

A Computer Simulation Study on Lateral Dynamics of Automated Guideway Transit Vehicle

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Abstract

A computer simulation study on lateral motion of the AGT vehicle running on a straight or curved guideway with and without guidewall irregularities has been carried out to analyze the effects of steering parameters on the vehicle lateral motion and deal with the appropriate design of mechanical guidance system which could accomplish acceptable lateral stability compatible with insensitivity to disturbances from the guidewall.

It has been found out conclusively that some additional restoring moments with respect to kingpins in the steering systems have favorable effects on the vehicle lateral stability, while leaving the vehicle insensitive to the guidewall irregularities, which has been contributive to the development practice of designing the optimum steering control system of the AGT vehicles under the real operation.

1. Introduction

The AGT vehicles, which are operated under automatic longitudinal and lateral control on exclusive guideway have mechanical guidance systems which directly sense lateral position and directional errors with respect to guideway sidewall (guidewall) and steer four-wheels (front and rear) through Ackerman steering link to track the guideway automatically. One of the particular interests for these vehicles relates to the steering control stability and to reducing disturbances from the guidewall to maximize the lateral ride quality, and numerous studies on the effects of vehicle and steering parameters on the steering performance and the lateral ride quality have been made to contribute to the development practice of AGT vehicles. The effects of fundamental vehicle and steering parameters such as vehicle steering gains on the steering performance have been considered, making use of linearized vehicle dynamic models. From practical view points, steering parameters such as the clearance between the guidewheel and the guidewall, the radial stiffness of the guidewheel and the pre-compression of the elastic guidewheel should be also considered to have significant effects on the vehicle lateral motion.

2. Nomenclature

β : vehicle side slip angle

ϕ : vehicle yaw angle

- ϕ : vehicle body roll angle
 δ_F, δ_R : front and rear wheel steering angles
 y_{GF}, y_{GR} : front and rear guidebar lateral displacements with respect to vehicle body
 ξ_F, ξ_R : front and rear guidebar displacements with respect to guidewall
 X, Y : position with respect to co-ordinate axis fixed on the earth
 S : vehicle travel distance measured along guideway
 W : vehicle weight (16000 kg)
 I : vehicle yaw moment of inertia (4400 kgms²)
 C_F, C_R : front and rear wheel cornering powers (28000 kg/rad)
 l : one-half wheel-base (2.5 m)
 W_S : sprung body weight (14600 kg)
 I_ϕ : sprung body roll moment of inertia (2700 kgms²)
 C_ϕ : roll damping coefficient (5000 kgms)
 K_ϕ : roll stiffness (5000 kgm/rad)
 r_S : distance from roll axis to sprung body center of gravity (0.71 m)
 r_F : distance from roll axis to vehicle floor (0.33 m)
 V : vehicle forward velocity
 I_S : lumped yaw moment of inertia of wheels and steering link with respect to kingpin (4.8 kgms²)
 C_S : steering system damping coefficient (150 kgms)
 A_F, A_R : front and rear wheel self-aligning powers (700 kgm/rad)
 l_S : one-half distance between guidebars (3.16 m)
 l_{ST} : one-half distance between guidebar mechanical stops (3.3 m)
 a_F, a_R : front and rear equivalent steering lever length
 b, c, d, d_F, d_R, e : steering link coupling radii
 K_G : steering link stiffness (50000 kg/m)
 ϵ_{KG} : one-half idle clearance between steering link and guidebar (0.0 m)
 W_G : lumped weight of a guidebar and guidewheels (190 kg)
 K_{GW} : nonlinear radial stiffness of guidewheel (9500 kg/cm², 350 kg/cm²)
 ϵ : one-half clearance between guidewheel and guidewall

Note that when ϵ is negative, it means no more clearance but radial pre-compression of guidewheel.

- K_{SS1} : rod stiffness (50000 kg/m)
 K_{SS2} : preloaded spring stiffness (1500 kg/m)
 P_0 : preload
 K_{SF} : stiffness of rod with solid friction (50000 kg/m)
 F_0 : solid friction force
 λ : unit rail span (10.0 m)
 θ : track cant angle

3. Vehicle Dynamic Model

The co-ordinate axis and variables as shown in Fig. 1 are adopted to describe the vehicle lateral motion with respect to guideway and following expressions are obtained as kinematic relations :

$$\dot{X} = V \cos (\beta + \psi) \quad (1)$$

$$\dot{Y} = V \sin (\beta + \psi) \quad (2)$$

$$\dot{S} = V \quad (3)$$

Here, angle, β , means side slip angle of roll axis, as roll motion of the vehicle body is taken into account which results in three degree-of-freedom model of the vehicle.

The mechanical guidance system consists of two guidewheels, a guidebar, a steering link and two wheels. The guidebar with massless guidewheels, which is connected to the steering link through the modeled spring, is allowed to move laterally with respect to the vehicle body. Furthermore, it contacts the guidewall through the guidewheel. The front and rear wheels with steering links are allowed to rotate about respective kingpins. Hence the lateral motions of the front and rear guidebars and the wheel rotational motions constitute 4 degree-of-freedom and a total of 7 degree-of-freedom dynamic model is yielded as shown in Fig. 2.

Here, two types of the mechanical guidance systems with nearly rigid and fairly elastic guidewheels respectively both of which have nonlinear spring characteristics in their radial stiffness are considered and the particular interest should be focused on the following concepts of each type of mechanical guidance system. In the case of the guidance system with the rigid guidewheels, the minimum dimension between the guidewalls on both sides of the guideway should be limited for the guidebar to be able to pass through between the both side guidewalls. Furthermore, as the construction precision of the dimension can't be so severe in practice, the clearance should be allowed between the guidewheel and guidewall. (Fig. 3 (a)) On the other hand, the elastic guidewheels can be allowed to have a pre-compres-

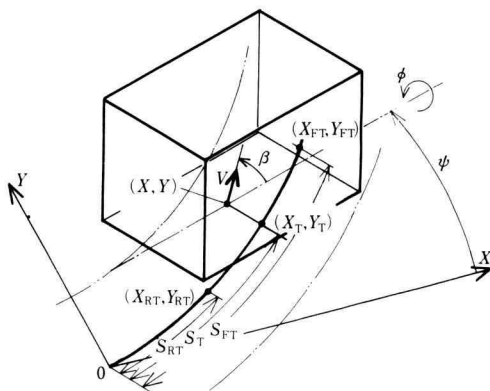


Fig. 1 co-ordinate axis and variables

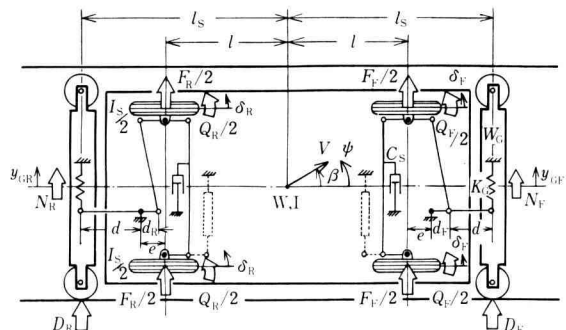


Fig. 2 vehicle dynamic model

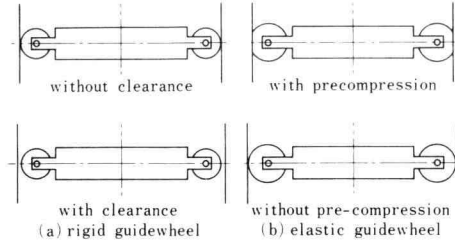


Fig. 3 two cases of guidewheel

sion in order to eliminate the clearance. However, the precompression is not always left constant because of the precision limit of the dimension between the both side guidewalls. Sometimes the dimension is equal to the free length of the guidebar with guidewheels and the pre-compression disappears. (Fig. 3 (b))

As a fundamental consideration suggests that the existence of the clearance or reducing the guidewheel stiffness both of which cause the time lag of the response of the guidance system results in deteriorations of the steering control stability²⁾, particular attentions should be paid on the vehicle motions with the above mentioned clearance or the elastic guidewheel without the pre-compression.

4. External Forces and Moments

One of the important assumption, here, is that the lateral dimension effect of the vehicle is neglected and all mathematical expressions are dealt with on the center-line of the vehicle.

4.1 Cornering Forces

For the sake of safty and ride comfort, it is desirable for lateral acceleration of the vehicle to be limited to less than 0.3 g and the vehicle side-slip yaw velocity and wheel steering angles would be small enough to be able to assume that the cornering force is proportional to wheel side-slip angle. Thus, following expressions are obtained for the forces which act on the front and rear wheels in lateral direction :

$$F_F = C_F(\delta - \beta - l\dot{\psi}/V) \quad (4)$$

$$F_R = C_R(\delta - \beta + l\dot{\psi}/V) \quad (5)$$

4.2 Reaction Forces from Guidewall

In the case of the guidance system with rigid guidewheels, the outline of the reaction forces against the guidebar displacement with respect to the guidewall are as shown in Fig. 4 and the mathematical expressions for the forces are as follows :

$$\begin{aligned} D_F &= -K_{GW}(|\xi_F| - \epsilon)^2 \text{sign}(\xi_F) \cdots (|\xi_F| \geq \epsilon) \\ &= 0 \cdots (|\xi_F| < \epsilon) \end{aligned} \quad (6)$$

$$\begin{aligned} D_R &= -K_{GW}(|\xi_R| - \epsilon)^2 \text{sign}(\xi_R) \cdots (|\xi_R| \geq \epsilon) \\ &= 0 \cdots (|\xi_R| < \epsilon) \end{aligned} \quad (7)$$

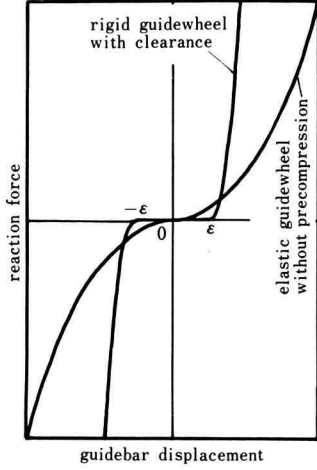


Fig. 4 reaction force from guidewall

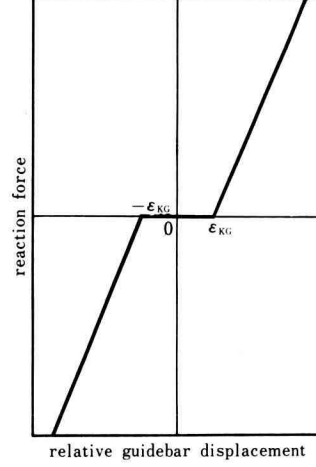


Fig. 5 reaction force of steering link spring

where, some vector analysis yields following expression :

$$\xi_F = \frac{1}{V} [\dot{X}_{TF}(Y_{GF} - Y_{TF}) - \dot{Y}_{TF}(X_{GF} - X_{TF})] \quad (8)$$

$$\xi_R = \frac{1}{V} [\dot{X}_{TR}(Y_{GR} - Y_{TR}) - \dot{Y}_{TR}(X_{GR} - X_{TR})] \quad (9)$$

further, X_{GF} , Y_{GF} , X_{GR} and Y_{GR} are given as following forms :

$$\begin{aligned} X_{GF} &= X + l_s \cos \phi - y_{GF} \sin \phi \\ Y_{GF} &= Y + l_s \sin \phi + y_{GF} \cos \phi \\ X_{GR} &= X - l_s \cos \phi - y_{GR} \sin \phi \\ Y_{GR} &= Y - l_s \sin \phi + y_{GR} \cos \phi \end{aligned} \quad (10)$$

The outline of the reaction forces in the case of the guidance system with the elastic guidewheels are also shown in Fig. 4 and mathematically expressed as follows :

$$\begin{aligned} D_F &= -K_{GW}(|\xi_F| - \epsilon)^2 \text{sign}(\xi_F) \cdots (|\xi_F| \geq \epsilon) \\ &= -4K_{GW}|\epsilon| \xi_F \cdots (|\xi_F| < \epsilon) \end{aligned} \quad (6')$$

$$\begin{aligned} D_R &= -K_{GW}(|\xi_R| - \epsilon)^2 \text{sign}(\xi_R) \cdots (|\xi_R| \geq \epsilon) \\ &= -4K_{GW}|\epsilon| \xi_R \cdots (|\xi_R| < \epsilon) \end{aligned} \quad (7')$$

where ϵ is negative and means no more the clearance but the initial deformation of the guidewheel caused by the radial pre-compression.

4.3 Reaction Forces of the Steering Link Spring

The reaction forces of the modeled spring of the steering link act on the guidebars and the wheels as restoring forces and moments respectively. Also, they act on the vehicle body as lateral forces and yaw moments through the support points of the steering link fixed on the body. The lateral displacements of the guidebars and the steering links at their connecting points with respect to the vehicle body are expressed by (y_{GF}, y_{GR}) and $(a_F \delta_F, a_R \delta_R)$ respectively resulting in the following expression for the reaction forces :

$$N_F = -K_G(|a_F\delta_F - y_{GF}| - \varepsilon_{KG}) \text{sign}(a_F\delta_F - y_{GF}) \cdots (|a_F\delta_F - y_{GF}| \geq \varepsilon_{KG}) \\ = 0 \cdots (|a_F\delta_F - y_{GF}| < \varepsilon_{KG}) \quad (11)$$

$$N_R = -K_G(|a_R\delta_R - y_{GR}| - \varepsilon_{KG}) \text{sign}(a_R\delta_R - y_{GR}) \cdots (|a_R\delta_R - y_{GR}| \geq \varepsilon_{KG}) \\ = 0 \cdots (|a_R\delta_R - y_{GR}| < \varepsilon_{KG}) \quad (12)$$

which are shown diagrammatically in Fig. 5. Here, ε_{KG} denotes the idle clearance between the guidebar and steering link, which is usually negligible.

4.4 Self-Aligning Moments

Under the assumption that self-aligning moments is proportional to wheel sideslip angle, following expression for the moments which act on the front and rear wheels about the respective kingpins are obtained :

$$Q_F = -A_F(\delta_F - \beta - l\dot{\psi}/V) \quad (13)$$

$$Q_R = -A_R(\delta_R - \beta + l\dot{\psi}/V) \quad (14)$$

5. Equations of Motion

Following equations of motion of vehicle side-slip, yaw and roll are obtained under the assumption that the the vehicle forward velocity is constant :

$$\frac{W}{g} V(\dot{\beta} + \dot{\psi}) - \frac{W_S}{g} r_S \dot{\phi} = F_F + F_R + N_F + N_R + W\theta \quad (15)$$

$$I\ddot{\psi} = lF_F - lF_R + (l + e - \frac{d}{d_F}e)N_F - (l + e - \frac{d}{d_R}e)N_R \quad (16)$$

$$-\frac{W_S}{g} r_S V(\dot{\beta} + \dot{\psi}) + I\ddot{\phi} + C_\phi \dot{\phi} + K_\phi(\phi - \theta) = r_S W_S \phi \quad (17)$$

where the lateral component of gravitational force of the vehicle, $W\theta$, caused by the track cant, θ , and gravitational roll moment $r_S W_S \phi$ caused by roll itself are taken into account as external force and moment. Referring to Fig. 2, Newton's second law of motion yields following forms of equation-of-motion for the lateral motions of the guidebars and the rotational ones of the steering system with respect to the vehicle body :

$$\frac{W_G}{g} \ddot{y}_{GF} = D_F - N_F + W_G \theta - \frac{W_G}{g} V \left(\dot{\beta} + \dot{\psi} + \frac{l_S}{V} \ddot{\psi} \right) \quad (18)$$

$$\frac{W_G}{g} \ddot{y}_{GR} = D_R - N_R + W_G \theta - \frac{W_G}{g} V \left(\dot{\beta} + \dot{\psi} - \frac{l_S}{V} \ddot{\psi} \right) \quad (19)$$

$$I_S \ddot{\delta}_F + C_S \dot{\delta}_F = a_F N_F + Q_F - I_S \ddot{\psi} \quad (20)$$

$$I_S \ddot{\delta}_R + C_S \dot{\delta}_R = a_R N_R + Q_R - I_S \ddot{\psi} \quad (21)$$

where taking account of D'Alembert's principle the inertia forces and moments are added to external forces and moments, and also gravitational forces caused by the track cant are taken into account.

6. Computer Simulation

6.1 Framework of the Simulation

The lateral motions of the vehicle running along straight or curved guideway with and without guidewall irregularities have been simulated by solving the equations of motion numerically. For computational purpose, it is convenient for the equations of motion, (15)–(21), to be transformed to following forms of first order differential equations:

$$\dot{\mathbf{X}} = f(\mathbf{X}) \quad (22)$$

where \mathbf{X} is vector of system state variables. In this study, Runge-Kutta-Gill-Method has been adopted to solve the equations of motion for the simulation. The computer simulation program written in FORTRAN of about 700 steps consist of a main program and six subroutine subprograms named HENKAN, UHEN, OUTPUT, PLOTLP, ORESEN and IRCURV. The basic structure and functions of the programs are as follows. The main program supervises the simulation and executes solving the equations of motion by the routine of Runge-Kutta-Gill-Method, making use of following subprograms.

- HENKAN : transform variables, \mathbf{X} , in first order differential equations to variables in original equations of motion
- UHEN : calculate $f(\mathbf{X})$ to predict new system state variables
- OUTPUT : calculate output variables
- PLOTLP : print out output variables
- ORESEN : generate straight guideway position
- IRCURV : generate curved guideway position

6.2 System Inputs and Outputs

The position of the modeled guideway, (X_T, Y_T) , (X_{FT}, Y_{FT}) and (X_{RT}, Y_{RT}) as shown in Fig.6 are considered to be the system inputs in the simulation and should be given as functions of the vehicle traveling distance S , S_F and S_R respectively. The model of

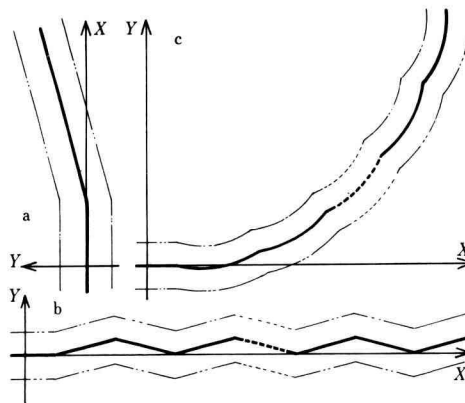


Fig. 6 modeled guideway

connected straight guideways is shown in Fig. 6 (a). The simulation of the vehicle motion subsequent to sudden direction change at the connecting point on this guideway has been executed to study especially on vehicle stability.

As the guidewall is usually composed of the connection of unit rails with the same span, the guidewall irregularities are assumed to be periodical which is due to the span of the unit rail. Therefore, the straight guideway with guidewall irregularities is, here, supposed to be the connections of the straight unit rails with directional error and the curved guideway with guidewall irregularities is supposed to be the connection of the curved unit rails with sinusoidally varying error of radius of curvature. These guideways are modeled as in Figs. 6 (b) and (c) respectively. The simulation of the vehicle motions on those guideways has been executed to study on the lateral ride quality under guidewall irregularities.

For the purpose of studying the vehicle lateral stability and ride quality, the attention is paid on the lateral accelerations at the vehicle floor above the front and rear axles, which are selected as system output variables in the simulation.

7. Effect of System Parameters on Vehicle Steering Performance

7.1 Effect of Clearance between Guidewall and Guidewheel

The simulation of the vehicle lateral motion on the connected straight guideway has been carried out to study on vehicle transient responses. In the simulated vehicle motions, the lateral acceleration shows larger amplitude and more oscillatory aspects at the rear part of the vehicle than at the front. Therefore, the attention is focused on the lateral acceleration on vehicle floor above the rear axle. The lateral acceleration responses of the vehicle with rigid guidewheel guidance systems are shown in Fig. 7 (a). It shows that the clearance causes the deterioration of the vehicle lateral stability.

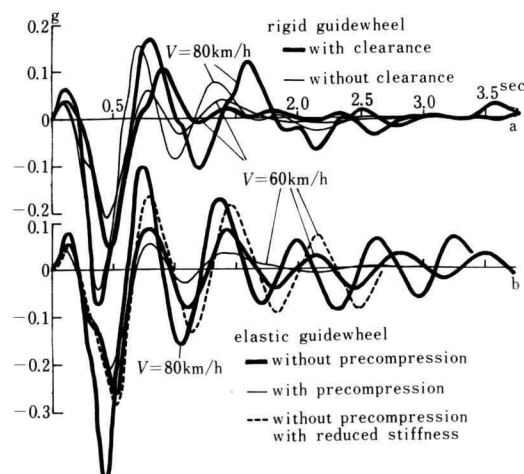


Fig. 7 vehicle lateral acceleration response

7.2 Effect of Guidewheel Pre-compression and Stiffness

The lateral acceleration responses of the vehicle with the elastic guidewheel guidance systems are shown in Fig. 7 (b). It shows that the elimination of the guidewheel pre-compression results in the serious deterioration of the vehicle stability, which is more remarkable when the guidewheel stiffness is reduced.

8. Improvement of Lateral Stability

8.1 Steering System with Devised Rod

For the improvement of lateral stability, the introduction of the nonlinear restoring moments or the solid friction moments with respect to kingpins bar has been considered in the steering system. The model of the steering systems with these additional moments are shown in Fig. 2 by the additional broken lines.

The nonlinear restoring moment, which is produced by the rod of pre-loaded spring, against the axial deformation of the rod is assumed as shown in Fig. 8 (a). The solid friction moment produced by the rod with solid friction is assumed to present the characteristics of the elastoplastic restoring moment shown in Fig. 8 (b). These moments mentioned above should be added to the Eqs. (22) and (23) in the simulation.

8.2 Effect of Restoring and Friction Moments

The effect of the rod of pre-loaded spring and the solid friction rod on the lateral acceleration responses of the vehicles with rigid and elastic guidewheels are shown in Fig. 9 (a) and (b) respectively. It is clear that, though, the solid friction rod causes less effects on reducing the oscillatory motion of the vehicle with rigid guidewheels than the rod of preloaded spring does, both devised rods yield nearly the same improvement effects on the lateral stability of the vehicle with elastic guidewheels without the pre-compression.

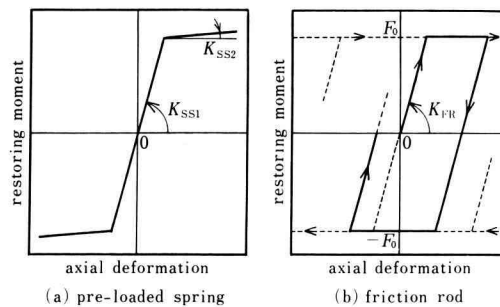


Fig. 8 additional steering moments for stability improvement

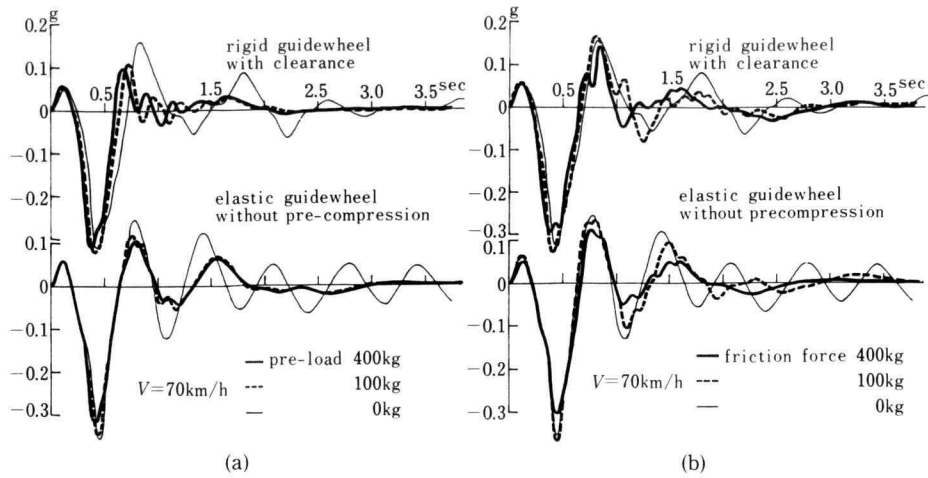
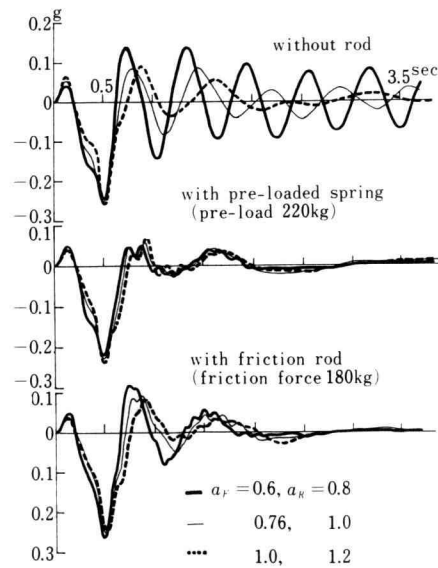


Fig. 9 effect of pre-loaded spring on acceleration response

Fig. 10 effect of steering lever length on lateral acceleration response (elastic guidewheel vehicle without pre-compression, $V=60\text{ km/h}$)

8.3 Effect of Equivalent Steering Lever Length

In general, large steering lever length improves the vehicle stability, however, the length may be limited from the vehicle tracking accuracy. It is found from Fig. 10 that, in particular, the response of the elastic guidewheel vehicle with the rod of pre-loaded spring in the steering system is insensitive to the variation of the steering lever length despite that the responses of the vehicle without any rod and with the friction rods are both considerably

sensitive to it, which means that designing the steering lever length of the steering system with the rod of pre-loaded spring should be fairly free from the restriction of stability points of view.

9. Ride Quality on Guideway with Guidewall Irregularity

The examples of the vehicle lateral acceleration response to the guidewall irregularities on the straight and curved guideways are as shown in Fig. 11. The amplitudes of the front and rear lateral acceleration responses, α_{FO} and α_{RO} , for several cases of the vehicles with the guidance systems considered above are summarized in Figs. 12 (a), (b) and (c). Especially, the rod of pre-loaded spring in the steering system yields the considerable effect on reducing the amplitudes of the lateral acceleration responses of the vehicle with the rigid guidewheel steering systems with the clearance on both straight and curved guideways. Also the friction rod in the steering system has a effect on reducing the amplitudes of the lateral acceleration of the rigid guidewheel vehicle with the clearance on the curved guideway. Furthermore, the amplitude of the lateral acceleration response of the vehicle with rigid guidewheels and friction rods to the guidewall irregularities on straight guideway is sensitive to the change of the friction force of the rod inserted into the guidance system.

It can be said conclusively that both rods of pre-loaded spring and friction rod in the rigid guidewheel guidance systems with the clearance are more effective on improving the lateral ride quality on the straight and curved guideways with the guidewall irregularities than in the elastic guidewheel guidance systems without precompression. It is pointed out from Fig. 12 (c) that though the introduction of the clearance between the guidewheel and guidewall itself alone can not always keep the vehicle insensitive to the guidewall irregularity but causes the deterioration of the lateral ride quality, the intentional introduction of the clearance can reduce considerably the amplitude of the lateral acceleration response of the rigid guidewheel vehicle with the rods of pre-loaded spring to the guidewall irregularities on the straight guideway.

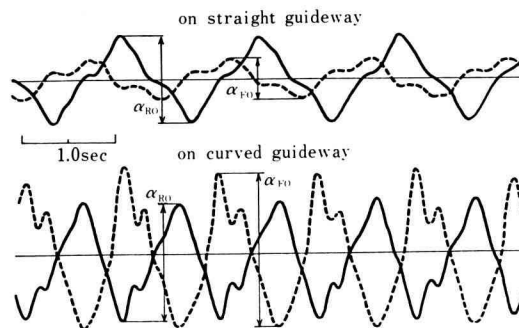


Fig. 11 lateral acceleration response to guideway irregularity (elastic guidewheel with pre-loaded spring)

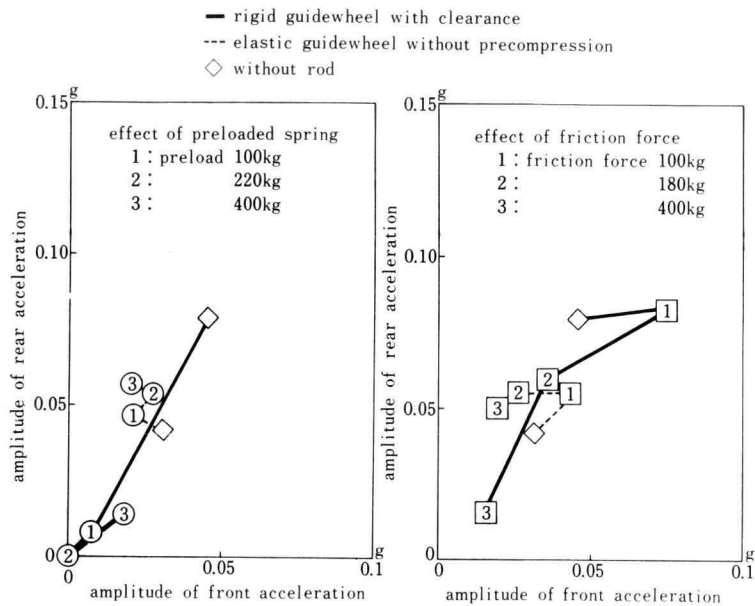
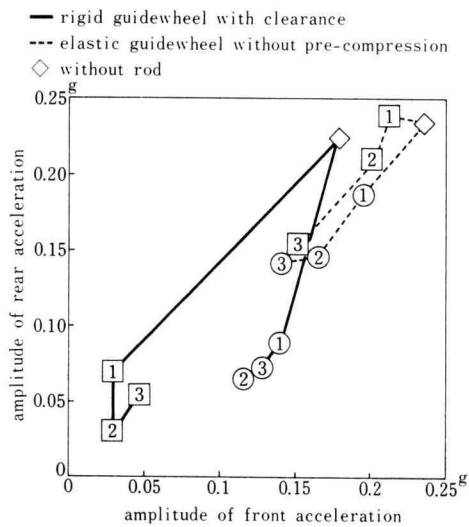
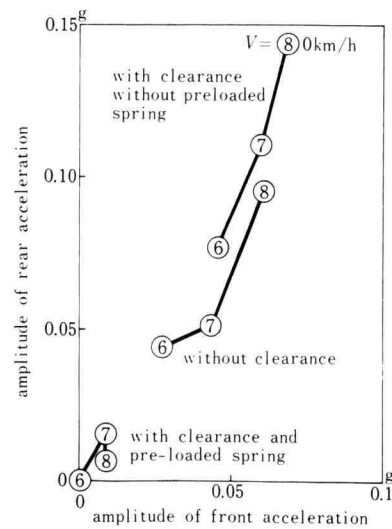
Fig. 12(a) effect of rods on straight guideway ($V = 60$ km/h)Fig. 12(b) effect of rods on lateral acceleration response to guidewall irregularity on curved guideway ($V = 60$ km/h)

Fig. 12(c) effect of preloaded spring and clearance on acceleration response of rigid guidewall vehicle to guidewall irregularity on straight guideway

10. Concluding Remarks

Introducing the nonlinear 7 degree-of-freedom lateral dynamic model of the AGT vehicle, the effect of the steering parameters on the vehicle lateral stability and ride quality has been studied using the computer simulation. The followings are summarized conclusively as a result of the simulation study.

The vehicle lateral stability is deteriorated by the clearance between the guidewall and the rigid guidewheel or by the elimination of the pre-compression of the elastic guidewheel, which are inevitable because of the precision limit in the construction of the guidewall. The rod of pre-loaded spring in the steering system improves the vehicle stability while leaving the vehicle insensitive to the disturbance from the guidewall on both straight and curved guideways. The friction rod in the steering system also has the effect on improving the vehicle stability and ride quality on the curved guideway with irregularities especially when the vehicle equips with rigid guidewheels with the clearance, however, the transient response of the vehicle with the friction rods to such as sudden direction change of the guideway is sensitive to the variations of the steering lever length. Particular attentions should be focused on that the vehicle with the friction rods in steady state runs on the straight guideway with fairly large value of the normal force of the guidewheel which balances to the friction force due to the friction rod in the guidance system. This steady state running which is undesirable from the view point of guidewheel tyre wear has been observed in the straight running after the transient response to sudden direction change of the straight guideway in this simulation.

From the above mentioned views, it has been found out that the devised rod of pre-loaded spring in the guidance system especially with the rigid guidewheels and the clearance results in more favorable effects on the vehicle lateral stability and ride quality than the friction rod does within the vehicle running conditions considered in this simulation. It should be noted that the vehicle lateral motion has been simulated under following ideal assumptions :

- (1) The vehicle forward velocity is assumed to be constant.
- (2) The effect of lateral dimension of the vehicle is assumed to be negligible.
- (3) The simplified guideways with guidewall irregularities is assumed.

However, the considerable effects of the rod of pre-loaded spring has been observed in the real vehicles operated in the Newtram System in Osaka to justify the results obtained in this simulation.

References

- 1) Segel, L., Theoretical prediction and experimental substantiation of the response of the automobile to steering control, Proc. I. Mech. E. (A.D.) 1956-1957
- 2) Shladover, S.E., Wormley, D.N., Richardson, H.H. and Fish, R., Steering controller design for

Automated Guideway Transit vehicles, Transactions of the ASME Journal of Dynamic Systems, Measurement, and Control Vol. 100, March 1978

- 3) Kwak, Y.K. and Smith, C.C., Coupled lateral-vertical dynamics of rubber-tired Automated Guideway Transit vehicle with random guideway inputs, Transactions of the ASME Journal of Dynamic Systems, Measurement, and Control Vol. 102, June 1980