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On

CONTROL METHODOLOGY TO ALTER AUTOMOBILE ROLLOVER TENDENCIES

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April 00

Control Methodology To Alter Automobile Rollover Tendencies

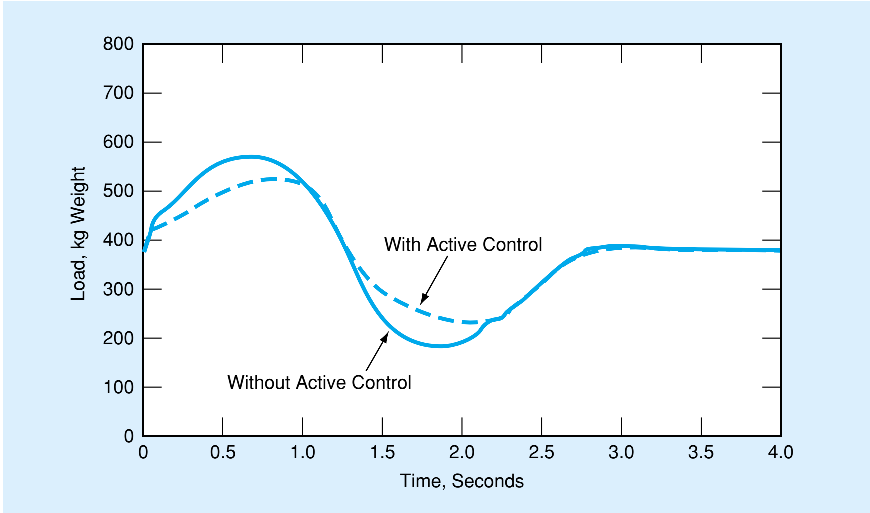
Dynamics would be altered to effect safer distribution of loads to wheels.

NASA's Jet Propulsion Laboratory, Pasadena, California

A methodology of active control has been developed in an effort to alter (preferably to reduce) the tendency of a four-wheel land vehicle to roll over during tight turns and similar maneuvers. At the time of reporting the information for this article, hardware and software to implement the methodology were undergoing development and testing for incorporation into a variable-dynamics testbed vehicle (VDTV) — an experimental automobile that will be capable of exhibiting a broad range of dynamic characteristics for research on crash avoidance and driving-related human factors.

The VDTV will have four-wheel steering, front and rear active antiroll-bar systems, four adjustable dampers, and other active control devices that will be controlled by a computer running algorithms based on mathematical models of the dynamics of the vehicle. These devices will be used to alter several rollover-related characteristics of the dynamics of the vehicle, including notably its understeer coefficient, front/rear load-transfer distribution in lateral maneuvers that involve high accelerations, and the frequency and damping coefficient of the vehicle roll mode. Load-transfer distributions are particularly significant because to prevent rollover, one must ensure that the load on any tire never approaches zero during a severe maneuver.

A study has been performed to investigate how changes in the algorithms that control the active devices could effect significant changes in these characteristics. In particular, the study focused on how (1) an increase in the stiffness of the front antiroll bar, in conjunction with an increase in the front damper rate and with out-of-phase rear steering, could increase resistance to rollover in high-acceleration lateral maneuvers without changing the vehicle understeer coefficient (see figure); and (2) conversely, an increase in the stiffness of the rear antiroll bar, in conjunction with decrease in the rear damper rate and with in-phase rear steering, could decrease resistance to rollover. The results of the study indicate that the design of the VDTV provides for the right combination of active devices that will make the VDTV an effective testbed for research on rollover-related human factors.



The **Load on the Left Rear Tire** of the VDTV was simulated by computer for a two-lane-change maneuver. By use of active control to increase the stiffness of the front antiroll bar, the minimum load at an extreme point of the maneuver was increased, thereby increasing the margin against rollover.

This work was done by Allan Y. Lee of Caltech for **NASA's Jet Propulsion Laboratory**.
NPO-20545

1. Novelty - Describe what is new and different about your work and its improvement over the prior art.

   To enhance vehicle safety against rollover accidents, one must try to ensure that none of the four tires' normal loadings is close to zero (lifting of tire). One simple way to achieve this is to alter the front/rear load transfer distribution of the vehicle in a lateral-g maneuver. The load transfer distribution can be effectively alter by changing the roll stiffnesses of the front and rear anti-roll bars of the car.

2. Technical Disclosure

   A. Problem - Motivation that led to development or problem that was solved

      How to improve vehicle rollover resistance at the time of high-g maneuver

   B. Solution

      Control the front to rear load transfer distribution of the car via changing its front (or front and rear) roll bar stiffness(es).

   C. Detailed description and Explanation

      See paper attached.

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EXHIBIT A

Coordinated control of active devices to alter vehicle rollover tendencies

U.S. Death toll
is 41K P.A.
30%

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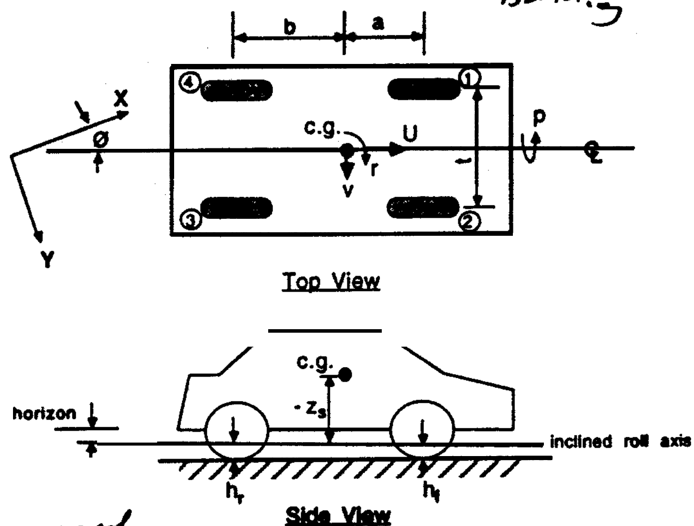
Abstract

A Variable Dynamic Testbed Vehicle is presently being built for the National Highway Traffic Safety Administration. It will have four-wheel steering, front and rear active antiroll bar systems, four adjustable dampers, and other active controls. Using these active devices, we can alter the vehicle's understeer coefficient, front/rear load transfer distribution in high-g lateral maneuvers, and roll mode frequency and damping. This study investigates how these active systems could be controlled to alter the vehicle rollover tendencies. In particular, we study how an increased front antiroll bar stiffness, in conjunction with an increased front damper rate and out-of-phase rear steering could improve vehicle rollover resistance and enhance vehicle safety. Similar but "reverse" algorithms could be used to artificially degrade the rollover resistance of a vehicle. Rollover-related accidents could then be studied using such a vehicle. Results obtained could also provide guidelines for the safe operation of the variable dynamic vehicle in limit lateral maneuvers.

→ research too]

1. Introduction

A Variable Dynamics Testbed Vehicle (VDTV) is presently being built for the National Highway Traffic Safety Administration's Office of Crash Avoidance Research (Lee, Marriott, and Le, 1997). This vehicle will be capable of emulating a broad range of automobile dynamic characteristics, allowing it to be used in collision detection systems development and in driving-related human factors research, among other areas. To accomplish this goal, the VDTV will have the following set of active devices: a four-wheel steering (4WS) system, both front and rear active antiroll bars, four adjustable dampers, among others. Software changes made to the algorithms that control these active devices can then effect significant changes in the vehicle's understeer coefficient, the front/rear load transfer distribution in lateral maneuvers, and the roll mode frequency and damping. This study investigates how these active devices could be controlled to alter the vehicle rollover tendencies. In particular, we study how an increased front antiroll bar stiffness, in conjunction with an increased front damper rate and out-of-phase rear steering could improve vehicle rollover resistance in high-g lateral maneuvers. Conversely, an increased rear antiroll bar stiffness, in conjunction with a decreased rear damper rate and in-phase rear steering could degrade the vehicle rollover resistance. Results obtained could also provide guidelines for the safe operation of the variable dynamic vehicle in limit lateral maneuvers.



S/L road construction

Figure 1: Schematic diagram of a passenger vehicle

2. Vehicle Dynamics Model

Consider a vehicle moving over a flat and level road surface (Fig. 1). If the forward speed U is kept constant, the vehicle has three degrees of freedom represented by the side velocity v, the roll rate p, and the yaw rate r. The cornering forces and aligning torques generated by the tires are denoted by F_{yi} and N_i (i = 1–4), respectively. Ignoring contributions due to pitching dynamics, the equations of motion are:

$$(I_{zz_s} + I_{zz_u})\dot{r} + I_{xz_s} \tan \Delta \dot{p} = a \cos \delta_f \sum_{i=1}^{2} F_{yi} - b \cos \delta_r \sum_{i=3}^{4} F_{yi} + \sum_{i=1}^{4} N_i + \frac{t}{2}\{\sin \delta_f (F_{y2} - F_{y1}) + \sin \delta_r (F_{y3} - F_{y4})\} \qquad (1)$$

$$(M_s + M_u)(Ur + \dot{v}) - M_s z_s \dot{p}$$
$$= \cos\delta_f \sum_{i=1}^{2} F_{yi} + \cos\delta_r \sum_{i=3}^{4} F_{yi} \quad (2)$$
$$I_{zz_s}\dot{p} - I_{zz_s}\tan\Delta\dot{r} - M_s z_s(Ur + \dot{v})$$
$$= -(D_f + D_r)p - (K_f + K_r + M_s g z_s)\phi \quad (3)$$
$$\dot{\phi} = p \quad (4)$$

where a and b define the location of the vehicle's c.g. between the axles, and M_s and I_{zz_s} denote the sprung mass and the yaw moment of inertia of the vehicle's sprung mass, respectively. Similarly, M_u and I_{zz_u} denote the unsprung mass and the yaw moment of inertia of the vehicle's unsprung mass, respectively. The roll moment of inertia of the sprung mass with respect to the roll axis is denoted by I_{xx_s}. In (3–4), ϕ denotes the roll angle of the vehicle's sprung mass about the inclined roll axis, and $-z_s$ is the height of the sprung mass c.g. above the inclined roil axis. The slope of the inclined roll axis is tan A $= (h_f - h_r)/(a + b)$, where h_f and h_r denote the heights of the inclined roll axis above the ground plane at the front and rear axles, respectively. Estimated magnitudes of various vehicle parameters are: (M_s, Mu) = (1819, 200) kg., (a, b, t, h_f, h_r, $-z_s$) = (1.02, 1.69, 1.55, 0.046, 0.0066, 0.538) m, (I_{zz_u}, I_{zz_s}, I_{xx_s}) = (475, 2442, 10.72) kg $- m^2$, (K_f, K_r) = (1186, 798) Nm/deg, and (D_f, D_r) = (48.9, 48.9) Nms/deg.

In (3), the roll stiffness of the vehicle at the front and rear axles each contains two components. The first component represents that contributed by the passive suspension springs (K_{fP} and K_{rP}), and the second component is that contributed by two active antiroll bars (K_{fA} and K_{rA}):

$$K_f \triangleq K_{fP} + K_{fA} \quad (5)$$
$$K_r \triangleq K_{rP} + K_{rA} \quad (6)$$

Since both K_{fA} and K_{rA} could be actively controlled, we can, in real time, alter the following load transfer distribution factor, κ_c:

$$\kappa_c \triangleq \frac{K_f - K_r}{K_f + K_r} \quad (7)$$

To account for the actuator dynamics of the front and rear antiroll-bar systems, the instantaneous load transfer distribution factor, κ, is related to κ_c by the following relation:

$$\tau_\phi \dot{\kappa} + \kappa = \kappa_c \quad (8)$$

where τ_ϕ is the time constant of the active antiroll-bar system. The antiroll bar actuator bandwidth is 12 Hz.

The front and rear damping coefficients in (3) could also be actively controlled using four adjustable dampers. The instantaneous damping rates are:

$$D_f = \eta_f D_{fP} \quad (9)$$
$$D_r = \eta_r D_{rP} \quad (10)$$

where D_{fP} and D_{rP} represent the nominal passive damping rates at the vehicle's front and rear axles, respectively. The control variables, η_f and η_r, which *vary* between 0.5 and 2.5, are used to alter the vehicle roll mode damping.

A nonlinear tire model proposed by Allen, Magdaleno, Rosenthal, Klyde, arid Hogue (1995) is used in this study. In this model, the *lateral* force and aligning torque produced by a tire are given as functions of the tire slip and camber angles, the normal loading on that tire, and the coefficient of friction between that tire and the road surface. The commanded slip angles α_{fc} and α_{rc} at the front and rear axles are given respectively by the following kinematic relations:

$$\alpha_{fc} = \tan^{-1}\left(\frac{v + ar - h_f p}{U}\right) - \delta_f \quad (11)$$
$$\alpha_{rc} = \tan^{-1}\left(\frac{v - br - h_r p}{U}\right) - \delta_r \quad (12)$$

where δ_f and δ_r denote the front and rear steering angles, respectively. These commanded slip angles are related to the instantaneous slip angles *via* (see Heydinger, Garrott, and Chrstos, 1991):

$$\ddot{\alpha}_f + 2\zeta\Omega\dot{\alpha}_f + \Omega^2\alpha_f = \Omega^2\alpha_{fc} \quad (13)$$
$$\ddot{\alpha}_r + 2\zeta\Omega\dot{\alpha}_r + \Omega^2\alpha_r = \Omega^2\alpha_{rc} \quad (14)$$

where ζ and Ω are the damping ratio and natural frequency of the second-order tire dynamics. Estimated values of ζ and Ω are 0.8 and 60 Ho, respectively.

The normal loadings on the four tires are given by the following expressions:

$$F_{z1} = (M_s + M_u)\frac{bg}{2(a+b)} + \Delta F_f \quad (15)$$
$$F_{z2} = (M_s + M_u)\frac{bg}{2(a+b)} - \Delta F_f \quad (16)$$
$$F_{z3} = (M_s + M_u)\frac{ag}{2(a+b)} - \Delta F_r \quad (17)$$
$$F_{z4} = (M_s + M_u)\frac{ag}{2(a+b)} + \Delta F_r \quad (18)$$

where ΔF_f and ΔF_r denote the load transfers at the front and rear axles of the vehicle, respectively. These load transfers are proportional to the lateral acceleration a_y of the vehicle's sprung mass:

$$a_y = Ur + \dot{v} + (-z_s)\dot{p} \quad (19)$$
$$\Delta F_f = (M_s + M_u)\frac{a_y(-z_s)}{2t}(1 + \kappa) \quad (20)$$
$$\Delta F_r = (M_s + M_u)\frac{a_y(-z_s)}{2t}(1 - \kappa) \quad (21)$$

where t is the average track width of the vehicle. The expression for the lateral acceleration a_y is true only if $K_f + K_r \gg M_s(-z_s)g$ and $K_f(-z_s) \gg (K_f + K_r)b/(a + b)h_f$. These inequality conditions are satisfied in our study.

In arriving at (12), we have assumed that the vehicle has a rear steering actuator. In our study, first-order models are used to account for steering actuator dynamics:

$$\tau_f\dot{\delta}_f + \delta_f = \delta_{fc} \quad (22)$$
$$\tau_r\dot{\delta}_r + \delta_r = \delta_{rc} \quad (23)$$

Here δ_{fc} and δ_{rc} are commands to the front and rear steering actuators, respectively. In (22-23), τ_f and τ_r are the time constants of the front and rear steering actuators, respectively. The bandwidth of these steering actuators is 15 Hz.

3. Kinematic Model

In addition to the above described dynamics equations, the following kinematic relations *are* used to compute the resultant

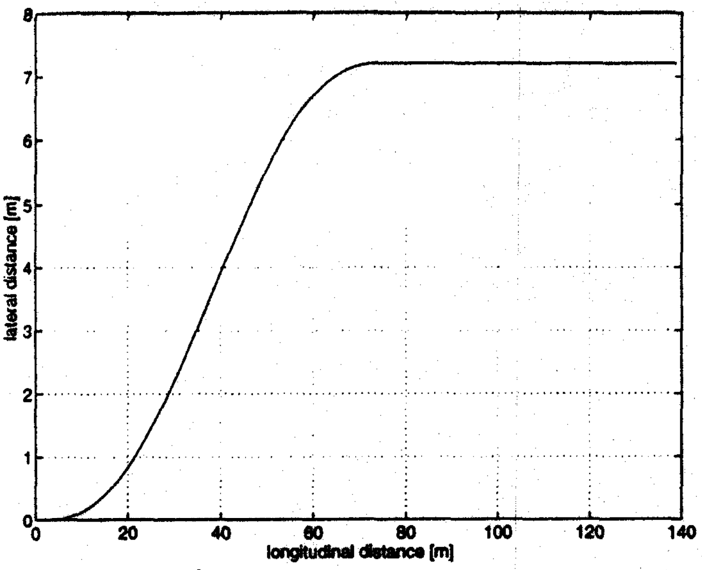


Figure 2: A reference two-lane change trajectory

vehicle heading and trajectory:

$$\dot{\psi} = r \quad (24)$$
$$\dot{x} = U\cos\psi - v\sin\psi \quad (25)$$
$$\dot{y} = U\sin\psi + v\cos\psi \quad (26)$$

In Fig. 1, x and y are the rectilinear coordinates of the vehicle's c.g. relative to an arbitrary reference origin. The angle ψ is the angle formed between the vehicle's longitudinal axis and the x axis, defined positive in the clockwise direction.

4. High-g Two-Lane Change Maneuvers

A severe two-lane change maneuver typically involves large two-way load transfers between the inside and outside tires. In extreme cases, it can lead to the lifting of one or more tires above ground, resulting in a rollover. In our study, we use the following fifth-order polynomial (Nelson, 1889) to represent the desired two-lane lane change trajectory that the driver would like to track closely:

$$y(x) = y_f\{10(\frac{x}{x_f})^3 - 15(\frac{x}{x_f})^4 + 6(\frac{x}{x_f})^5\} \quad (27)$$

where x_f and y_f denote the position of the vehicle at the end of the lane change. In our two-lane change maneuver, $y_f = D$ and $x_f = U \times T_{lc}$, where D is the lateral displacement of the vehicle after the lane change and T_{lc} is the lane change time. Fig. 2 illustrates this fifth-order trajectory for the following nominal scenario: $U = 125$ km/h, $D = 7.2$ *m*, and $T_{lc} = 2.2$ *s*. By computing $\frac{dy}{dx}$ and $\frac{d^2y}{dx^2}$ using (27), the reference curvature c_r and reference heading angle ψ_r of this trajectory are derived:

$$\psi_r(x) = \tan^{-1}(\frac{dy}{dx}) \quad (28)$$
$$c_r(x) = \frac{d^2y}{dx^2}/\{1 + (\frac{dy}{dx})^2\}^{\frac{3}{2}} \quad (29)$$

To track this trajectory, a driver uses a combined feedforward and feedback control law to steer the vehicle. Let us define: $\Delta c(t) = c_r(t) - r/U$, $\Delta y(t) = y_r(t) - y(t)$, and $\Delta\psi(t) = \psi_r(t) - \psi(t)$. Here, $\Delta c(t)$ denotes the difference between the reference curvature and the current curvature of the vehicle, r/U; $\Delta y(t)$ denotes how far the vehicle's c.g. has deviated from the centerline of the reference trajectory, while $\Delta\psi(t)$ denotes how much the vehicle's centerline has deviated from the local tangent of the reference trajectory. The steering command is:

$$\delta_{fc}(t) = \overbrace{K_c c_r(t + t_p)}^{\text{feedforward}} + \underbrace{K_{\Delta c}\Delta c(t) + K_{\Delta\psi}\{\Delta\psi(t) + K_{\Delta y}\Delta y(t)\}}_{\text{feedback}} \quad (30)$$

Here, $c_r(t + t_p)$ is the reference curvature at a look-ahead distance as determined by the driver's preview time t,. The feedforward gain is K_c and the feedback gains are $K_{\Delta c}$, $K_{\Delta y}$ and $K_{\Delta\psi}$. As conjectured by Lee (1995), we assume in this study that an experienced driver does not use the positional and heading errors to generate the steering command during the reflexive phase of the lane change maneuver. Instead, he executes an "open-loop" steering command based upon the estimated *roadway* curvature and his learnt knowledge of the vehicle's lateral characteristics. Accordingly, in this study, we use $K_{\Delta y} = K_{\Delta\psi} = 0$, while $K_{\Delta c}$ and K_c *are* determined using an approach described in Lee, 1995. In the subsequent regulatory phase of the lane change maneuver, the driver will use small steering adjustments to zero out residuals in the vehicle's yaw rate, side velocity, heading angle, etc., in a "closed loop."

5. Vehicle Performance: Passive Vehicle

A two-land change maneuver made using the steering command computed using (30) is simulated. Time histories of the responses, Figs. be used as the baseline vehicle responses to which results obtained using an actively controlled VDTV are compared with. Figs. 34 depict the time histories of the vehicle's yaw rate and front steering angle, respectively. Figs. *5-6* depict the transient variations of the normal loadings at the left and right rear tires. In these figures, results obtained for the passive vehicle and the VDTV are given by solid and dashed lines, respectively. Details on how the VDTV results were obtained will be *given in* the following.

For the vehicle used in our study, the vehicle's c.g. is closer Hence, tires tires. As such, in *a* high-g maneuver, the lowest tire loading will likely to occur at one of the two rear tires. To avoid a rollover, none of these tire loadings should be near zero. From Fig. 5 we observe that *minimum* loading on the left rear tire occurred at about the same time that the yaw rate peaks in the negative direction. This minimum tire loading, about 186 kg.wt., is also the lowest tire loading among all the tires throughout the lane change maneuver. With a minimum tire loading this high, there is no incipient rollover in this particular lane change maneuver. This is to be expected since the $t/(2h_{cg})$ ratio of the vehicle, about 1.45, is among the highest of all production vehicles. If, instead, the $t/(2h_{cg})$ ratio of the vehicle is relatively low, and/or that the peak "g" level of the lane change maneuver (or other types of aggressive accident-avoidance maneuvers) is high, then the vehicle will be on the verge of an unrecoverable rollover.

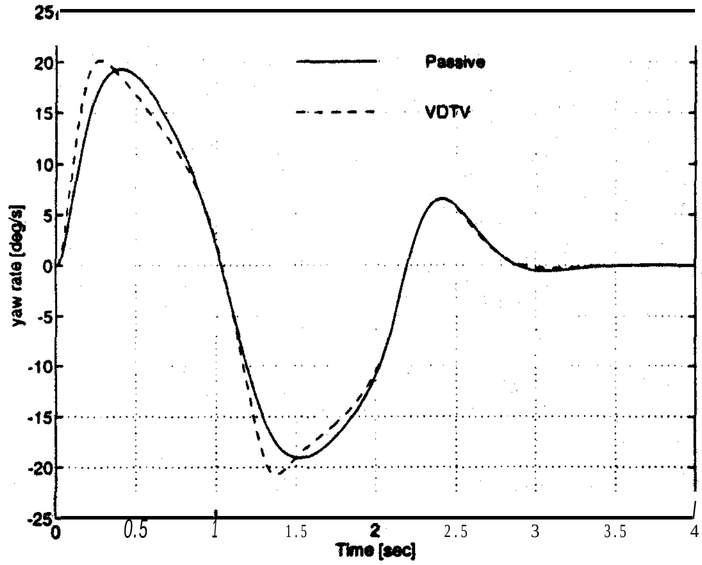


Figure 3: Yaw rate responses of VDTV and passive vehicles

6. Rollover resistance: How to improve it?

To increase further the vehicle resistance to rollover, we could increase the stiffness of the active front antiroll bar K_{fA} with the vehicle yaw rate r: $K_{fA} \propto |r|$. The rationale is as follows. The vehicle's lateral acceleration is related to the vehicle's yaw rate. If we increase the vehicle's front antiroll bar stiffness in a high-g maneuver, a larger proportion of the load transfer will be carried by the front axle, and that at the rear axle will be reduced. That being the case, the magnitude of the minimum loading on the rear left tire will increase, and the vehicle rollover resistance is improved. Alternative control algorithms that could achieve the same effect include: $K_{fA} \propto |\delta_{fc}|$, $K_{fA} \propto |a_y|$, K_{fA} a a'_y, etc. See also Wiiams and Haddard, 1995.

On a level road surface, a vehicle does not rollover in low-g lateral maneuvers. Hence, there is no motivation to activate the front active antiroll bar in these low-g scenarios. On the other hand, there is a mechanical limit to which the front antiroll-bar stiffness can generate. Considering these factors, the above suggested control algorithm is modified as follows:

$$K^c_{fA} = \begin{cases} 0 & \text{if } |r| \le r_{db} \\ k_r \, | r - \mathrm{sgn}(r) r_{db} | & \text{otherwise} \end{cases} \quad (31)$$

$$K_{fA} = \min\{K^{\mathrm{mech}}_{fA}, K^c_{fA}\} \quad (32)$$

Here, K^{mech}_{fA} denotes the mechanical limit of the front antiroll bar stiffness, and r_{db} denotes a yaw rate deadband. If the vehicle yaw rate falls within $\pm r_{db}$, the front antiroll-bar control system will not be activated.

In this study, we select r_{db} to be 1.25 deg/sec. The feedback gain k_r was selected as follows: Let us assume that our objective is to increase the minimum tire loading (at the left rear tire) from 186 to 240 kg.wt. in a 0.75-g lateral maneuver. The required value of κ that can accomplish this goal is 0.462. From (5-7), it is clear that a $\kappa_c = 0.462$ could be achieved if we

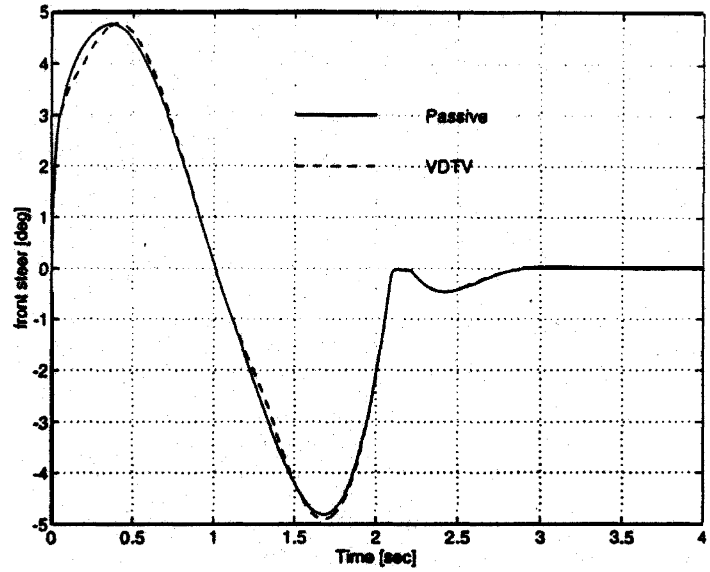


Figure 4: Front steer angle of VDTV and passive vehicles

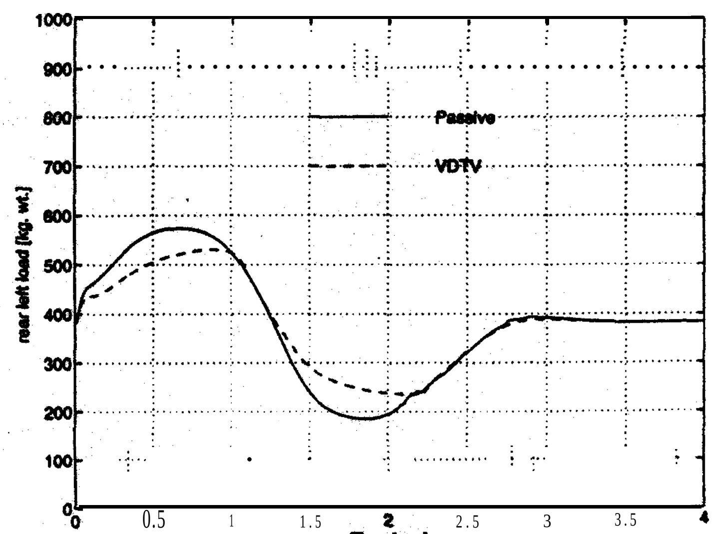


Figure 5: Left rear vehicle loadings

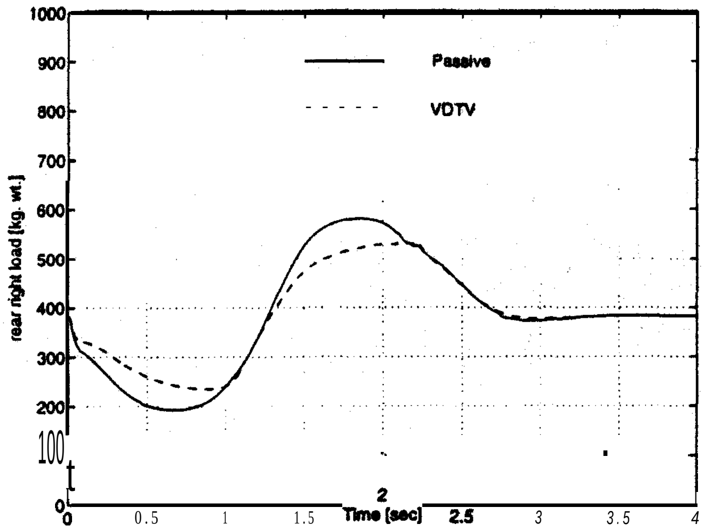


Figure 6: Right rear vehicle loadings

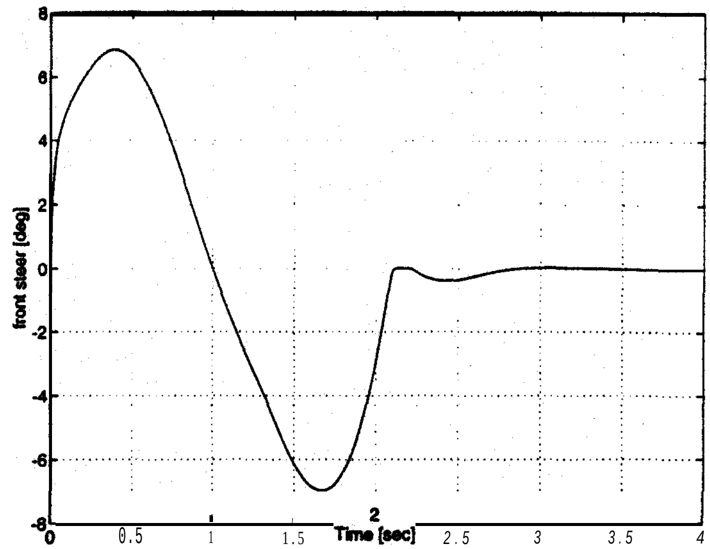


Figure 7: Front steering angle of VDTV

have $K_{fA} = 984.2$ Nm/deg and $K_{rA} = 0$ Nm/deg (i.e., without using the rear anti-roll bar actuator). The peak value of the vehicle's yaw rate for a 0.75-g maneuver is about 20 deg/sec. Hence, the value of k_r should be $984.2/(20-1.25)\times 57.3^2 \approx$ 172,340 Nm/rad per rad/s. This feedback gain is rounded to 200,000 Nm/rad per rad/s in this study.

The increased front antiroll bar stiffness will cause the damping ratio of the vehicle roll motion to drop. To maintam the same roll damping ratio, the front damper rates must be increased, via the adjustable dampers, by a factor that is proportional to the square root of the passive-to-active front roll stiffness ratio. The same is true for dampers at the rear axle.

$$\eta_f = \sqrt{\frac{K_f}{K_{fP}}} \tag{33}$$

$$\eta_r = \sqrt{\frac{K_r}{K_{rP}}} \tag{34}$$

where η_f and η_r are the damping rate ratios defined in (9,10).

The time history of the front steering command computed according to control algorithms (31-32) is given in Fii. 7. Relative to the steering command used by the passive vehicle (cf. Fig. 4), the one given in Fig. 7 is significantly larger. This is not unexpected because a vehicle's understeer coefficient typically increases with an increase in the front antiroll bar stiffness. Hence, a larger steering angle must be used to achieve the same level of lateral acceleration ($\approx \pm 0.75$ g).

We can use the VDTV's 4WS system to overcome problems associated with the increased vehicle understeer coefficient. In particular, out-of-phase steering of the rear wheel typically produces an oversteer effect that can be used to negate the understeer effect produced by the increased front antiroll bar stiffness. That is, whenever the front antiroll bar system is activated, we will simultaneously steer the rear wheels accordingly to the following relation: $\delta_{rc} = R \times \delta_{fc}$ Here, R denotes the front-to-rear wheel angle ratio, which is defined positive if the wheels *are* steered in phase. In this study, we use a ratio of -0.35 to produce an oversteer effect. Results obtained are given in Figs. 3-6 and 8.

The solid line in Fig. 8 depicts the time history of the load transfer distribution factor κ. The "static" value of κ is about +20%. This means that the front and rear axles carry about 60% and 40%, respectively, of the total load transferred in a lateral maneuver (κ is the difference between 60% and 40%). With the front antiroll bar activated, the peals value of κ is near 50%. That is, the front and rear axles, at the time when κ peaked, carry 75% and 25%, respectively, of the total load transfer. The reduced load transfer at the *rear* axle causes the minimum loading at the left rear tire to increase from 186 kg. wt. (for the passive vehicle) to 240 kg. wt. In this way the vehicle rollover resistance is improved.

7. Rollover resistance: How to lower it?

To study accidents involving a vehicle rollover, there might be a need to artificially lower the rollover resistance of a test vehicle. To this end, we need to increase the roll stiffness of the *rear* antiroll bar:

$$K^c_{rA} = \begin{cases} 0 & \text{if } |r| \leq r_{db} \\ k_r\,|\,r - \text{sgn}(r) r_{db}| & \text{otherwise} \end{cases} \tag{35}$$

$$K_{rA} = \min\{K^{mech}_{rA}, K^c_{rA}\} \tag{36}$$

Here, K^{mech}_{rA} denotes the mechanical limit of the rear antiroll bar stiffness, and r_{db} denotes a yaw rate deadband. The results given in the following were obtained with $r_{db} = 1.25$ deg/s and $k_r = 50{,}000$ Nm/rad per rad/s which were selected using an approach similar to that described in Section 6.

The increased rear antiroll bar stiffness causes the vehicle understeer coefficient to drop. To restore the understeer coefficient, *we* will steer the rear wheels in phase with the front wheels: $\delta_{rc} = R \times \delta_{fc}$. In this study, we use a ratio of +0.10 to produce the required understeer effect. The dashed line in Fig. 8 depicts the resultant time history of the load transfer

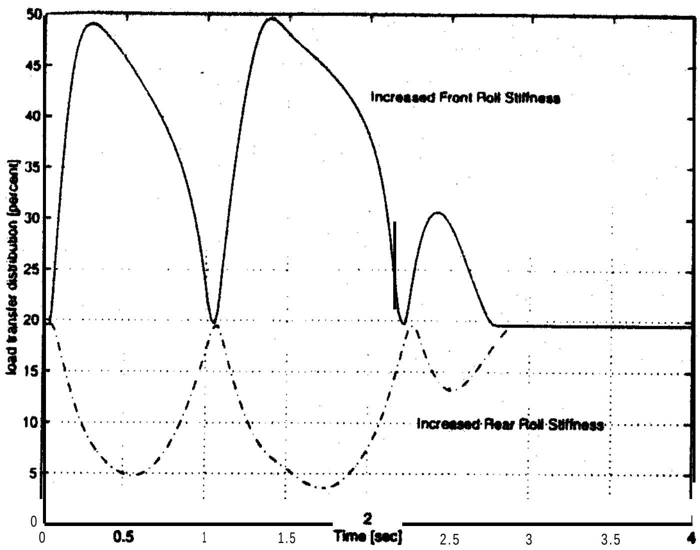


Figure 8: Load transfer distribution factor of the 4WS VDTV

distribution factor a. Here, we note that stiffening the rear antiroll bar system causes κ to drop to as low as 4%. That is, the front and rear axles, at the time when κ is at a local minimum, carry 52% and 48%, respectively, of the total load transfer. The larger load transfer at the rear axle causes the minimum loading at the rear left tire to drop from 186 kg. wt. (for the passive vehicle) to 140 kg. wt., degrading the vehicle rollover resistance.

8. Summary

The Variable Dynamics Testbed Vehicle has a set of active devices which could be used to advantage to alter the rollover tendency of a vehicle in lateral maneuvers, among other applications. In this research, we investigate how an increased front antiroll bar stiffness, in conjunction with an increased front damper rate and out-of-phase rear steering could improve vehicle rollover resistance in high-g lateral maneuvers. The approach taken suggests an effective way to enhance vehicle safety against rollover related accidents. Similar bat "reverse" algorithms were used to artificially degrade the rollover resistance of a vehicle. Rollover-related human factors issues could then be studied using such a vehicle. The effectiveness of early warning devices against vehicle rollover could also be evaluated using such a vehicle. The effectiveness of the control algorithms proposed in this study has been validated via comprehensive simulations. Results obtained from this study indicate that the variable dynamic vehicle comprises the right combination of active devices which makes it an effective testbed for performing rollover-related human factors research, among other applications.

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Disclaimer

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