AUTOMATED CONTROL OF GUIDEWAY TRANSIT VEHICLES

by

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ABSTRACT

The control of vehicles for ground transport is an important element of automated people mover systems. An overview of the system control structure is presented, and attention is focused on individual vehicle longitudinal and lateral control.

A longitudinal and lateral control system which utilizes a fixed guideway reference system has been designed, tested, and developed. The fixed guideway reference system is comprised of guideway-mounted measurement markers and an inductive communication link which provides the means for vehicle state error measurement and control. Vehicle status information and longitudinal commands are transmitted to/from the wayside via the inductive link. In addition, measurements of the electromagnetic field surrounding the communication link are used to obtain estimates of vehicle lateral position error with respect to a prescribed path. These vehicle state error measurements are processed by an on-board vehicle computer, and the proper longitudinal and lateral control commands are generated and applied to the appropriate control equipment to effect the necessary longitudinal and lateral control responses.

Preliminary conclusions from this ongoing study are as follows:

1. Using the approach described, specific vehicle dynamics data can be used to provide a good estimate of potential wire following lateral control performance.

2. Lateral paths requiring vehicle yaw acceleration provide the most severe input to a wire-following vehicle.

3. Good longitudinal control accuracy and ride comfort are inversely related. Although analytical predictions of longitudinal control accuracy are useful, final control parameter value selection requires actual hardware testing to select a level of accuracy consistent with ride comfort requirements.

I. INTRODUCTION

The Transportation Systems Division (TSD) of General Motors is currently evaluating automatic guideway transit vehicle controls which have been developed as a part of the Automated Guideway Systems (AGS) program. The AGS program is an internally-funded technical development program directed by the Automated Transportation Department of GM TSD. The objective of the program is the development of automated guideway system technology.

Notable vehicle control hardware development efforts have been performed and documented by Gardels[1], Olsen[2], Fenton[3], Bender[4], and others. However, these efforts have been concerned with automotive size vehicles. The AGS effort, on the other hand, is focused on technology development as it would apply to larger vehicles which are more representative of those needed in an actual people mover system.

This paper focuses on two of the control concepts selected for development; wire-following lateral control and point-follower longitudinal control. The lateral reference signal generation and detection process utilizes an electromagnetic field surrounding a single current-carrying conductor which is embedded in the guideway surface. The longitudinal reference is command data providing velocity and absolute vehicle position to each individual vehicle on the guideway. Bi-directional vehicle/wayside communications is accomplished by an electromagnetic Very Low Frequency (VLF) link over the same single embedded conductor used for lateral control. The measure of vehicle longitudinal position and velocity on the guideway is obtained from dc wheel tachometers. This position estimate is updated by detection of magnetic benchmarks located along the longitudinal axis of the guideway.

The guideway is comprised of an instrumented 3.7 km (2.3 mile) test track; located at the General Motors Proving Ground, Milford, Michigan. Both the vehicle and wayside control equipment incorporate programmable digital computers. Engineering prototype vehicles are currently based on the GMC Transmode chassis. Primary vehicle characteristics are:

Length - 7m (23 ft.)
Front Wheel Drive
7.5 ℓ (455 cu. in.) Engine
Tandem Rear Wheels
Weight 5000 kg (11,000 lbs.) loaded

A wayside mobile test laboratory, situated near the guideway, houses the necessary support and maintenance equipment required for the field testing portion of the program.

II. LONGITUDINAL CONTROL DEVELOPMENT

Automatic velocity and position control of individual vehicles is necessary for many forms of automated transit systems. The use of an internal combustion engine allows certain advantages in guideway construction and system flexibility but does introduce certain complexities into the area of propulsion control.

The basic approach being taken in the development of an automatic longitudinal control system (see Figure 1) is outlined below. This approach methodology was initiated by J.G. Bender [4] and R. E. Fenton of the Ohio State University. The four main steps are:

1. Perform simple throttle step response tests on the uncompensated vehicle to determine a simple (gain plus lag, C_1 & C_2) model of the vehicle dynamics.

2. Use velocity feedback (δ) around the vehicle to desensitize the vehicle system to anticipated day-to-day variations in vehicle characteristics.

3. Use proportional plus integral compensation (K_a & K_b) to effectively cancel the vehicle's lag characteristics (pole) identified from 1 and 2 above. This step, in effect, makes the compensated vehicle look like a simple integration element.

4. Provide additional velocity and position feedback (K_1, K_2 & K_3) around the compensated vehicle such that position and/or velocity control can be achieved.

Unfortunately in actual practice the above four steps do not necessarily ensure good velocity and position control. The typical real world vehicle is not a simple gain and lag but rather a complex assortment of dead zones, hysteresis, and non-linearities, many of which change on a day-to-day basis. In order to accommodate these complexities, compromises in performance must be traded-off against adequate system stability and repeatability.

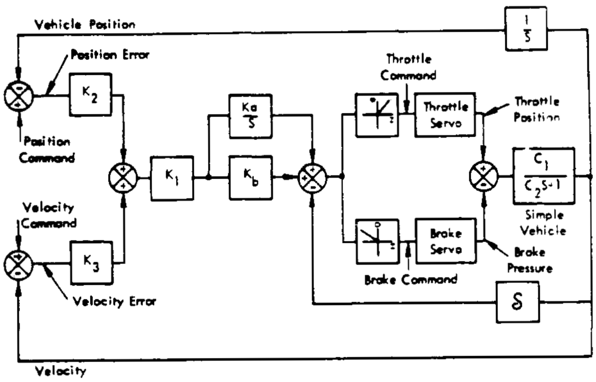
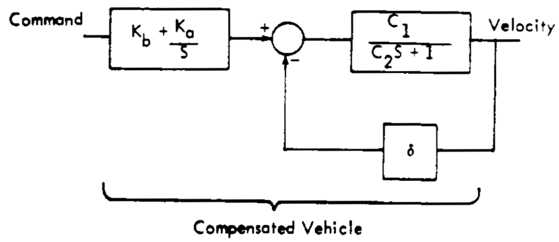


FIGURE 1. VEHICLE LONGITUDINAL CONTROL

Performance Analysis

In order to establish the completion of the first three steps outlined above, a simple step command, proportional to desired vehicle acceleration/deceleration, was applied to the compensated vehicle. It should be noted that the maximum value of the command signal would be constrained by the physical acceleration/deceleration characteristics of the vehicle dynamics.



Theoretically the vehicle should respond as a simple integrator ($\frac{1}{\tau S}$) based on the following pole/zero cancellation.

$$\frac{\text{Velocity}}{\text{Command}} = \frac{K_a}{S}\left(\frac{K_b}{K_a} S + 1\right)\left[\frac{\frac{C_1}{1+\delta C_1}}{\frac{C_2}{1+\delta C_1} S + 1}\right]$$

$$= \frac{1}{\tau S} \quad \text{where } \tau = \frac{1 + SC_1}{K_a C_1}$$

provided K_a and K_b are selected as

$$\frac{K_b}{K_a} = \frac{C_2}{1+\delta C_1}$$

As expected, the properly compensated vehicle did respond as an integration element, $1/\tau S$, to a step command change of acceleration/deceleration. A time history of both the input command signal and resulting vehicle velocity is shown in Figure 2.

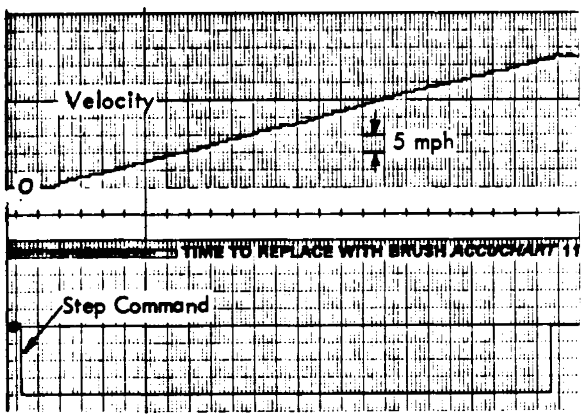
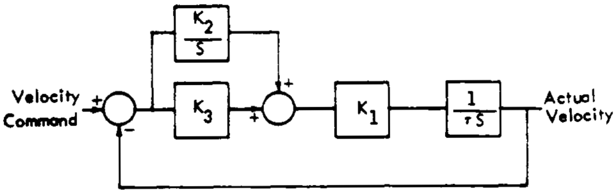


FIGURE 2. COMPENSATED VEHICLE RESPONSE

Continuing on with some assurance that the vehicle is adequately represented by $1/\tau S$, Figure 1 can be reduced to:



The closed loop transfer function resulting from the above diagram is given by

$$G_{cl} = \frac{\frac{K_1K_3}{\tau} S + \frac{K_1K_2}{\tau}}{S^2 + \frac{K_1K_3}{\tau} S + \frac{K_1K_2}{\tau}}$$

For this system the natural resonant frequency (ω_η) is

$$\omega_\eta = \sqrt{\frac{K_1K_2}{\tau}}$$

while the damping ratio ζ is

$$\zeta = \frac{K_3}{2K_2} \sqrt{\frac{K_1K_2}{\tau}}$$

Both ω_η and ζ should be made as large as allowed by stability, speed of response, and passenger comfort requirements.

After ω_η has been chosen, then the theoretical steady state errors to a simple acceleration, cruise, deceleration command profile can be determined. In this case, the steady state position and velocity errors should be zero while the position error during the acceleration and deceleration steps would be determined by the relationship

$$\text{Position Error} = \frac{\text{Acceleration Command}}{\omega_\eta^2}$$

Compensation Selection

As expected, the use of such a simple vehicle model did not provide the necessary insight into actual longitudinal control performance. Considerable time was spent in both measuring vehicle parameters and trying to select proper compensation values. The evaluation program quickly and clearly illustrated that with this control approach a significant tradeoff must be made between control performance accuracy and passenger comfort.

For example, Figures 3 and 4 indicate the vehicle's response to a simple 0.75 m/s^2 acceleration command to 13 m/s (30 mph), a short segment of a 13 m/s constant velocity command followed by a 0.75 m/s^2 deceleration to zero. The value of ω_η for Figure 3 is 0.11 rad/sec and illustrates the control performance corresponding to a "comfortable" ride. On the other hand, the data in Figure 4 results from an ω_η of 0.24 rad/sec and corresponds to a very uncomfortable ride. However, it can be seen that the maximum position error associated with Figure 4 (4 m) is approximately 5 times less than the position error shown in Figure 3 (22 m).

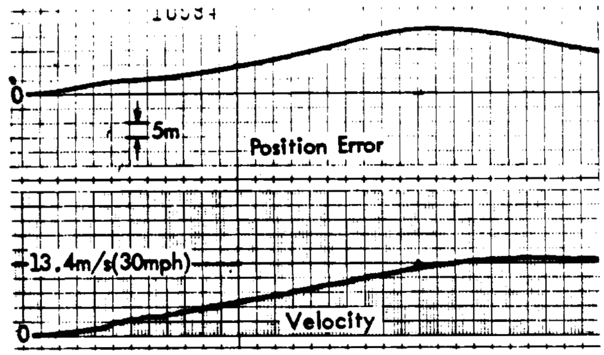


FIGURE 3. VEHICLES RESPONSE WITH ω_η = 0.11 rad/sec

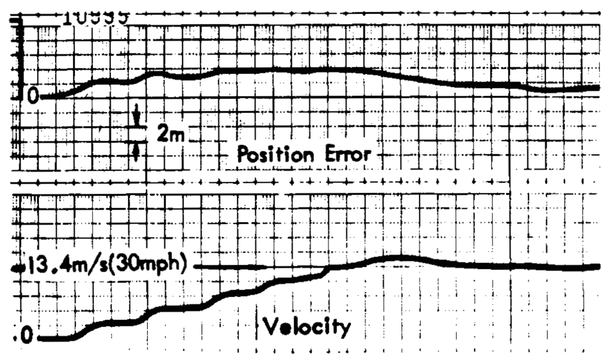


FIGURE 4. VEHICLES RESPONSE WITH ω_η = 0.24 rad/sec

Perhaps the biggest factor limiting the value of ω_η is the dead zone between the application of brakes and throttle. Control of position via thrust control alone is very difficult at low speeds due to the very significant asymmetrical thrust/drag characteristics of an automotive drive train with an automatic transmission. Possible solutions to this problem (if indeed small position errors are required) are:

1. Continually apply a small brake command (25 PSI) to reduce the deadzone such that a more linear thrust/drag relationship can be obtained.

2. Obtain more accurate knowledge of the vehicle's longitudinal dynamic response characteristics, thereby allowing inverse compensation of the vehicle's asymmetrical qualities. This approach is currently being investigated at the Ohio State University[5].

3. Since the vehicle's error characteristics are predictable, the information can be used to determine actual vehicle position based on past commands, or, converesly, base

future longitudinal commands on actual vehicle position and known error characteristics; in essence provide lead data.

III. LATERAL CONTROL DEVELOPMENT

The purpose of the lateral control system is to maintain the vehicle near the center of the guideway as it traverses different guideway geometries at various velocities. The chosen wire-following approach uses an electromagnetic field generated by a vehicle/wayside communication antenna buried in the roadway; the antenna provides a lateral reference signal for the controlled vehicle. A lateral error sensor mounted on the vehicle inductively senses its position relative to the vertical electromagnetic field null and generates an error signal proportional to the vehicle's lateral displacement from the desired path prescribed by the antenna.

A simple block diagram of the lateral control system can be seen in Figure 5.

The processed error signal is acted on by system compensation which is designed to improve both steady-state and transient response modes of operation. The compensator output signal is then used to drive an electromechanical position servo which in turn, applies corrective steering action to the vehicle's power steering system.

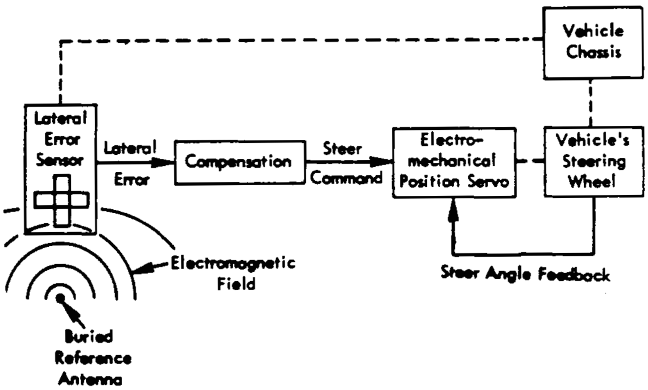


FIGURE 5. LATERAL CONTROL SYSTEM

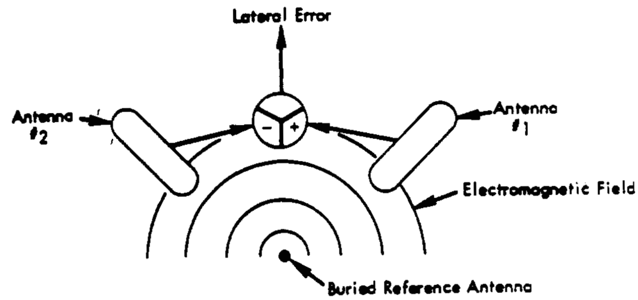
The two major areas of investigation focused on:

1. Developing a simple and effective wire following lateral error sensor

2. Developing a methodology for selecting control compensation and analyzing potential lateral control performance based on known vehicle dynamics data and desired performance and comfort criteria.

Lateral Error Sensor

Previous efforts in the area of wire following typically used the one wire-two antenna amplitude system [1] as shown in the following diagram.



This approach requires the sensing antennas to be relatively high (≈ 45 cm) above the reference antenna in order to provide a linear error output verses lateral deviation signal. However, field amplitude sensing at this height is very sensitive to electromagnetic field distortions and consequently does not always provide an accurate measure of vehicle lateral deviation.

Olsen[2] has documented the benefits of true vertical field null sensing at relatively close distances to the buried reference antenna. For this effort Olsen and others utilized a digitally linearized phased array sensor to provide good lateral control performance and ride comfort. Unfortunately, the phase array sensor requires a large number of components and tends to be quite bulky.

Based on this information, the following list of requirements was used in developing a lateral error sensor.

- Operates by sensing true vertical field null
- Provides output data for errors up to $\pm$ 8 inches (20 cm)
- Insensitive to current variations in guideway reference antenna
- Insensitive to frequency variations in guideway reference antenna
- Accuracy and linearity of system shall be within 5% when the sensor is within its proper operating region.

Figure 6 illustrates the developed approach that is both simple and relatively insensitive to field distortions. Field distortion insensitivity is achieved by the sensor's ability to operate within close (≈ 15 cm) proximity of the buried reference wire.

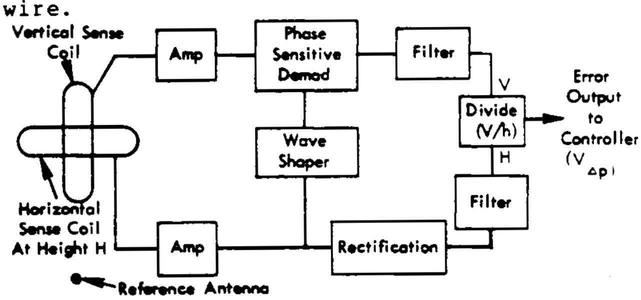
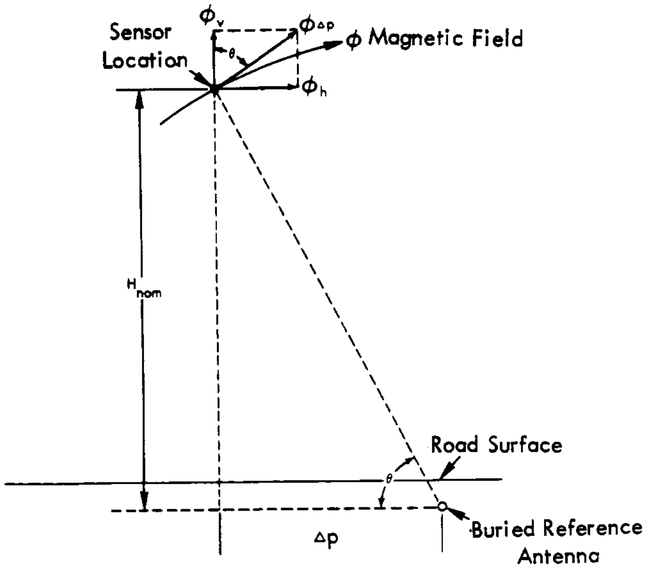


FIGURE 6. LATERAL SENSOR MECHANIZATION

The following diagram and equations illustrated the theoretical operation of the sensor.



With a magnetic field of ϕ, the vertical coil will sense a voltage proportional to $d\phi_v/dt$ while the horizontal coil will sense a voltage proportional to $d\phi_h/dt$. Dividing the vertical coil output by the horizontal coil output produces a voltage proportional to

$$\text{Output} = \int \frac{\frac{d\phi_v}{dt}}{\frac{d\phi_h}{dt}} = \int \text{Cot}\ \theta = \int \frac{\Delta P}{H_{nom}}$$

In other words, the output is a linear function of the lateral error (Δp) if the height of the antenna is held constant. This output is also insensitive to variations in reference antenna frequency and current. Several antennas of this configuration have been constructed and are now used by all test vehicles for lateral error sensing. A comparison between the output of this type of sensor and the phase array sensor can be seen in Figure 7.

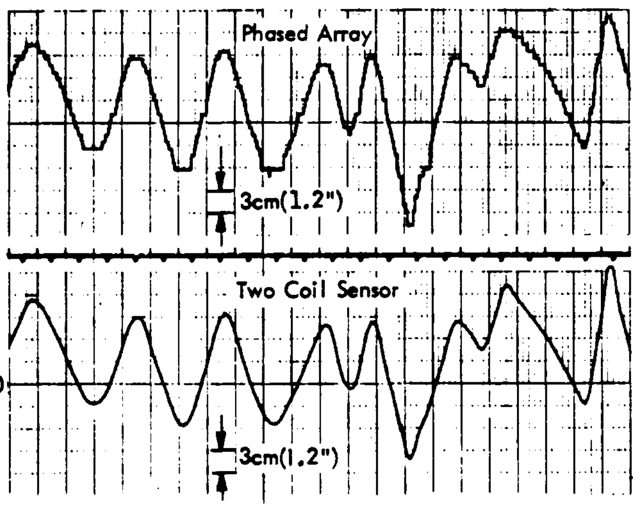


FIGURE 7. LATERAL SENSOR COMPARISON

Lateral Compensation Selection and Performance Estimation

A survey of the literature in the field indicated that simple lead compensation would provide adequate lateral control performance. Initial testing of the system with simple lead compensation proved quite disappointing. Stable lateral control was obtained only at speeds below 16 km/h (10 mph), and even at this speed significant lateral errors were encountered. Since all components of the lateral loop with the exception of the vehicle itself were known in detail, emphasis was placed on obtaining a better understanding of pertinent vehicle lateral dynamics.

Two specific types of tests were performed during the vehicle dynamics investigation. The first test involved monitoring a stabilized (no roll coupling) lateral accelerometer mounted near the front of the vehicle as the steer servo was excited with a small amplitude sine wave signal. A servo analyzer compared the actual lateral acceleration with the applied steer angle to determine the frequency response characteristics of the vehicle's dynamic behavior. An example of the vehicle's response at 32 km/h (20 mph) is shown in Figure 8. Test runs were performed at 16, 32, 48 and 64 km/h. Assorted structural resonances were quite noticeable at the higher speeds and higher frequencies.

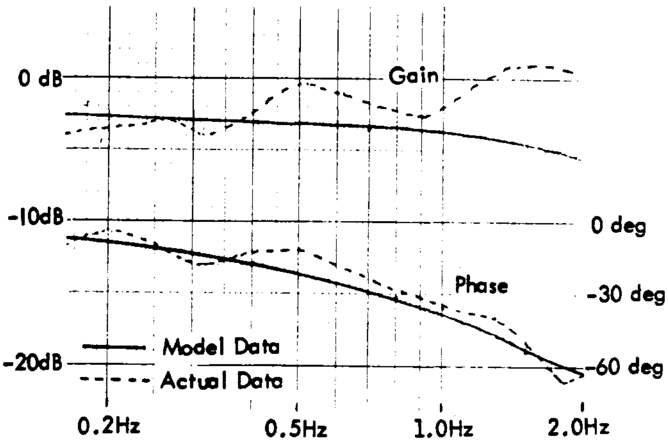


FIGURE 8. ACTUAL & MODEL VEHICLE RESPONSE

The second test type is referred to as the "Standard Lateral Control Response" tests. It was the purpose of these tests to determine specific vehicle parameter values necessary for accurate vehicle characterization and mathematical modelling. Principle parameters of interest were yaw velocity response times and front and rear cornering compliance.

Using general purpose vehicle dynamics equations[6], a simple second order model appeared to fit the test data quite well at the lower speeds that were of interest. The second order model shown below was then studied in an effort to determine what could be done to the vehicle in order to improve its response characteristics.

$$\frac{a_y}{\delta}(s) = \frac{u\left[\frac{D_r u}{57.3g}s + 1\right]}{\left[\dfrac{\dfrac{D_f D_r u^2}{(57.3g)^2}}{1 + \dfrac{u^2(D_f - D_r)}{(a+b)57.3g}}\right]s^2 + \left[\dfrac{\dfrac{u(D_f + D_r)}{57.3g}}{1 + \dfrac{u^2(D_f - D_r)}{(a+b)57.3g}}\right]s + 1}\left(\frac{r}{\delta}\right)_{ss}$$

where $$\left(\frac{r}{\delta}\right)_{ss} = \frac{\dfrac{u}{a+b}}{1 + \dfrac{u^2(D_f - D_r)}{(57.3g)(a+b)}}$$

a_y = lateral acceleration, ft/sec^2

r = yaw velocity, radians/sec

δ = front wheel steer angle

u = forward velocity, ft/sec

a,b = distances from center of gravity to front and rear wheels, ft

D_f, D_r = front and rear cornering compliances, deg/g

g = acceleration due to gravity, 32.2 ft/sec^2 (9.8 m/sec^2)

57.3 = constant relating degrees and radians

As can be seen from the preceding equations, a faster responding vehicle is produced if the front cornering compliance is much larger than the rear cornering compliance, and the product of the two cornering compliances should be as large as possible. Based on this insight, appropriate tires were selected and properly inflated to improve vehicle response times. The resultant improvement as measured on the vehicle was significant. For example, this modification reduced the vehicle's lateral acceleration phase lag at 1 Hz by 45 degrees. The simple lead compensator was now employed with immediate success.

Many combinations of poles and zeros were tried on both straight and curved roadway segments. In addition, as suggested by the mathematical model, a velocity-dependent gain was employed to improve over-all performance. At this point, the over-all lateral control loop can be modeled by the block diagram in Figure 9. A pure integrator was added to the simple lead compensation in order to reduce steady state errors to zero when the vehicle is in a constant radius curve. Preliminary testing indicates that the control system is not very sensitive to changes in the compensation. This implies that any increase in performance levels must come from other methods for increasing the gain/bandwidth of the over-all loop.

As seen in Figure 10, the lateral control subsystem performs quite well on straightaways and constant radius curves. However, moderate lateral errors are generated when a test track spiral requires yaw acceleration of the vehicle. An example of the lateral error accompanying a vehicle undergoing a yaw acceleration of 1.2 deg/sec^2 can be seen in Figure 11. At low or moderate velocities, this type of error is not critical; however, in order to better understand this problem, the following analysis was undertaken.

A vehicle requires a constant yaw rate command (constant steer angle) to successfully travel a constant radius curve - at a constant speed. The required yaw rate for a curve is given by $r = \frac{u}{R} 57.3$ where:

r = yaw rate (deg/sec)

u = vehicle velocity (ft/sec)

R = curve radius (ft)

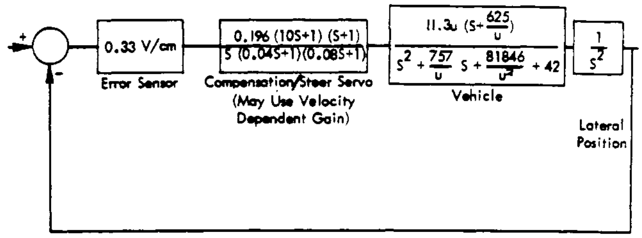


FIGURE 9. LATERAL CONTROL LOOP

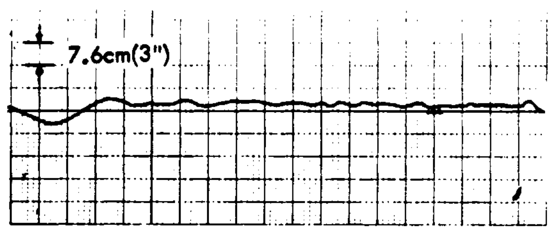


FIGURE 10. LATERAL ERROR ON STRAIGHTAWAYS AND CONSTANT RADIUS CURVES

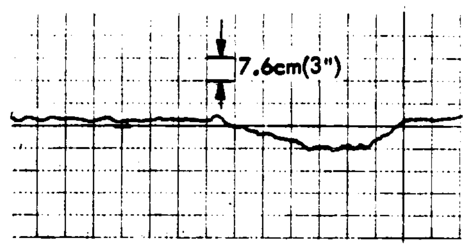


FIGURE 11. LATERAL ERROR DURING ANGULAR YAW ACCELERATION

While the average yaw acceleration encountered in a spiral is given by:

$$\dot{r} = \frac{u}{d}\left(r_f - r_i\right)$$

where

$\dot{r}$ = yaw angular acceleration (deg/sec^2)

u = vehicle velocity (ft/sec)

d = length of spiral (ft)

r_f = final yaw rate (deg/sec)

r_i = initial yaw rate (deg/sec)

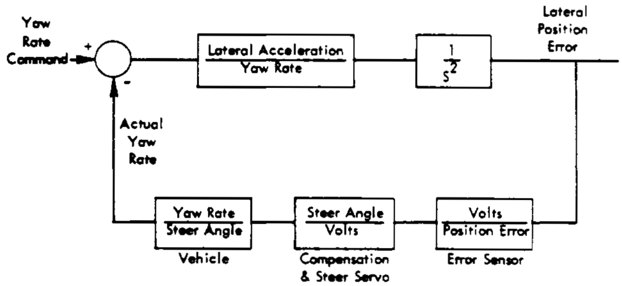
In our case typical test track values are:

r = 206 m (678 ft)
u = 72 km/h (45 mph)
d = 91 m (300 ft)

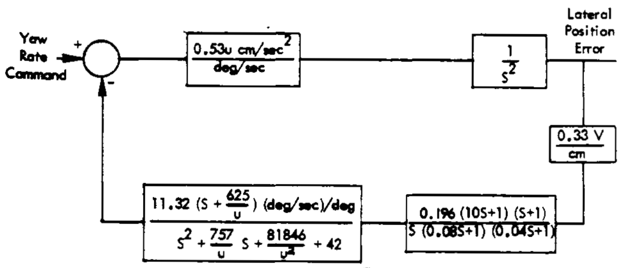
r = 5.58 deg/sec
$\dot{r}$ = 1.23 deg/sec^2

Also, there is no appreciable superelevation of the the roadway.

A reorganization of the block diagram in Figure 9 produces



Applying simplified real world values to the above blocks produces



The forward path in the above mechanization is type 2; therefore, the yaw rate error will be zero for a ramp (spiral) input.

The steady state lateral position error necessary for this condition (i.e., actual yaw rate = cmd. yaw rate) can be found from classical steady state error techniques.

The predicted steady state lateral error for a yaw acceleration of 1.23 deg/sec^2 is 10.7 cm. This value has been essentially verified during the on guideway testing effort.

It should be noted that if the proper amount of super elevation is applied, yaw acceleration (spirals) ceases to be a problem. However, if super elevation is not available and additional lateral control accuracy is required, it appears that additional state data must be sensed and used before any appreciable performance improvements can be expected.

Approaches currently being considered are:

1. Mount a second antenna on the rear of the vehicle such that both yaw and lateral position information can be obtained.

2. In addition to the front and rear antenna use a yaw rate gyro to provide both absolute and relative yaw information.

3. Configure a system which has apriori knowledge of the track layout and can apply the proper steering bias based on known longitudinal position.

Mathematical models are being used to investigate the above approaches. Initial testing of a meachanization utilizing both a front and a rear antenna has yielded excellent results as shown in Figure 12. More work is continuing in this area.

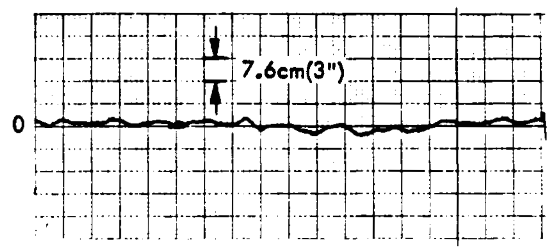


FIGURE 12. FRONT LATERAL ERROR EXPERIENCED WITH FRONT AND REAR LATERAL ERROR SENSORS

(u = 66 ft/sec $\dot{r}$ = 1.23 deg/sec^2

IV. SUMMARY AND CONCLUSIONS

The findings to date from this ongoing study are as follows:

- Good longitudinal control accuracy and ride comfort are inversely related. Although analytical predictions of longitudinal control accuracy are useful, final control parameter value selection requires actual hardware testing to select a level of accuracy consistent with ride comfort requirements.
- Good wire-following lateral control performance can be obtained on larger sized rubber-tire vehicles, provided that the specific vehicle's lateral dynamics are taken into account during the design phase.
- A simple vehicle model can provide useful estimates of the steady state lateral errors associated with a vehicle undergoing low to moderate yaw accelerations.
- Further reduction in lateral control errors can be obtained by using vehicle rear lateral error information.

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