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TWO STRATEGIES FOR REDUCING THE ROLLOVER RISK OF HEAVY VEHICLES

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Abstract

In this paper different control methods are proposed for the prevention of rollover of heavy vehicles. In the control structure either active anti-roll bars or an active brake mechanism is applied. Since the forward velocity of the vehicle changes in time, the combined yaw-roll dynamics of the vehicle has a nonlinear structure. Selecting the velocity as a scheduling parameter, a Linear Parameter Varying (LPV) model is constructed. The control design is also based on LPV method, in which both the performance specifications and the model uncertainties are taken into consideration. In the paper the control solutions are demonstrated, the different control methods are analysed and compared with each other. The operation of the control mechanisms are demonstrated in a cornering and a double lane change maneuver.

Keywords: linear parameter varying control, nonlinear modelling, robustness, uncertainty, vehicle dynamics, active anti-roll bars, active brake.

1. Introduction

The aim of the rollover prevention is to provide the vehicle with the ability to resist overturning moments generated during maneuvers. The problem with heavy vehicles is a relatively high mass center and narrow track width. In the literature there are many papers with different approaches on the active roll control of the heavy vehicles, e.g. [1, 3, 7, 5]. In this paper two strategies for reducing the rollover risk are developed. The active anti-roll bars apply a pair of hydraulic actuators in the suspension system. They generate a stabilizing moment to balance the overturning moment caused by lateral acceleration. The active brake mechanism applies unilateral brake forces to each of the wheels. When a wheel lift-off is detected, unilateral braking forces are generated to reduce the lateral tire forces acting on the wheel.

In the paper the control solutions are analysed and compared with each other. The yaw-roll dynamics of vehicles in which controllers are applied are nonlinear with respect to the forward velocity. The velocity is handled as a scheduling parameter and an LPV model is formalized. It allows us to take into consideration the nonlinear effect in the state space description. The controller based on this LPV model is adjusted continuously by measuring the vehicle velocity in real-time.

The structure of the paper is as follows: In Section 2 the combined yaw-roll model in which the forward velocity changes in time is constructed. Section 3 presents the control design, in which either active anti-roll bars or an active brake mechanism is applied. Section 4 demonstrates the results of the control design. Finally, Section 5 contains some concluding remarks.

2. Nonlinear Model of the Yaw-roll Dynamics

Fig. 1 illustrates the combined yaw-roll dynamics of the vehicle, which is modelled by a three-body system, in which m_s is the sprung mass, $m_{u,f}$ is the unsprung mass at the front including the front wheels and axle, and $m_{u,r}$ is the unsprung mass at the rear with the rear wheels and axle, and m is the total vehicle mass. I_{xx}, I_{xz}, I_{zz} are the roll moment of the inertia of the sprung mass, the yaw-roll product, and the yaw moment of inertia, respectively. The signals are the lateral acceleration a_y, the side slip angle of the sprung mass β, the heading angle ψ, the yaw rate $\dot{\psi}$, the roll angle ϕ, the roll rate $\dot{\phi}$, the roll angle of the unsprung mass at the front axle $\phi_{t,f}$ and at the rear axle $\phi_{t,r}$. δ_f is the front wheel steering angle. v is the forward velocity. The total axle loads are F_{zl} and F_{zr}. The roll motion of the sprung mass is damped by suspensions with damping coefficients b_f, b_r and stiffness coefficients k_f, k_r. The tire stiffnesses are denoted by $k_{t,f}$, $k_{t,r}$. h is the height of CG of sprung mass and $h_{u,f}$, $h_{u,r}$ are the heights of CG of unsprung masses and r is the height of roll axis from ground.

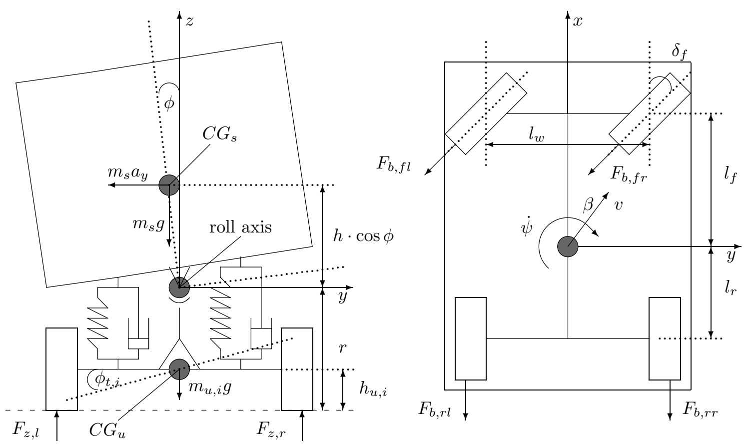

Fig. 1. Rollover vehicle model

In the following the motion differential equations of the yaw-roll dynamics of a single unit vehicle are formalized. The first equation considers forces for lateral dynamics, the other equations are the torque balance equations for the yaw and roll moments.

$$mv(\dot\beta+\dot\psi)-m_sh\ddot\phi = F_{y,f}+F_{y,r} \tag{1a}$$
$$-I_{xz}\ddot\phi + I_{zz}\ddot\psi = F_{y,f}l_f - F_{y,r}l_r + l_w\Delta F_b \tag{1b}$$
$$\begin{aligned}\left(I_{xx}+m_sh^2\right)\ddot\phi - I_{xz}\ddot\psi &= m_sgh\phi + m_svh(\dot\beta+\dot\psi) - k_f(\phi-\phi_{t,f})\\ &\quad - b_f(\dot\phi-\dot\phi_{t,f}) - k_r(\phi-\phi_{t,r}) - b_r(\dot\phi-\dot\phi_{t,r})\\ &\quad + u_f + u_r\end{aligned} \tag{1c}$$
$$\begin{aligned}-h_rF_{y,f} &= m_{u,f}v(h_r-h_{u,f})(\dot\beta+\dot\psi) + m_{u,f}gh_{u,f}\phi_{t,f} - k_{t,f}\phi_{t,f}\\ &\quad + k_f(\phi-\phi_{t,f}) + b_f(\dot\phi-\dot\phi_{t,f}) + u_f\end{aligned} \tag{1d}$$
$$\begin{aligned}-h_rF_{y,r} &= m_{u,r}v(h_r-h_{u,r})(\dot\beta+\dot\psi) - m_{u,r}gh_{u,r}\phi_{t,r} - k_{t,r}\phi_{t,r}\\ &\quad + k_r(\phi-\phi_{t,r}) + b_r(\dot\phi-\dot\phi_{t,r}) + u_r\end{aligned} \tag{1e}$$

These equations include the influence both of the active anti-roll bars and the active brake. However, in this paper we propose control mechanisms applying either active anti-roll bars or an active brake. The active anti-roll bars generate a roll moment between the sprung and unsprung mass, u_{af} and u_{ar}. If active anti-roll bars are applied, the last component of equation (1b) is missing. The active brake generates brake forces, which are considered in the paper as a difference in brake forces between the left and right-hand side of the vehicle ΔF_b. In this case the u_f and u_r are missing in the equations (1c,1d,1e).

These equations can be expressed in a state space representation form:

$$\dot x = A(v)x + B_1(v)\delta_f + B_2(v)u \tag{2}$$

where the state vector is $x=\begin{bmatrix}\beta & \dot\psi & \phi & \dot\phi & \phi_{t,f} & \phi_{t,r}\end{bmatrix}^T$. δ_f is the front wheel steering angle, which is considered as a disturbance signal. In case of active anti-roll bars the control inputs are roll moments between the sprung and unsprung mass. $u=\begin{bmatrix}u_f & u_r\end{bmatrix}^T$. In this paper the actuator dynamics is not taken into consideration. In case of an active brake the control input is $u = \Delta F_b$. In practice ΔF_b is generated by a sharing logic of the brake forces. The reason for sharing the control force between the front and rear wheels is to minimize the wear of the tires.

In *Eq.* (2) matrix $A(v)$ depends on the forward velocity of the vehicle nonlinearly. Selected the forward velocity as scheduling parameter an LPV model can be formalized. The idea behind using LPV systems is to take advantage of the casual knowledge of the dynamics of the system, see [2, 6]. One of the characteristics of the LPV system is that it must be linear in the pair formed by the state vector x, and the control input vector u. The matrices A and B are generally nonlinear functions of the scheduling vector ρ. If v is chosen as a scheduling parameter, the differential equations of the yaw-roll motion are linear in the state variables: $\rho = v$.

3. Control Design Based on an LPV Method

The objective of the roll control system is to increase the roll stability of the vehicle. The rollover is caused by the high lateral inertial force generated by lateral acceleration. The roll stability of the vehicle is determined by the ability of the vehicle to generate a stabilizing moment to balance the overturning moment caused by lateral acceleration. The roll-over situation can be detected if the lateral load transfers for both axles are monitored. The lateral load transfer can be given: $\Delta F_{z,i} = 2\frac{k_{t,i}\phi_{t,i}}{l_w}$, where the subscript i denotes the front and rear axles. The lateral load transfer can be normalized in such a way that the load transfer is divided by the total axle load: $R_i = \frac{\Delta F_{z,i}}{F_{z,i}}$, where the $F_{z,i}$ is the total axle load. The normalized load transfer R_i value corresponds to the largest possible load transfer. If $|R_i|$ exceeds 1, the inner wheels in the bend lift-off. The roll stability achieved by limiting the lateral load transfers to below the levels required for wheel lift off. Thus, the normalized lateral load transfer at the front and the rear are selected as performance outputs.

In this paper two controlled systems are designed, and the controlled system which uses active anti-roll bars is compared with the one which uses an active brake mechanism. An important difference between these solutions is the following. The active anti-roll bars are active all the time and they generate stabilizing moment if any destabilizing moment is created by vehicle maneuvers. However, the active brake is activated only when the vehicle comes close to rolling over. In a normal cruising (driving) situation the active brake mechanism should not be activated. However, in a critical situation the brake mechanism must be activated.

The input signals are selected performance outputs, since the actuator saturation must be avoided. In the design of the active brake system, the lateral acceleration is also selected as a performance signal, since the active brake influences the lateral acceleration directly. The vector of the performance signals are the following: $z = \begin{bmatrix} a_y & \Delta F_b \end{bmatrix}^T$. Note, that in the case of active anti-roll bars, the performance vector includes the axle loads, however, it does not include a_y. $z = \begin{bmatrix} a_y & \Delta F_{z,f} & \Delta F_{z,r} & u_f & u_r \end{bmatrix}^T$. The measured outputs are the lateral acceleration and the roll rate of the sprung mass: $y = \begin{bmatrix} a_y & \dot{\phi} \end{bmatrix}^T$.

The closed-loop interconnection structure includes the feedback structure of the model $G(\rho)$ and controller $K(\rho)$, and elements associated with the uncertainty models and performance objectives.

First the selection of the performance weighting functions in the design of the active anti-roll bars is shown. The purpose of the selection is to keep the lateral load transfers and control inputs small. The weighting function for lateral load transfer is $W_{p_{F_z}} = \mathrm{diag}(1/10^2, 1/10^3)$, what means that the maximal gain of the lateral load transfers can be 10^2 for the front axle and 10^3 for the rear axle. In the design

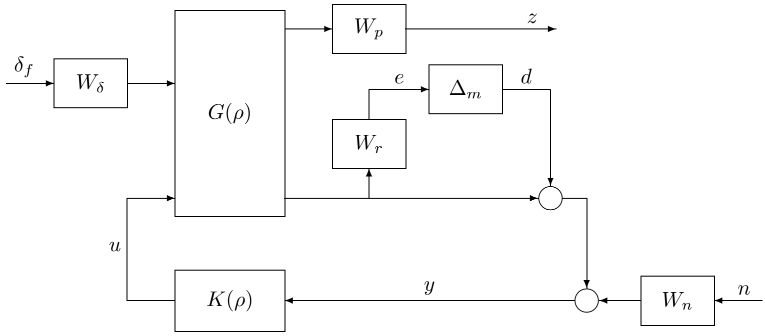

Fig. 2. The closed-loop interconnection structure for control design

of active anti-roll bars W_{p_u} is a diagonal matrix with diagonal entries $1/20$, which corresponds to the front and rear control torque generated by active anti-roll bars.

Second, the selection of the weighting functions is shown in the design of the active brake. A weighting function for the lateral acceleration is $W_{p_a} = \phi_a \frac{(s/2000+1)}{(s/12+1)}$, in which the role of the factor ϕ_a is to minimize the influence of the acceleration in steady state. Since the brake system must be activated only in a critical situation, the normalized load transfer must be monitored. The weighting for the active brake is selected $W_{p_{Fb}} = 1 \cdot 10^3$, which corresponds to the maximal gain of the brake force difference.

The factor ϕ_a is chosen to be parameter-dependent, i.e. the function of the normalized load transfer. When the vehicle is not in an emergency, i.e. $|R| < R_1$, $\phi_a(R)$ is zero. When R approaches the critical value, $\phi_a(R) = \frac{|R|-R_1}{R_2-R_1}$ increases. When $|R| > R_2$ exceeds a critical value, $\phi_a(R)$ is one. Here, the parameter dependence of the gain is characterized by the constants R_1 and R_2. R_1 defines a status when the vehicle tends to rolling over. The closer R_1 to 1, the later the controller will be activated. R_2 defines the critical value, when the controller should focus on minimizing the lateral acceleration. In the control design the constants are selected as $R_1 = 0.85$ and $R_2 = 0.95$.

The uncertainties of the model are represented by $W_r = 2.25\frac{S+20}{S+450}$. The disturbance w includes the steering angle and the sensor noises: $w = \begin{bmatrix}\delta_f & n\end{bmatrix}^T$. The input weight W_δ normalizes the steering angle to the maximum expected command. It is selected as $5\pi/180$, which corresponds to 5 degrees of steering angle command. The noise weight W_n is selected as a diagonal matrix. The weight for the lateral acceleration is chosen 0.01 m/s^2 and the weight for the roll rate is chosen 0.01 deg/sec.

In order to describe the control objective, the parameter-dependent augmented plant $P(\varrho)$ must be built up using the closed-loop interconnection structure. These augmented plants include the parameter-dependent vehicle dynamics and the weighting functions, which are defined above.

$$\begin{bmatrix} \tilde{z} \\ \hline y \end{bmatrix} = \left[\begin{array}{c|c} P_{11}(\varrho) & P_{12}(\varrho) \\ \hline P_{21}(\varrho) & P_{22}(\varrho) \end{array}\right] \begin{bmatrix} w \\ \hline u \end{bmatrix}, \tag{3}$$

where $w = \begin{bmatrix} d_a & d_\phi & \delta_f & n_a & n_\phi \end{bmatrix}$, $\tilde{z}_r = \begin{bmatrix} e_a & e_\phi & z \end{bmatrix}$. In the LPV model of $P(\varrho)$ two parameters are selected: the forward velocity v and the normalized lateral load transfer at the rear side R_r, i.e. $\varrho = \begin{bmatrix} v & R_r \end{bmatrix}$. Here v is measured directly, and parameters R_r are calculated by using the measured roll angle $\phi_{t,r}$.

The closed-loop system $M(\varrho)$ is given by a lower linear fractional transformation (LFT) structure:

$$M(\varrho) = \mathcal{F}_\ell(P(\varrho), K(\varrho)), \tag{4}$$

where $K(\varrho)$ depends on the scheduling parameter ϱ. The goal of the control design is to minimize the induced $\mathcal{L}_2$ norm of a LPV system $M(\varrho)$, with zero initial conditions, which is given by

$$\|M_r(\varrho)\|_\infty = \sup_{\varrho \in \mathcal{F}_\mathcal{P}} \sup_{\|w\|_2 \neq 0, w \in \mathcal{L}_2} \frac{\|\tilde{z}\|_2}{\|w\|_2} \tag{5}$$

The control of LPV systems with induced L_2-norm performance is proposed by several authors, see [2, 4, 8, 9].

4. Experimental Results

In the demonstration example, the controlled system which uses active anti-roll bars is compared with the controlled system which uses an active brake mechanism. Note, that in the simulation example the sharing logic of the active brake is the implementation in the following way: $\Delta F_b = (F_{b,rl} + d_2 F_{b,fl}) - (F_{b,rr} + d_1 F_{b,fr})$, where d_1 and d_2 can be calculated by geometry data. It is assumed, that the driver does not push down on the brake pedal, hence the only change in forward velocity is caused by the compensator. Two vehicle maneuvers, a cornering and a double lane change maneuver, are illustrated. The velocity of the vehicle is 75 kph.

The first example illustrates the vehicle dynamics in a cornering maneuver in *Fig. 3*. The solid line illustrates the controlled system using an active brake control, while the dashed line illustrates the active anti-roll bars. The figures show the lateral acceleration a_y, the roll angle of the sprung mass ϕ, the normalized load transfers at the front and rear R_f, R_r, the brake forces at the wheels F_{brf}, F_{brr} and the moments of the active anti-roll bars u_f, u_r.

As the lateral acceleration increases, the normalized load transfers also increase, and they lift up the rear axle. Since the active anti-roll bars are active all the time, they generate a stabilizing moment to balance the destabilizing moment. Approximately 200 kNm control torque is required for the rear axle during this

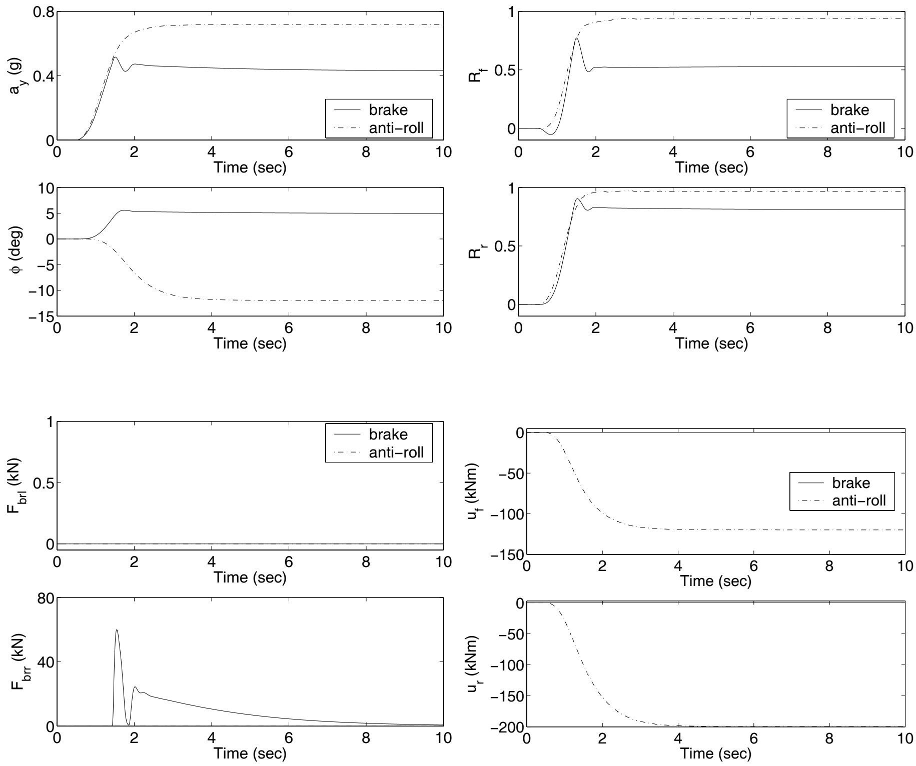

Fig. 3. Time responses cornering manoeuver

manoeuver. This moment results in a -180 degree phase shifting in the roll angle of the sprung mass, and it decreases the normalized lateral load transfers. The active brake mechanism is not activated until the normalized load transfers have exceeded the critical value, which is 0.85 in this example. The lateral acceleration is the same as in the active anti-roll case as long as it is below the critical value of the normalized load transfers. When the normalized load transfer reaches its critical value (at 1.4 sec), a braking force at the right-hand-side is created, which is about 60 kN. It results in decreasing the normalized load transfers, and the compensator also decreases the velocity, i.e. the forward velocity is not constant.

In the second example the active anti-roll bars are compared with an active brake in the double lane change maneuver. The vehicle dynamics is illustrated in *Fig. 4*. The second driving maneuver is more critical than the first one, the active brake is only activated in the second maneuver (at 2.5 sec) and decreases the lateral acceleration and so the normalized load transfers. The braking force required at the left-hand-side wheel is about 60 kN. The active anti-roll bars operate all the time, and the maximal control torque required for the rear axle is about 100 kNm.

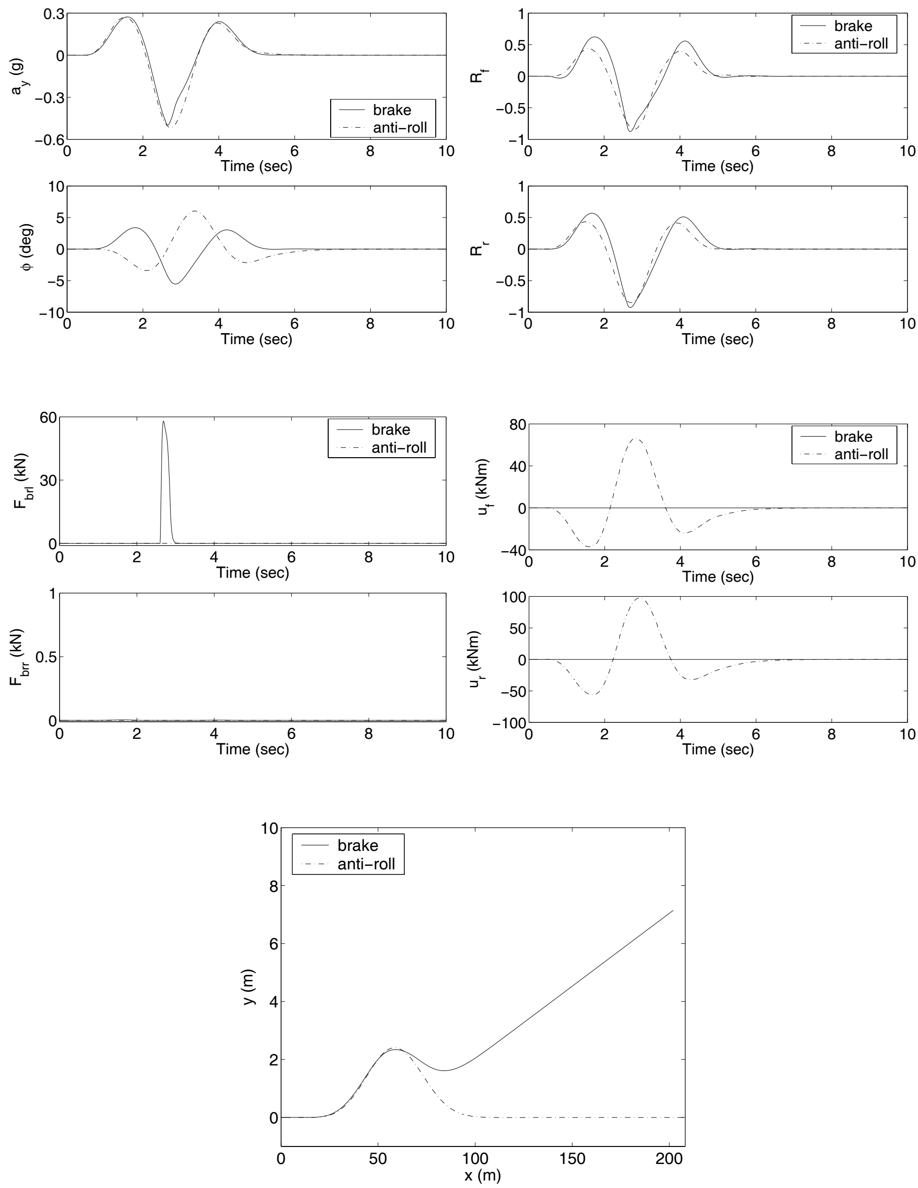

Fig. 4. Time responses to double lane change steering input

Fig. 4 also illustrates the displacement of the vehicle. In the case of active anti-roll bars the vehicle keeps the desired path required by the driver. In the case of the brake control the real path is significantly different from the desired path due to the brake moment which affects the yaw motion. This is a disadvantage of the active brake, since this maneuver requires the drivers intervention.

5. Conclusions

In this paper the controlled system using active anti-roll bars is compared with the system using an active brake mechanism. The control design is based on the LPV method, in which both the performance specifications and the uncertainties are taken into consideration. The active anti-roll bars generate a stabilizing moment to balance the destabilizing moment. Since they do not influence the lateral acceleration, the active anti-roll bars operate all the time. The active brake mechanism influences the lateral acceleration directly. Since the active brake is activated only when the vehicle comes close to rolling over, the normalized load transfers must be monitored.

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