RESEARCH ON DYNAMIC TURNING PROCESS OF TRACTORS TRAILERS AGGREGATE WITH MOUNTED DAMPERS AGAINST SUDDEN BENDING

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Abstract: *This paper introduces cycle dynamic model of combined tractors - trailer with mounted damper against sudden bend in order to increase stability when turning round. The research results can make rational basis for selection of appropriate parameters of the damper.*

Keywords: TURNAROUND, DYNAMIC MODEL, DAMPER.

1. Introduction

During rotation of the tractor - trailer aggregate (TTA), there is sometimes the case that the trailer pushs the tractor. Thrust of trailers has the effect of turning tractor in the same direction with the rotating direction, increasing the relative rotation between the tractor and the trailer, it means that TTA will be self- folding its body. This leads to an instability TTA's motion in rotation [1], particularly dangerous when turnaround times, which can lead to loss of control.

In order to limit the sudden bending of TTA, the authors have proposed mounting a hydraulic damper at joints (see Figure 1) to increase safety while rotating motion.

2. Research Methodology

Using the basic equations of solid dynamics, deflective motion features of elastic wheel, the methods of solving nonlinear equations.

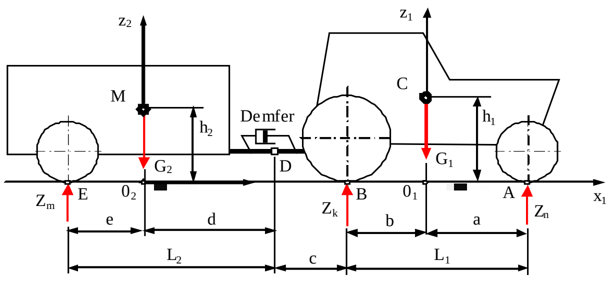

Figure 1. *TTA tractor diagram in vertical plane*

The model parameters are denoted as follows :

δ_1, δ_2, δ_3 – deflection motion angle of the tractor's wheel and trailers;

V_{x1}, V_{x2} – axial velocity components of tractors and trailers;

β – rotation angle of the steering wheel of the tractor.

α_D – the angle between the velocity of the juncture of the vertical axis hooks and trailers;

δ_D - the angle between the velocity of mechanical joints and vertical axis of the tractor ;

θ - the angle between the longitudinal axis of the tractor and trailer's longitudinal axis ;

φ_1, φ_2 – rotation angles of traktor and trailer.

3. Research Results and Discussion

*3.1. The revolving dynamic model TT*A

To simplify in building revolving dynamic model without loss of generality of the problem, in this paper we accept some of the following assumptions:

- The flat line;

- Axial velocity of the tractor is not changed;

- The focus of tractors and trailers distributed along the plane of symmetry of it and ignore the weight distribution on the bridge.

With the above assumptions, giving the revolving dynamic model TTA with a trailer to flat model, a trace, shown in Figure 1 with the coordinate system: absolute coordinate system associated with road base OX0Y0Z0; relative coordinate system attached to the tractor CX1Y1Z1 and relative coordinates attached to the trailers MX2Y2Z2., where points C and M are centers of garvity of traktor and trailer.

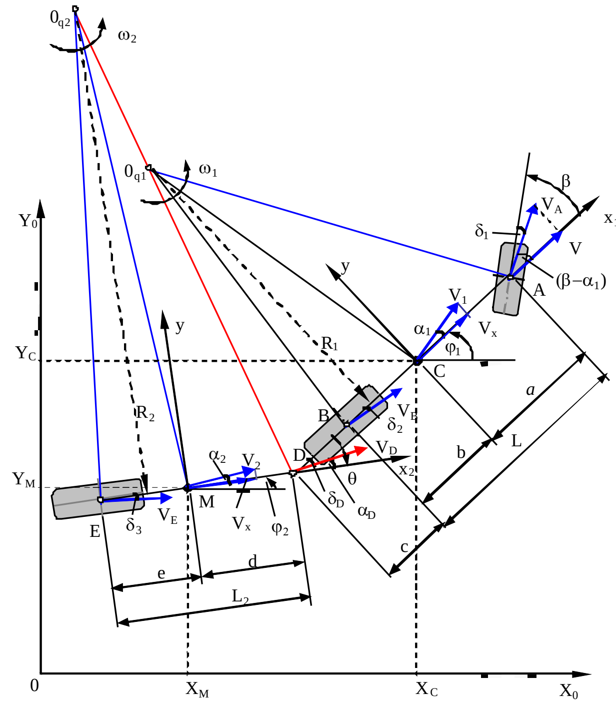

Figure 2. *TTA kinetic diagram in the horizontal plane*

The speed vector of the tractor consists 2 components: the axial component (in X_1) V_{cx} and side-oriented component (in Y_1) V_{cy} . V_{cy} is calculated by the formula:

$$V_{Cy} = \frac{V_{x1}}{L_1}\left[btg(\beta - \delta_1) - a.tg\delta_2\right] \tag{1}$$

Conversion of velocity vector of tractor's center of gravity from relative coordinate system $CX_1Y_1Z_1$ to the absolute coordinate system $OX_0Y_0Z_0$, then differentiate in time we obtain the absolute acceleration of the center of gravity. Project absolute acceleration vector to the axes X_1 and Y_1 we obtain the components of absolute acceleration in the relative coordinates of the tractor $CX_1Y_1Z_1$.

$$\begin{cases} a_{Cx} = \dot{V}_{x1} - \dfrac{V_{x1}}{L_1}\omega_1\left[b.tg(\beta-\delta_1) - a.tg\delta_2\right] \\ a_{Cy} = \dfrac{\dot{V}_{x1}}{L_1}\left[b.tg(\beta-\delta_1) - a.tg\delta_2\right] + \\ +\dfrac{V_{x1}}{L_1}\left\{L_1\omega_1 + \left[\dfrac{b}{\cos^2(\beta-\delta_1)}(\dot{\beta}-\dot{\delta}_1) - \dfrac{a}{\cos^2\delta_2}.\dot{\delta}_2\right]\right\} \end{cases} \quad (2)$$

Rotation speed of the tractor is calculated using the formula:

$$\omega_1 = \frac{V_{x1}}{L_1}\left[tg(\beta-\delta_1) + tg\delta_2\right] \quad (3)$$

Then angular acceleration determined by the formula:

$$\begin{aligned} \dot{\omega}_1 = &\frac{\dot{V}_{x1}}{L_1}\left[tg(\beta-\delta_1) + tg\delta_2\right] + \\ &+\frac{V_{x1}}{L_1}\left\{\left[1+tg^2(\beta-\delta_1)\right](\dot{\beta}-\dot{\delta}_1) + (1+tg^2\delta_2)\dot{\delta}_2\right\} \end{aligned} \quad (4)$$

Similarly, we define the absolute velocity components for trailers.

Combines the formulas for the tractor and trailer, we obtain

$$\begin{cases} a_{Cx} = \dot{V}_{x1} - \dfrac{V_{x1}}{L}\omega_1\left[b.tg(\beta-\delta_1) - a.tg\delta_2\right] \\ a_{Cy} = \dfrac{\dot{V}_{x1}}{L_1}\left[b.tg(\beta-\delta_1) - a.tg\delta_2\right] + \\ +\dfrac{V_{x1}}{L_1}\left\{L\omega_1 + \left[\dfrac{b}{\cos^2(\beta-\delta_1)}(\dot{\beta}-\dot{\delta}_1) - \dfrac{a}{\cos^2\delta_2}.\dot{\delta}_2\right]\right\} \\ \dot{\omega}_1 = \dfrac{\dot{V}_{x1}}{L}\left[tg(\beta-\delta_1) + tg\delta_2\right] + \\ +\dfrac{V_{x1}}{L_1}\left\{\left[1+tg^2(\beta-\delta_1)\right](\dot{\beta}-\dot{\delta}_1) + (1+tg^2\delta_2)\dot{\delta}_2\right\} \\ a_{Mx} = \dot{V}_{x2} - \dfrac{V_{x2}}{L_2}\omega_2\left[e.tg(\theta-\delta_D) - d.tg\delta_3\right] \\ a_{My} = \dfrac{\dot{V}_{x2}}{L_2}\left[e.tg(\theta-\delta_D) - d.tg\delta_3\right] + \\ +\dfrac{V_{x2}}{L_2}\left\{L_2\omega_2 + \left[\dfrac{e}{\cos^2(\theta-\delta_D)}(\dot{\theta}-\dot{\delta}_D) - \dfrac{d}{\cos^2\delta_3}.\dot{\delta}_3\right]\right\} \\ \dot{\omega}_2 = \dfrac{\dot{V}_{x2}}{L_2}\left[tg(\theta-\delta_D) + tg\delta_3\right] + \\ +\dfrac{V_{x2}}{L_2}\left\{\left[1+tg^2(\theta-\delta_D)\right](\dot{\theta}-\dot{\delta}_D) + (1+tg^2\delta_3)\dot{\delta}_3\right\} \end{cases} \quad (5)$$

The kinetic link

The binding kinetics at joints D, here is the hinge to the speed of the tractor and trailers are equal. On that basis, we define:

$$\begin{cases} V_{x2} = V_{x1}\cos\theta + \dfrac{V_{x1}}{L_1}\left[(L_1+c)tg\delta_2 + c.tg(\beta-\delta_1)\right]\sin\theta \\ \delta_D = arctg\dfrac{(L_1+c)tg\delta_2 + c.tg(\beta-\delta_1)}{L_1} \end{cases} \quad (6)$$

Taking the derivative in time we will get the acceleration components:

$$\begin{aligned} \dot{V}_{x2} = &\dot{V}_{x1}.\cos\theta - V_{x1}.\dot{\theta}\sin\theta + \frac{\dot{V}_{x1}}{L_1}\begin{bmatrix}(L_1+c)tg\delta_2 + \\ c.tg(\beta-\delta_1)\end{bmatrix}\sin\theta + \\ &+\frac{V_{x1}}{L_1}\left\{\begin{matrix}(L_1+c)(1+tg^2\delta_2)\dot{\delta}_2 + \\ c.\left[1+tg^2(\beta-\delta_1)\right](\dot{\beta}-\dot{\delta}_1)\end{matrix}\right\}\sin\theta + \\ &+\frac{V_{x1}}{L_1}\left[(L_1+c)tg\delta_2 + c.tg(\beta-\delta_1)\right]\dot{\theta}\cos\theta; \end{aligned}$$

$$\dot{\delta}_D = \frac{(L_1+c)(1+tg^2\delta_2)\dot{\delta}_2 + c.(1+tg^2(\beta-\delta_1)(\dot{\beta}-\dot{\delta}_1)}{L_1\left\{1+\left[\dfrac{(L+c)tg\delta_2 + c.tg(\beta-\delta_1)}{L_1}\right]^2\right\}}$$

$$\begin{aligned} \dot{\theta} = \omega_1 - \omega_2 = &\frac{V_{x1}}{L_1}\left[tg(\beta-\delta_1) + tg\delta_2\right] - \\ &-\frac{V_{x2}}{L_2}\left[tg(\theta-\delta_D) + tg\delta_3\right] \end{aligned} \quad (7)$$

3.2. Dynamics of TTA

The forces on the tractor:

- Weight of tractor G_t.
- Forces in the coupling point: F_{Dx}; F_{Dy}; F_{Dz}
- The normal (radial) reaction forces of the wheels R_{z1}, R_{z2};
- The lateral reaction forces of the front and rear tires Ry_1, Ry_2
- Longitudinal reaction forces of tire Rx_1, Rx_2;

External forces acting on the trailer are:

- Weight of the trailer GM.
- Forces in the coupling point: F_{Dx2}; F_{Dy2}; F_{Dz2}
- The normal (radial) reaction force of the wheels Rz_3;
- The lateral reaction force Ry_3;
- Longitudinal reaction force of tire Rx_3;

The cross-reactions of the front and rear tires of tractor R_{y1} , R_{y2} và R_{y3} of trailer depend on sideslip angle δ_i of wheel, can be calculated: [2,3].

Where: k_1, k_2 and k_3 is cornering stiffness of the wheel correspond with the front axle, rear axle and trailer. The coefficients k_i with nonlinear model itself depends on the deflection angle δ_i.

Damper:

Dampers have to achieve the requirements to create torque at small θ angle with the aim not obstruct the driving in normal conditions, but in the large θ angle to create great moment to prevent sudden bending of TTA. With the above requirements, author suggests the working principle diagram of a hydraulic damper as shown in Figure 3.

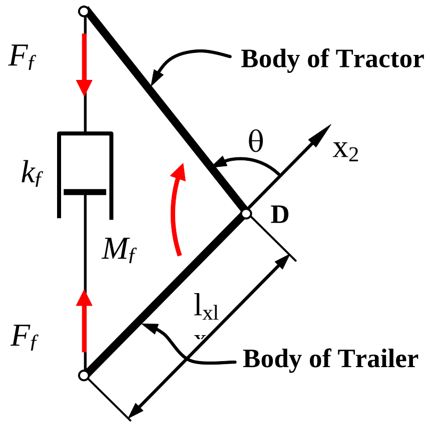

Figure 3. *Principle diagram of the damper*

Damping moment Mdemf is placed at the point of the hook, generating by damping force Fdmf of the hydraulic cylinder and is calculated by the formula:

$$M_f = k_f * \left(l_{xl}.\sin(\theta/2)\right)^2 .\dot{\theta}\,, \qquad (9)$$

Where k_f e damping coefficient of the cylinder;

l_{xl} – hydraulic cylinder arm

The force component links hook points at F_{Dx2} ,F_{Dy2} và F_{Dz2}, in the coordinate system of the trailer, which is determined by the formula:

$$\begin{aligned} F_{Dx2} &= -F_{Dx}.\cos\theta + F_{Dy}.\sin\theta \\ F_{Dy2} &= -F_{Dx}.\sin\theta - F_{Dy}.\cos\theta \\ F_{Dz2} &= -F_{Dz} \end{aligned} \qquad (10)$$

Apply Dalamber principles for mechanical system, we obtain system of differential equations:

$$\begin{cases} \dot{V}_{x1} - \dfrac{V_{x1}}{L_1}\omega_1\left[b.tg(\beta-\delta_1) - a.tg\delta_2\right] = \dfrac{F_{Dx} + F_k - f.(R_{z1}cos\beta + R_{z2}) - k_1\delta_1.\sin\beta + G_{Tx}}{m_T} \\ \dfrac{\dot{V}_{x1}}{L_1}\left[b.tg(\beta-\delta_1) - a.tg\delta_2\right] + \dfrac{V_{x1}}{L_1}\left\{L_1\omega_1 + \left[\dfrac{b}{\cos^2(\beta-\delta_1)}(\dot{\beta}-\dot{\delta}_1) - \dfrac{a}{\cos^2\delta_2}.\dot{\delta}_2\right]\right\} = \\ \qquad \dfrac{F_{Dy} - F_{f1}\sin\beta + k_1\delta_1\cos\beta + k_2\delta_2 + G_{Ty}}{m_T} \\ \dfrac{\dot{V}_{x1}}{L_1}\left[tg(\beta-\delta_1) + tg\delta_2\right] + \dfrac{V_{x1}}{L_1}\left\{\left[1 + tg^2(\beta-\delta_1)\right](\dot{\beta}-\dot{\delta}_1) + (1 + tg^2\delta_2)\dot{\delta}_2\right\} = \\ \qquad \dfrac{-F_{f1}a\sin\beta + k_1 a\delta_1\cos\beta - k_2 b\delta_2 - F_{Dy}*(b+c) + M_{demf}}{J_{yT}} \\ \dot{V}_{x2} - \dfrac{V_{x2}}{L_2}\omega_2\left[e.tg(\theta-\delta_D) - d.tg\delta_3\right] = \dfrac{-F_{Dx}*\cos\theta + F_{Dy}*\sin\theta - f*R_{z3} + G_{Mx}}{m_M} \\ \dfrac{\dot{V}_{x2}}{L_2}\left[e.tg(\theta-\delta_D) - d.tg\delta_3\right] + \dfrac{V_{x2}}{L_2}\left\{L_2\omega_2 + \left[\dfrac{e}{\cos^2(\theta-\delta_D)}(\dot{\theta}-\dot{\delta}_D) - \dfrac{d}{\cos^2\delta_3}.\dot{\delta}_3\right]\right\} = \\ \qquad \dfrac{-F_{Dx}*\sin\theta - F_{Dy}*\cos\theta + k_3\delta_3 + G_{My}}{m_M} \\ \dfrac{\dot{V}_{x2}}{L_2}\left[tg(\theta-\delta_D) + tg\delta_3\right] + \dfrac{V_{x2}}{L_2}\left\{\left[1 + tg^2(\theta-\delta_D)\right]\left\{\dot{\theta}-\dot{\delta}_D\right\} + (1 + tg^2\delta_3)\dot{\delta}_3\right\} = \\ \qquad \dfrac{(-F_{Dx}*\sin\theta - F_{Dy}*\cos\theta).d - e.k_3\delta_3 - M_{demf}}{J_{yM}} \end{cases}$$

Where J_{yT}, J_{yM} is the moment of inertia of the tractor and trailer in the vertical axis passing through the center of it.

$\dot{\delta}_3$, $\dot{\theta}$, $\dot{\delta}_D$, F_{Dx}, F_{Dy}. From the values of $\dot{V}_{x1}$, $\dot{V}_{x2}$, $\dot{\delta}_1$, $\dot{\delta}_2$, $\dot{\delta}_3$, $\dot{\theta}$, $\dot{\delta}_D$ can be solved to find the value of them in time by Runge-Kutta and it's modificated method. Based on the basis, we can examine the effects of some structural factors and the mode used to target of TTA dynamics when turnaround.

3.3. Some illustrative results

With the purpose of examining the effects of damping to the basic parameters of the TTA when turnaround, we have developed the program to solve the problem and applied for KUBOTA 18L tractor pulling a shafting trailer, mounted with hydraulic dempfer absorbers against sudden frame folding with dempfer arm l_{xl}=0,4 m;

Rotaring angle of front axle is assumed to be sin function:

* When TTA entering the corner:

for $0 \le t \le t_{max}$

$$\beta = \frac{\beta_{max}}{2} + \frac{1}{2}\beta_{max} * \sin\left(\pi.\frac{t}{t_{max}} - \frac{\pi}{2}\right)$$

$$\dot{\beta} = \frac{\pi}{2 * t_{max}} * \beta_{max} * \cos\left(\pi.\frac{t}{t_{max}} - \frac{\pi}{2}\right)$$

for $t_{max} < t < t_2$ $\qquad \beta = \beta_{max}$; $\quad \dot{\beta} = 0$

* When TTA exiting the corner:

for $t_2 \le t \le t_{max} + t_2$

$$\beta = \frac{\beta_{max}}{2} - \frac{1}{2}\beta_{max} * \sin\left(\pi.\frac{t - t_2}{t_{max}} - \frac{\pi}{2}\right)$$

$$\dot{\beta} = -\frac{\pi}{2 * t_{max}} * \beta_{max} * \cos\left(\pi.\frac{t - t_2}{t_{max}} - \frac{\pi}{2}\right)$$

for $t > t_2 + t_{max}$ $\qquad \beta = 0$; $\quad \dot{\beta} = 0$

Where t_{max} =4 time to drive wheel; β_{max} =10^0 - maximum rotation angle of the steering wheel; t_2 = 20s - time when exiting the corner.

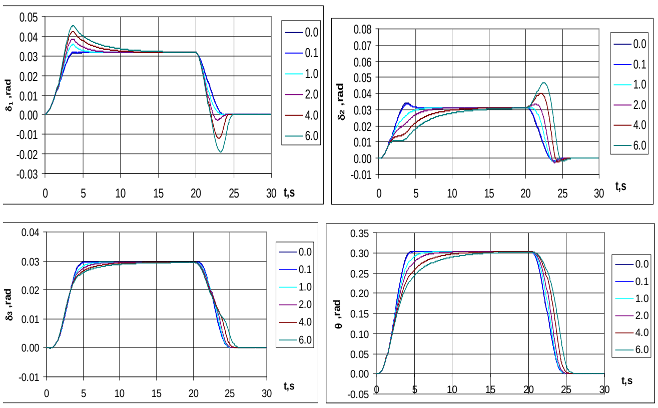

Figure 4. *Changes result of the basic parameters in time of the TTA pulling Kubota trailer with i_T = 30; V_{X1} = 3.53 m / s; k_f= 0-6 MNS / m*

In Figure 4 and 5 are graphs showing changing process of δ deviation moving angle of the wheel TTA and θ folding angle between tractors and trailers when changing the damping coefficient kf = 0-6 MNS/m with 2 transmission ratio i_t = 30 and it = 20, corresponding to the initial velocity of the tractor v_{1x} = *3,53 m/s* and v_{1x} = *5,28 m/s.*

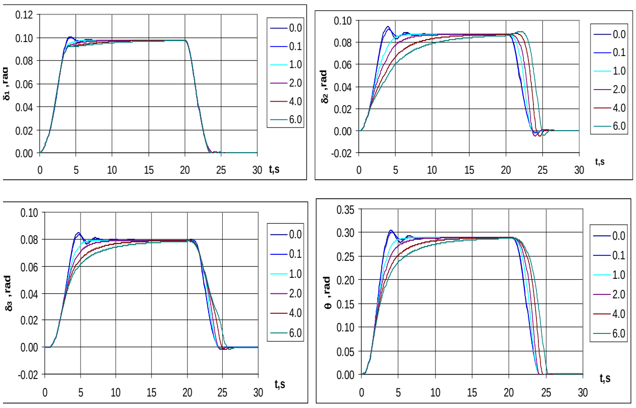

***Figure 5**. Changes result of the basic parameters in time of the TTA pulling Kubota trailer with it = 20; VX1 = 5.28 m / s; kf = 0-6 Nm.s/m*

The results showed that:

- Dampfer in some case has improved pretty well the stability of tractor when cornering. However in all cases when exiting corner, it absorber adversely affects the stability of TTA features. This suggests that, damper have to be one-side working.

- When increasing the damping coefficient $k_{f,}$ in the process of increased rotation β=0 to $\beta= \beta_{max,}$ the angles of deviation motion of δ_1, δ_2 , δ_3 and folding angles are more likely downward. In contrast, in the process turning back steering, β angle down from β_{max} to 0, then δ_1, δ_2 , δ_3 angles and θ increase relatively with increasing damping coefficient k_f.

- When changing the speed of movement of the tractor $V_{1x,}$ then the angle δ_1, δ_2, δ_3 and θ and also vary with covariates.

4. Conclusion

1 - Model has shown quite sufficient the relationship between kinetics and revolving dynamics of combined tractor - trailer fitted with hydraulic dampers against sudden bending, allowing to survey effects of structural elements and properties used to control rotation of the TTA.

2 - Damping coefficient can significantly influence the folding angulation of TTA and therefore also affect the angle of deflection motion of the wheels, when the conditions in steering angle rotation the same. This reflects the effects of hydraulic shock absorber, increasing the safe in the sudden turnaround.

3 - The graph of the change of folding angle and deviation movement angle can be used as a basis for choosing appropriate damping coefficient for a specific combined tractor.

5. References

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