relating to the centre of rotation, while R_z is the instantaneous turning radius.

The circumferential speed v_{pP} of the wheel, which is situated on the internal side of curve, can then be written in the following form:

$$v_{pP} = \omega_z . \left(R_z - \frac{B}{2} \right) \quad , \tag{2}$$

which means that the circumferential speed v_{pL} of the wheel on the external side of curve will be:

$$v_{pL} = \omega_z . \left(R_z + \frac{B}{2} \right) \; . \tag{3}$$

In the case of the cranked loader configuration, the turning angle between the front and rear parts of the vehicle is γ (Fig. 3), whereas:

$$\gamma = \gamma_1 + \gamma_2 \; , \tag{4}$$

where γ_1 is the turning angle of the loader front part, while γ_2 is the turning angle at the rear of the loader.

According to the right-angled triangles Δ(O1,SP,VK) and Δ(O1,SZ,VK), the length of the hypotenuse H equates to $H = \dfrac{L_1}{\sin \gamma_1}$ as well as $H = \dfrac{L_2}{\sin \gamma_2}$ (see Fig. 3).

By comparing both relations, which are valid for the value H of the hypotenuse length, the following equation is obtained:

$$\frac{L_1}{\sin \gamma_1} = \frac{L_2}{\sin \gamma_2} \, . \tag{5}$$

From Equation (5),

$$\sin \gamma_2 = \frac{L_2}{L_1} \sin \gamma_1 \, . \tag{6}$$

If $\gamma = \gamma_1 + \gamma_2$, then $\gamma_2 = \gamma - \gamma_1$ and Equation (6) is modified as follows:

$$\sin(\gamma - \gamma_1) = \frac{L_2}{L_1} \sin \gamma_1 . \tag{7}$$

Using the goniometric rule, $\sin(\alpha - \beta) = \sin \alpha \cos \beta - \cos \alpha \sin \beta$, Relation (7) can be rewritten as follows:

$$\sin \gamma \cos \gamma_1 - \cos \gamma \sin \gamma_1 = \frac{L_2}{L_1} \sin \gamma_1 . \tag{8}$$

After an adjustment of Equation (8),

$$\frac{\sin \gamma}{tg\gamma_1} - \cos \gamma = \frac{L_2}{L_1} . \tag{9}$$

The relation, which is valid for $tg\gamma_1$, is obtained from (9) above:

$$tg\gamma_1 = \frac{\sin \gamma}{\frac{L_2}{L_1} + \cos \gamma} . \tag{10}$$

According to Fig. 3, it simultaneously follows from triangle Δ(O1,SP,VK) that:

$$tg\gamma_1 = \frac{L_1}{R_z} \tag{11}$$

By comparison the left sides from Equations (10) and (11), the following relation is obtained:

$$\frac{L_1}{R_z} = \frac{\sin \gamma}{\frac{L_2}{L_1} + \cos \gamma} \tag{12}$$

From Equation (12), the value of the instantaneous turning radius R_z can be determined:

$$R_z = \frac{L_2 + L_1 \cos \gamma}{\sin \gamma} . \tag{13}$$

From Equation (1), the following is obtained:

$$\omega_z = \frac{v_p}{R_z} . \tag{14}$$

The circumferential speed v_{pP} of the wheel, which is situated on the internal side of the curve, as well as the circumferential speed v_{pL} of the wheel on the external side, can be obtained after applying Relation (14) to Equations (2) and (3), as follows:

$$v_{pP} = v_p \left(1 - \frac{B.\sin\gamma}{2.(L_2 + L_1 \cos\gamma)} \right) \tag{15}$$

$$v_{pL} = v_p \left(1 + \frac{B.\sin\gamma}{2.(L_2 + L_1 \cos\gamma)} \right). \tag{16}$$

At the same time, it is valid that:

$$v_{pP} = r_k \ \omega_{pP} \quad \text{and} \quad v_{pL} = r_k \ \omega_{pL},$$

where r_k is the dynamic wheel radius.

Thus, the values of the angular speed ω_{pP} of the wheel, which is situated on the internal side, and the angular speed ω_{pL} of the external side wheel are as follows:

$$\omega_{pP} = \frac{v_p}{r_k} \left(1 - \frac{B.\sin\gamma}{2.(L_2 + L_1 \cos\gamma)} \right) \ , \tag{17}$$

$$\omega_{pL} = \frac{v_p}{r_k} \left(1 + \frac{B.\sin\gamma}{2.(L_2 + L_1 \cos\gamma)} \right) \ . \tag{18}$$

According to Equations (17) and (18), a significant difference between the angular speed values of the wheels rotating on the both sides of axle is evident.

This fact corresponds to the function of the axle differential of the given wheel loader, whereas Relations (17) and (18) represent the analytical description of its function.

4. CONCLUSION

This paper presents a kinematic analysis of driving in the desired direction of travel in the case of a hinged wheel loader.

The analysis of kinematic relations, which occur during the driving process, involves a basic assumption, which is necessary for the creation of the resulting dynamic model of the wheeled undercarriage for a mobile working machine [4].

Equations (17) and (18) represent the final result of the performed kinematic analysis of the loader travel involving a curve. These equations are useful for the next stage of vehicle travel simulation.

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