VEHICLE STABILITY STUDY ON CURVED TRAJECTORY

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Situaţia deplasării în curbă constituie un subiect de studiu intens din cauza riscului mai ridicat pe care acest tip de mişcare îl prezintă. Mişcarea în curbă este determinată strict de interacţiunea dintre calea de rulare şi pneurile autovehiculului, mai exact de forţele care iau naştere în momentul în care conducătorul auto roteşte volanul. În această lucrare se prezintă un model matematic pe baza căruia se studiază influenţa diferiţilor parametrii (unghi bracare, viteză) asupra stabilităţii autovehiculului în curbă. Din studiu, reiese faptul că, la viteze mai mari de 80 km/h, unghiuri de bracare relativ mici pot destabiliza autovehiculul.

A subject of comprehensive experimental study is the movement on a curved path, due to the higher danger in such a motion. Moving on a curved trajectory is strictly determined by the interaction between the rolling surface and the tyres of the vehicle. More precisely, by the forces occurring exactly the moment the driver turns the steering wheel. This thesis attempts to show a mathematical model as the basis for the study of the influence of the different input values (velocity, friction) on the stability of the vehicle. The study reveals the fact that at more than 80 km/h relatively small brackage angles can destabilize the vehicle.

Keywords: stability, control, speed, brackage angle

1. Introduction

While moving in a curve, forces appear at the contact surfaces between each wheel and the road, and they maintain the vehicle on the track. Under these forces, the tyres are deformed and the velocity on each wheel is deviated from the wheel plane under a certain angle, depending on the lateral rigidity of the tyre and force magnitude.

To consider the longitudinal forces with two degrees of freedom allows a precise description of the vehicle movement in the rolling plane, X-Y. A model

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with three degrees of freedom can be described using the simplified representation for two wheels as well as for four wheels.

The movement on a curved trajectory has been treated in numerous papers of the domain, among which [5] where a three freedom degrees model is considered, subjected to the wheel forces given by the “magic formula” of H.B. Pacejka, [8] from the same conference where a two freedom degrees model is studied and the tire model proposed by Dugoff modified by the authors. In the article [4] where a three freedom degrees model is presented on a self made tire model . In [7] the authors have studied the wheel dynamics on a curved path and in straight line and the result obtained is very usefull to establish the sideslip angles.

The vehicle stability is studied in [6] using coefficients to correct the forces that act upon each wheel.

In [2] a common bicycle model is made equivalent to a model in which instead of the four forces on each wheel there are considered two forces with a sliding point of application along the symmetry axes of the vehicle.

Among the many articles it is worth mentioned [1] in which the authors define a way to control the sideslip of the vehicle starting from a two freedom degree model .This method is used together with the control of the spinning speed.

Last but not least, article[3] presents a study of the vehicle stability using a nonlinear model with a pear of forces on each wheel but the same slip angle for the front and for the back wheels.

The study model of an autovehicle is made from a rectangular shape body wich imagines the autochasis with everything on it and four wheels. This model has four freedom degrees which means that allows four types of independent movements: two translatory movements along the longitudinal and transversal axes and two rotations around the vertical and longitudinal axes respectively.

There are studied models at which the rotation around the longitudinal axe is neglected thus the model will contain only two wheels, one on the front and one on the rear, they being subjected to the resultant front and rear forces respectively. In this case, the vehicle body becomes a thin line which connects the two wheels.

This paper debates the movement on curved trajectory using a three degree of freedom autovehicle with two wheels, where the rotation around the vertical axe is taken into consideration, being an essential movement for the stability study. The three degree of freedom and four wheels vehicle models come closer to the reality but, in order to achieve the goal of the study, the more complicated mathematical calculations don’t justify the use of such a model confunted with precision gain.

The purpose of the paper is to identify the critical situations in wich the vehicle might find its self during the studied motion.

2. The mathematical model

As in Fig.1 presented, in the three freedom degrees model the longitudinal forces were included on the front axis, F_{xf} respectively, F_{xs}, corresponding to the wheel forces F_{x1} and F_{x2}, F_{x3} and F_{x4}. The direction of the longitudinal forces corresponds to a traction moment, while by breaking the direction changes by 180°.

The case to be studied of acceleration curved motion when the power bridge is in front.

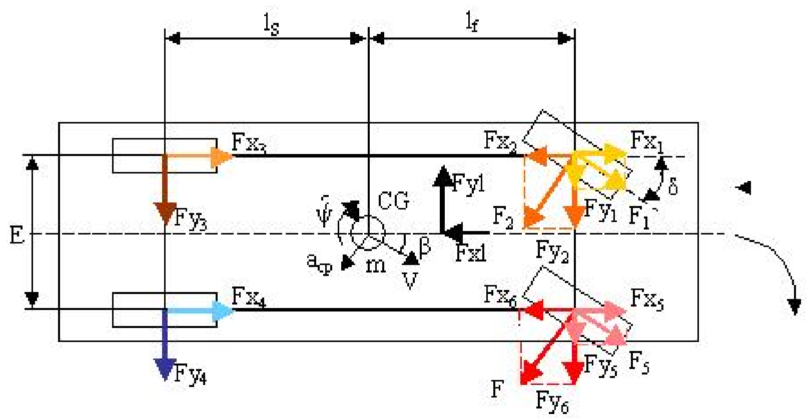

Fig.1 The model of the vehicle with three freedom degrees used for the study of the longitudinal, tranverse and gyratory dynamics

In the Fig.1 the following elements appear :

C.G. – gravity center;

E – gauge (the distance between the wheels of the axis);

l_f – distance between C.G. and the front axis;

l_s – distance between C.G. and the rear axis;

m – mass of the vehicle;

$\dot{\psi}$ - angular rotation velocity;

β – deviation angle of the vehicle;

δ – the turning angle of the front wheels;

F_{xi} şi F_{yi} are friction forces between tyres and the road;

F_{xi} – longitudinal forces (forces in the wheel plane);

F_{yi} – lateral forces oriented towards the curve center (perpendicular forces on the wheel).

$\dot{V}$ - tangent acceleration;

$a_{cp} = \frac{V^2}{\rho}$ - center-oriented acceleration.

Similar to the two freedom degree model, a linear model can be introduced also in the longitudinal dynamics, in the case of small direction and sliding angles, such that the sinus and the tangent can be approximated with the value of the angle in radian, and the lateral forces can be considered proportional to the sliding angle. If different nonlinear models of the tyre are used for the lateral and longitudinal forces, then the three freedom degree model is also a nonlinear one.

To obtain the equations of the three freedom degree model the relations of forces and momentum dynamic equilibrium are used, on the x, y and z directions.

The longitudinal and lateral accelerations are also used.

Decomposition of the forces on the x axis:

$$m*\dot{V}*\cos\beta - m*\frac{V^2}{\rho}*\sin\beta = F_{x1} + F_{x5} - F_{x2} - F_{x6} + F_{x3} + F_{x4} - F_{xl} \quad (1)$$

where: F_{x1} – resistive force of the air on the x axis;

$\dot{V}*\cos\beta$ - the tangent acceleration on the x axis;

$\frac{V^2}{\rho}*\sin\beta$ - the center-oriented acceleration on the x axis.

Assume F_{x1} negligible because it is very small compared to the other forces.

$$m*\dot{V}*\cos\beta - m*\frac{V^2}{\rho}*\sin\beta = (F_1 + F_5)*\cos\delta - (F_2 + F_6)*\sin\delta + (F_{x3} + F_{x4}) \quad (2)$$

Decomposition of the forces on the y axis:

$$m*\dot{V}*\sin\beta + m*\frac{V^2}{\rho}*\cos\beta = F_{y1} + F_{y5} + F_{y2} + F_{y6} + F_{y3} + F_{y4} - F_{yl} \quad (3)$$

where: F_{yl} – resistive force of the air on the y axis;

$\dot{V}*\sin\beta$ - the tangent acceleration on the x axis;

$\frac{V^2}{\rho}*\cos\beta$ - the centripet acceleration on the y axis.

Assume F_{yl} negligible because it is very small compared to the other forces.

$$m*\dot{V}*\sin\beta + m*\frac{V^2}{\rho}*\cos\beta = (F_1 + F_5)*\sin\beta + (F_2 + F_6)*\cos\delta + (F_{y3} + F_{y4}) \quad (4)$$

The equations of the moment against the z axis passing through the mass center of the vehicle:

$$J_z * \ddot{\psi} = \frac{E}{2} * F_{x1} - \frac{E}{2} * F_{x5} + l_f * F_{y1} + l_f * F_{y5} - \frac{E}{2} * F_{x2} + \frac{E}{2} * F_{x6} + l_f * F_{y2}$$
$$+ l_f * F_{y6} + \frac{E}{2} * F_{x3} - \frac{E}{2} * F_{x4} - l_s * F_{y3} - l_s * F_{y4} - l_{CG} * F_{yl} \quad (5)$$

or:

$$J_z * \ddot{\psi} = \frac{E}{2} * (F_1 - F_5) * \cos\delta + l_f * (F_1 + F_5) * \sin\delta - \frac{E}{2} * (F_2 - F_6) * \sin\delta$$
$$+ l_f * (F_2 + F_6) * \cos\delta + \frac{E}{2} * (F_{x3} - F_{x4}) - l_s * (F_{y3} + F_{y4}) \quad (6)$$

where: $\ddot{\psi}$ - angular acceleration;

J_z – the rotational inertia.

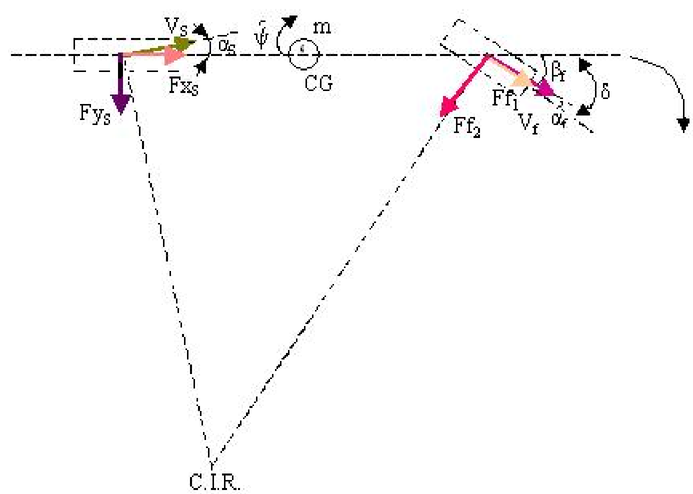

Fig.2 The simplified model of the vehicle with three freedom degrees

where: C.I.R. – Instantaneous Centre of Rotation;

α_f - the front sliding angle;

α_s the rear sliding angle;

β_f – the deviation angle of the wheel;

F_{f1} – the sum of the front longitudinal forces;

F_{f2} – the sum of the front lateral forces;
F_{xs} – the sum of the rear longitudinal forces;
F_{ys} – the sum of the rear lateral forces;
V_f – the front velocity of the wheel;
V_s – the rear velocity of the wheel.

The resulting forces on the two directions, longitudinal and transversal, for the front and rear half respectively, reside from the following equations:

$F_1 + F_5 = F_{f1}$, (7) $F_{x3} + F_{x4} = F_{xs}$, (9)

$F_2 + F_6 = F_{f2}$, (8) $F_{y3} + F_{y4} = F_{ys}$ (10)

The differences are negligible compared with the sums.

Replacing the relations (7), (8), (9) in the relation (2) we obtain:

$$m*\dot{V}*\cos\beta - m*\frac{V^2}{\rho}*\sin\beta = F_{f1}*\cos\delta - F_{f2}*\sin\delta + F_{xs} \quad (11)$$

Replacing the relations (7), (8), (10) in the relation (4) we obtain:

$$m*\dot{V}*\sin\beta + m*\frac{V^2}{\rho}*\cos\beta = F_{f1}*\sin\delta + F_{f2}*\cos\delta + F_{ys} \quad (12)$$

Replacing the relations (7), (9), (10) in the relation (6) we obtain:

$$J_z*\ddot{\psi} = l_f*F_{f1}*\sin\delta + l_f*F_{f2}*\cos\delta - l_s*F_{ys} \quad (13)$$

The perpendicular forces on the wheel are expressed in the following, according to the HRSI model:

$$F = \frac{1}{1-\lambda}*c_\alpha*\left(\frac{1}{H} - \frac{1}{4*H^2}\right)*tg\alpha$$

where:

λ – the sliding of the wheel which can be expressed as: $\lambda = \frac{V-\omega*r}{V} = 1 - \frac{\omega*r}{V}$

where:
V – velocity of the vehicle;
ω – angular velocity of the wheel;
r – the rolling radius of the wheel.
c_α – the transverse elasticity coefficient of the tyre (experimentally determined) ;
α – the sliding angle of the wheel;

$$H = \sqrt{\left(\frac{\lambda}{\lambda-1}*\frac{c_\lambda}{\mu*G_r}\right)^2 + \left(\frac{1}{\lambda-1}*\frac{c_\alpha}{\mu*G_r}*tg\alpha\right)^2}$$

where:

c_λ – the longitudinal elasticity coefficient of the tyre (experimentally determined) ;
μ – the friction coefficient between the tyre and the rolling path;
G_r – the vertical charge of the wheel.

The sliding angle of the wheel it is considered to be very small $\Rightarrow tg\alpha \simeq \alpha \Rightarrow F = \frac{1}{1-\lambda} * c_\alpha * \left(\frac{1}{H} - \frac{1}{4*H^2}\right) * \alpha$

We denote $c_\alpha' \simeq \frac{1}{1-\lambda} * c_\alpha * \left(\frac{1}{H} - \frac{1}{4*H^2}\right)$ where $F \simeq c_\alpha' * \alpha$ (14)

$$m * \dot{V} * \cos\beta - m * \frac{V^2}{\rho} * \sin\beta = -F_{f2} * \sin\delta \quad (15)$$

$$m * \dot{V} * \sin\beta + m * \frac{V^2}{\rho} * \cos\beta = F_{f2} * \cos\delta + F_{ys} \quad (16)$$

$$J_z * \ddot{\psi} = l_f * F_{f2} * \cos\delta - l_s * F_{ys} \quad (17)$$

In the case of circular motion $V = \omega * r$ (18)

where:
V – tangent velocity;
ω – angular velocity;
r – the radius of the trajectory.

Transforming the relation (18) for the studied case one can write:

$$V = \left(\dot{\beta} + \dot{\psi}\right) * \rho \quad (19)$$

By multiplying the relation (19), both left and right sides with V, we obtain

$$V * V = V * \left(\dot{\beta} + \dot{\psi}\right) * \rho \quad (20)$$

$$V * \frac{V}{\rho} = V * \left(\dot{\beta} + \dot{\psi}\right) \quad (21)$$

By replacing the relations (14) and (21) in the relation (16) and considering the turning angle small ($\sin\delta = \delta$ and $\cos\delta = 1$) we obtain:

$$m * \dot{V} * \sin\beta + m * V * \left(\dot{\beta} + \dot{\psi}\right) * \cos\beta = 2 * c_{\alpha s}' * \alpha_f + 2 * c_{\alpha s}' * \alpha_s \quad (22)$$

Introducing the relation (14) into (17) relația (14) we will have:

$$J_z * \ddot{\psi} = 2 * c_{\alpha f}' * \alpha_f * l_f - 2 * c_{\alpha s}' * \alpha_s * l_s \quad (23)$$

where:

$$\alpha_f = \delta - \beta - \frac{l_f * \dot{\psi}}{V} \quad (24)$$

$$\alpha_s = \frac{l_s * \dot{\psi}}{V} - \beta \tag{25}$$

The mathematical model of the motion of a vehicle on the curved trajectory is represented by the equations (22) and (23). Using these relations the stability of the vehicle in the curve can be studied.

The present model considers as negligible any influence of the wind, that the turning angle of the front wheels is small (around 15°) and that the vehicle can be compared with a simplified model called the bycicle model, i.e. the moment of the forces parallel to the symmetry axis of the vehicle are neglected.

Based on the mathematical model the representation of the system can be rewritten:

$$y = C * x + d * u \Rightarrow \begin{bmatrix} \beta \\ \dot{\psi} \\ \ddot{y} \end{bmatrix} = \begin{bmatrix} g & h \\ i & j \\ k & l \end{bmatrix} * \begin{bmatrix} \beta \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} m \\ n \\ p \end{bmatrix} * \delta \tag{26}$$

Identifying the coefficients and taking into consideration the relations (24) and (25) and the relation:

$$\ddot{y} = V * (\dot{\beta} + \dot{\psi}) * \cos\beta \tag{27}$$

The matrices C and d can be written as:

$$C = \begin{bmatrix} 1 & 0 \\ 0 & 1 \\ -\frac{2 * c'_{\alpha f} + 2 * c'_{\alpha s} + m * \dot{V}}{m} & \frac{2 * \left[c'_{\alpha s} * l_S - c'_{\alpha f} * l_f \right]}{m * V} \end{bmatrix}, \quad d = \begin{bmatrix} 0 \\ 0 \\ \frac{2 * c'_{\alpha f}}{m} \end{bmatrix}$$

In order to determine the stability of the vehicle in curves, the two parameters characterizing the movement of the vehicle in this situation are $\dot{\beta}$ and $\ddot{\psi}$, expressed in:

$$\begin{bmatrix} \dot{\beta} \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} a_1 & b_1 \\ c_1 & d_1 \end{bmatrix} * \begin{bmatrix} \beta \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} e \\ f \end{bmatrix} * \delta =$$
$$\begin{bmatrix} a_1 * \beta + b_1 * \dot{\psi} \\ c_1 * \beta + d_1 * \dot{\psi} \end{bmatrix} + \begin{bmatrix} e * \delta \\ f * \delta \end{bmatrix} = \begin{bmatrix} a_1 * \beta + b_1 * \dot{\psi} + e * \delta \\ c_1 * \beta + d_1 * \dot{\psi} + f * \delta \end{bmatrix} \tag{28}$$

From the model equations and the relations (24) and (25) the following expressions can be written:

$$\ddot{\psi} = \beta * \frac{2 * \left[c'_{\alpha s} * l_S - c'_{\alpha f} * l_f \right]}{J_z} + \dot{\psi} * \frac{-2 * \left[c'_{\alpha f} * l_f^2 + c'_{\alpha s} * l_S^2) \right]}{V * J_z} + \delta * \frac{2 * c'_{\alpha f} * l_f}{J_z} \tag{29}$$

$$\dot{\beta}=\beta*\frac{-2*c'_{\alpha f}-2*c'_{\alpha s}-m*\dot{V}}{m*V}+\dot{\psi}*\frac{2*\left[c'_{\alpha s}*l_S-c'_{\alpha f}*l_f)\right]-m*V^2}{m*V^2}+\delta*\frac{2*c'_{\alpha f}}{m*V} \tag{30}$$

Identifying the coefficients the values can be computed a_1, b_1, c_1, d_1, e and f.

The coeficients *k*, *l*, *p* can be determined from the dynamic ecuations of the model:

$$k=-\frac{2*c'_{\alpha f}+2*c'_{\alpha s}+m*\dot{V}}{m}; \tag{34}$$

$$l=\frac{2*\left[c'_{\alpha s}*l_S-c'_{\alpha f}*l_f)\right]}{m*V}; \tag{35}$$

$$p=\frac{2*c'_{\alpha f}}{m}. \tag{36}$$

In this paper is studied the case of movement on a curve with constant speed that implies $\dot{V}=0$.

Knowing C and d matrices with the help of relation $H(s)=C*(s*I-A)^{-1}*b+d$, where $b=\begin{bmatrix} e \\ f \end{bmatrix}$ and $A=\begin{bmatrix} a_1 & b_1 \\ c_1 & d_1 \end{bmatrix}$ the transfer functions for each of the three items β, $\dot{\psi}$ şi $\ddot{y}$ can be written:

$$H_1(s)=\frac{\beta(s)}{\delta(s)}=\frac{1}{(s-a_1)*(s-d_1)-c_1*b_1}*\left[e*(s-d_1)+b_1*f\right] \tag{37}$$

$$H_2(s)=\frac{\dot{\psi}(s)}{\delta(s)}=\frac{1}{(s-a_1)*(s-d_1)-c_1*b_1}*\left[c_1*e+(s-a_1)*f\right] \tag{38}$$

$$H_3(s)=\frac{\ddot{y}(s)}{\delta(s)}=\frac{1}{(s-a_1)*(s-d_1)-c_1*b_1}* \left\{e*\left[k*(s-d_1)+l*c_1\right]+f*\left[k*b_1+l*(s-a_1)\right]\right\}+p \tag{39}$$

In the Fig. bellow the model of movement on curved trajectory is presented.

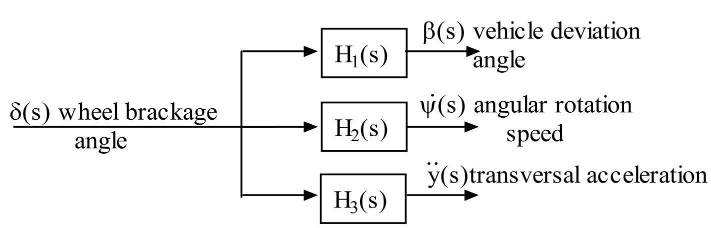

Fig. 3 The model of movement on curved trajectory

Assuming that the vehicle runs on a road without moisture or mud, it can be considered that $\lambda = 0$. From the (14) relation resides:

$$H_f = \frac{c_\alpha}{\mu * m * g} * tg\alpha_f * \frac{l}{l_s} \tag{40}$$

respectively,

$$H_s = \frac{c_\alpha}{\mu * m * g} * tg\alpha_s * \frac{l}{l_f} \tag{41}$$

where:

g is the gravitational acceleration.

Relations (27) and (28) can also be written :

$$\alpha_f = \delta - \beta - \frac{l_f}{\rho} \tag{42}$$

$$\alpha_s = \frac{l_s}{\rho} - \beta \tag{43}$$

where ρ is vehicle trajectory radius and it is given by the following relation :

$$\rho = \sqrt{\left(\frac{\sqrt{l^2\left(tg^2\delta + 1\right)^2 + E^2 tg^2\delta} - l\left(tg^2\delta - 1\right)}{2 \cdot tg\delta}\right)^2 + l_s^2} \tag{44}$$

From relations (41),(42),(43),(44) and (16) resides :

$$c'_{\alpha f} = c_\alpha * \left(\frac{1}{H_f} - \frac{1}{4 * H_f^{\,2}}\right) \tag{45}$$

and

$$c'_{\alpha s} = c_\alpha * \left(\frac{1}{H_s} - \frac{1}{4 * H_s^{\,2}}\right) \tag{46}$$

Table.1

The calculation parameter values for Opel Omega car

Simbol	c_α	l_f	l_s	m	Jz
Valoare	80000	1,3	1,45	1450	1920
Unitate	$\left[\frac{N}{rad}\right]$	[m]	[m]	[kg]	[Kg*m]

The coeficient values a_1, b_1, e, c_1, d_1, f ,k, l and p can be calculated based on (39),(40),(41),(42),(32),(33) and (34) relations numericaly replacing the constants given in table.1. With the aide of Maple 7 programme the $H_1(s)$, $H_2(s)$ and $H_3(s)$ transfer function coefficients can be calculated.

In the table.2 the real parts of the transfer functions poles are being calculated for five different speed and brackage angle values.

Table.2.

The real parts of the transfer functions poles for different speed and brackage angle values

V(km/h) / δ(°)	50	60	80	100	120
5	**-560.92** **-20.51**	**-478.72** **-20.43**	**-390.97** **-18.38**	**-327.17** **-17.99**	**-341.94** **-18.61**
10	**-1.66** **-0.01**	**-2.99** **-0.01**	**878.95** **-13.99**	306.69 -15.13	88.32 -15.38
15	**-2.26** **-0.01**	**-2.35** **-0.01**	4.25 -0.01	1.45 -0.01	1.42 0.02
20	**-4.02** **-0.01**	2.81 -0.01	2.75 -0.01	2.87 0.01	2.39 0.01

The external stability criterium says that a system (A,b,c^T) is:

- **externaly stable**, if and only if the transfer functions poles have $\operatorname{Re} p_i \leq 0$ and those poles wich have the real part equal to zero have to be simple.
- **strictly externaly stable**, if and only if the transfer functions poles have $\operatorname{Re} p_i < 0$, this condition being equivalent to: $\mathcal{P}\left[H(s)\right] \subset \mathbb{C}^-$

Accordingly to the external stability criterium enounced above it resides the the vehicle is stable only for the 5-50, 5-60,5-80,5-100,5-120,10-50,10-60,10-80,15-50,15-60,20-50 pear of values.

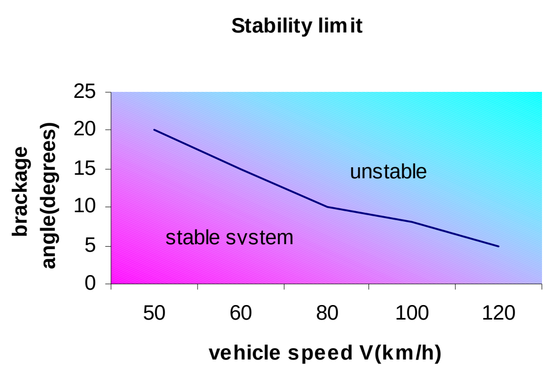

Fig. 4.The vehicle stability limit on a curved trajectory

The curve from the picture above represents the studied vehicle stability limit, in other words if the vehicle has a speed and a brackage angle so that the coresponding point (wich has theese coordinates) is under the curve, than the vehicle movement on that path is stable and if the coreponding point is placed above the curve that the movement is unstable. If this representation is implemented in the on-board computer memory the computer coud allert the driver to reduce the speed if the vehicle aproaches his stability limit when cornering that way being able to avoid in time the critical situations of vehicle controll loss.

4. Conclusions

The conclusion that can be drawn from this study is that for speeds wich exced 80 km/h, relatively small brackage angles can destabilize the vehicle in his movement on a curved path. Also, for brackage angles greater than 15 the vehicle speed has to decrease under 60 km/h in order to mentain a stable course. If theese prety narrow limits are exceded, the vehicle may loose the ground contact to the inner wheels that leading to a powerfull draw-in tendency wich will destabilise even more the vehicle.

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