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An Active Roll Mode Suspension System for Ground Vehicles

This paper describes an automatic roll mode control system for ground vehicles. The primary objective of the so-called "active suspension system" is to maintain a coordinated vehicle banking attitude during cornering and steering through traffic. Efforts were also made to render the vehicle insensitive to the undulation of the road surface, wind gusts, and other disturbance inputs. Emphasis was placed upon the development of design logic in the application of control system concepts to a physical system. Realization of the active suspension concept was achieved by parameter optimization of a simplified system on the analog computer and the design and construction of an experimental vehicle. Laboratory and road tests of the physical system confirmed the feasibility of the active suspension concept and brought to focus additional design considerations such as vehicle elastic mode and the effects of man vehicle coupling. For the road tests, a manual bias was incorporated in the automatic roll control loop to improve the transient response of the system, and the resultant man-machine multiloop interaction was investigated.

Motivation

The inefficient use of land for traffic flow appears to be a major contributing factor to traffic congestion in many modern cities. While skyscrapers are built to increase the utility of land for social and business purposes, the premium value of the land prevents further widening of the city streets to speed the flow of people and goods between buildings. The slowly crawling bus and the transfer problems of rapid transit systems force increasing numbers of commuters to rely upon private automobiles for daily transportation, which further adds to the congestion of city traffic.

Without further elaboration on the dynamics of city growth and the optimum evolution of its associated transportation systems, the following two approaches appear to be logical in general:

1 Inducing more commuters to leave private vehicles at home and to rely upon public transportation.
2 Reducing the size of private vehicles so that more units could fit without congestion on present city streets.

Implementation of the first scheme may be aided by the development of a high speed monorail system having "no change" surburban distribution and route flexibility, by virtue of a unique capability of each passenger module to operate over present roadways as well as on the elevated monorail. Monorail express lines represent a good compromise between aesthetic and economic demands, and its use together with the dual-mode vehicles may adapt to the rapidly changing society better than other forms of transportation systems.

Implementation of the second proposal may require a new type of narrow vehicle which has the capacity of carrying two passengers in tandem, rides with the freedom of a motorcycle, and yet is equipped with a body to protect the passengers from weather and accidents. It should be able also to stand erect automatically.

Fig. 1 shows an artist's view of the proposed vehicle types—three rapid transit modules coming down off the elevated monorail to continue the bus service on the suburban avenue and two of the narrow private vehicles following behind a conventional automobile to illustrate their relative size. In both cases, the stability of the vehicles is maintained with a roll mode active suspension system. A report on the development of this type of system is the primary objective of this paper.

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The Active Roll Mode Suspension System

Mission Analysis. The active roll mode suspension system is a mechanism for shifting the center of gravity inboard during cornering so that the resultant force vector, composed of both gravitational and centrifugal components, is oriented along the vertical body axis of the vehicle. Aircraft can be made to execute such coordinated turns through the proper use of rudder and ailerons, but conventional automobiles and trains must rely upon banking of the roadway to eliminate the sidewise tendency. Coordinated operation could improve passenger comfort in all types of vehicles, but it becomes imperative in vehicles of narrow track in order to allow high speed turns. While banking of the roadway allows coordinated operation of conventional road vehicles for a single specified forward speed, the active suspension system accommodates a broad range of travel speeds in coordinated operation.

To satisfy the intended objectives of the vehicles with multi-degree of freedom, as discussed earlier, the roll mode active suspension system not only has to provide steady-state stability, but must also perform satisfactorily under the following operating conditions:

(*a*) Riding smoothly over uneven road surfaces.
(*b*) Exposure to wind or other disturbing lateral forces while running or standing.
(*c*) Subjection to quick steering control as required for avoiding obstacles in the traffic.

For the first operating condition, the active roll mode suspension system must allow the wheel to ride over bumps on the road with little motion or force transmitted to the vehicle body. In a passive suspension system this is accomplished with a soft spring and a soft damper.

To prevent excessive sagging of the vehicle under load which usually is associated with a soft suspension system, a self-leveling scheme is often used. This system may be considered as a vertical mode active suspension system which is different from the active roll mode suspension discussed in this paper. The former is also much less sophisticated in dynamic considerations.

In general, the roll mode motion of a vehicle body with respect

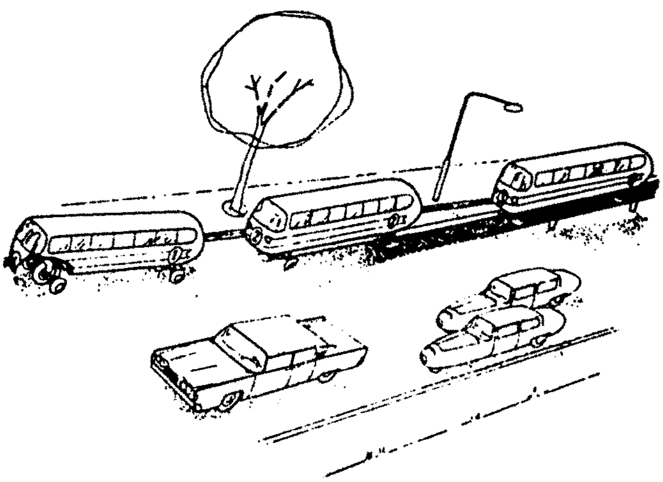


Fig. 1 Proposed vehicles with a conventional automobile

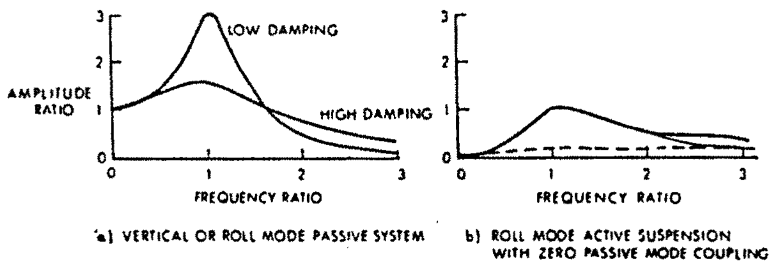


Fig. 2 Frequency response of various-type suspension systems

to the wheel suspension system could be either coupled or uncoupled with the vertical mode motion. For a vehicle designed to perform automatic coordinated turns, it is desirable to decouple the roll mode of motion from the vertical mode of motion so that separate suspension systems with different dynamic characteristics may be used for each mode. Thus the soft vertical mode suspension system may be either passive or active, while the active roll mode suspension system must provide sufficient stiffness with respect to the apparent vertical and offer little resistance against roll mode ground disturbance applied to the wheels. In the ideal case such as a motorcycle, this objective is easily accomplished since the in-line wheel arrangement simply cannot transmit much rolling torque. On the other hand, a narrow vehicle with the side by side wheel arrangement as required for providing the roll mode rigidity against disturbing forces while standing, would be quite susceptible to the roll mode undulation of the road surface while in motion.

The dynamic characteristics of a coupled system as compared with that of an uncoupled roll mode suspension system are best illustrated by the frequency response of the two systems as shown in Fig. 2. Fig. 2(*a*) shows the amplitude ratio of a coupled roll mode suspension system which behaves in the same general way as that of a vertical mode suspension system; whereas, in Fig. 2(*b*), the steady-state response of an uncoupled active roll mode suspension system is shown to be zero as achieved by the feedback system through torquing the vehicle with reference to the apparent vertical. In doing so, it also reduces the amplitude ratio over the entire low frequency range. Thus, by mechanical decoupling of the roll mode suspension system from the vertical mode, a full development of the intended system performance is realized. The light line in Fig. 2(*b*) shows the effect of an overpressure relieving valve used in the hydraulic ram of the roll mode suspension system to allow the use of higher effective damping and thus help to reduce the frequency response at known

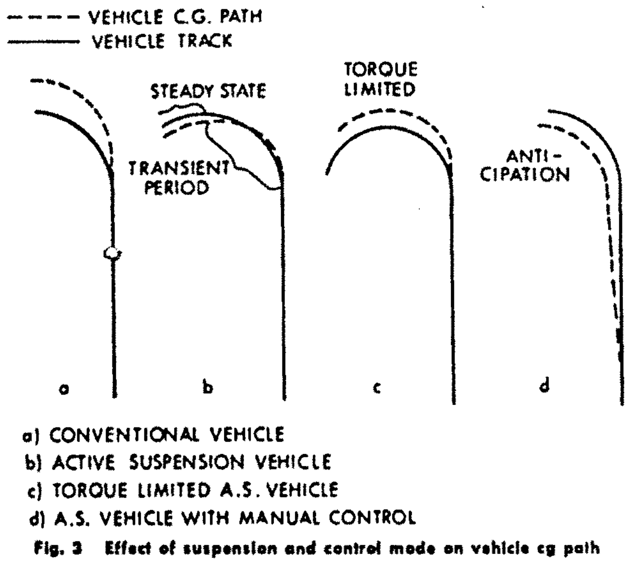


Fig. 3 Effect of suspension and control mode on vehicle cg path

frequency range. The dotted line in Fig. 2(*b*) represents the ultimate performance of a suspension system utilizing a road surface sensor in conjunction with the apparent vertical sensor in the active suspension system. The experimental work on this type of suspension system is under study now as the continuation of this present report.

The most demanding mission for the active roll mode suspension system to accomplish is to maintain a coordinated turn in transient condition while the vehicle is quickly steered from one turning rate to another. The difficulty is compounded by the inherent dynamic delay of a feedback system and the roll mode

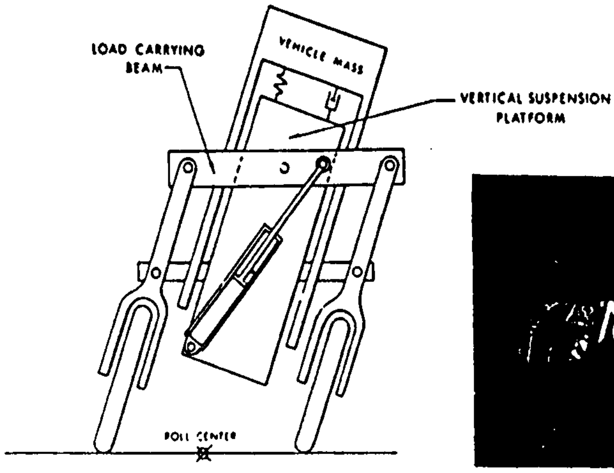


Fig. 4(a) Roll actuator and suspension system

Fig. 4(b) Experimental vehicle (Yamaha motorcycle)

gravitational torque limitation of the narrow vehicle due to its narrow wheel base. This limitation sets the upper bound of the overall system capability in terms of the permissible rate of change of turning rate. To illustrate this concept the following figures are used to show the response of a vehicle subjected to a step-function change of turn rate under different operating conditions. Fig. 3(a) illustrates the path of the cg of a conventional vehicle with passive suspension system (dotted line) when the track of the vehicle is entering into a curve from a straight-line path (solid line). Fig. 3(b) illustrates the same operating condition for a vehicle equipped with an active roll mode suspension system. In contrast with Fig. 3(a), the steady-state portion of the dotted curve in Fig. 3(b) is located inside the solid curve indicating the steady-state coordinated turn of the vehicle. However, during the initial transient period, there is a delay for the vehicle roll angle to be coordinated with the turn rate of the vehicle. This transient becomes longer as the curve becomes tighter until at Fig. 3(c) a critical condition is reached when the system is torque limited and the desired steady-state condition can never be achieved. To overcome this difficulty, the first approach one might use is to relax the steering control so that the turning rate is introduced gradually to allow the system to stay within the limit of the effective roll torque needed to swing the cg of the vehicle gradually toward the inside of the curve of the track. The determination of this maneuver can be a good exercise of optimum control logic [5][1] such as to find the path which would yield the minimum time to reach a steady-state coordinated turn or the path for minimizing the integrated square of the deviation from the specified reference path or, to go one step further, to compute the path for the optimizing of a cost function involving a function of the time, the deviation, and possibly the riding comfort factor. Computationwise, this has little direct application value, but it serves as a most meaningful reference for studying the result of some near optimum control performed by a human operator after some training.

A second and more effective scheme [6] is to allow the driver, in anticipating the turn, to torque the vehicle in advance of the steering through the use of a manual control. The result of this maneuver is depicted in Fig. 3(d). This is not only a faster and safer maneuver, it is also more compatible with the human sense of security and therefore provides better riding comfort. Here during the initial transient section, even though the system is apparently operating in an uncoordinated manner following the definition applied to steady-state operation, the lateral acceleration is in fact "coordinated" with other state variables for the optimum path connecting the two steady-state operations. Humans, through years of experience in learning to run or to ride bicycles, are not only capable of executing the near-optimum maneuver during the transient state, but they would also feel most comfortable being consciously aware of their maneuver in coordination with the change of the steady-state condition.

A third scheme for performing the foregoing maneuver may also be conceived with an automatic scheme in which steering command is used to control the tilt mechanism directly while the direction control of the front wheel is slaved to the tilt angle through some delaying action. This scheme, however, lacks reliability since, in case of emergency, it is always desirable to have direct manual control of the steering.

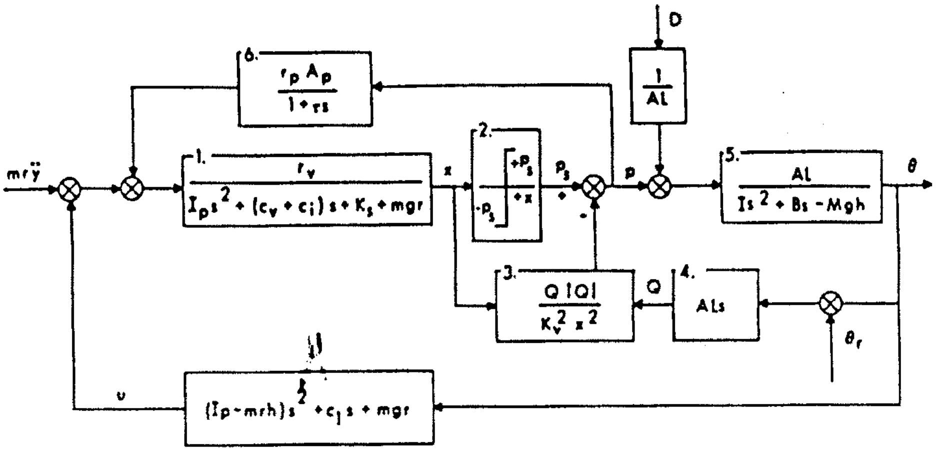
Design Concept of the Control System. To demonstrate the feasibility of a roll mode active suspension system, an experimental three-wheeled vehicle was designed and tested. Fig. 4(a) shows the schematic diagram of the hydraulic ram drive and the rear wheel arrangement of this vehicle, while Fig. 4(b) shows the actual experimental vehicle. In this system the two rear wheels are guided to move in parallel with the vertical axis of the vehicle. They are further constrained by a horizontal bar hinged at the middle to a platform which, in turn, carries the load of the vehicle. The complete system is thus free to roll about an axis as marked on the diagram, at the center between the track of the two wheels. The roll motion of the vehicle is then controlled by the action of the hydraulic ram attached between the horizontal bar and the platform. The vertical mode suspension system is carried between the platform and the vehicle body. Thus the vertical mode of motion is decoupled from the roll mode and, furthermore, through this arrangement, the compliance of the vertical mode suspension system cannot impair the dynamic behavior of the roll mode control.

Even with the vertical mode suspension system thus decoupled, the roll mode control system is still coupled with the elastic property of the tires, the elastic mode of the structure [7], and the resilience of the rider to complicate the dynamic behavior of the design. However, for the initial phase of the design, a rigid body vehicle was assumed, with the elastic mode problems left to be handled during the experimental stage.

The block diagram in Fig. 5 shows the layout of the essential component characteristics of the roll mode control system. The main forward control loop involves an apparent vertical sensor in the form of a tuned pendulum (block 1), the hydraulic control valve (blocks 2, 3, and 4), and the combined effect of the hydraulic ram and the vehicle (block 5).

The dynamic characteristics of the vehicle are essentially the same as those of an inverted pendulum with some damping [8] derived from the friction of the wheels on the ground and the pumping effect of the hydraulic ram. For an ideal system with an inverted pendulum as the load, a unit feedback and a simple gain K in the forward loop, the closed-loop-system stability can be achieved if the product of "KAL" (see Fig. 5 for notation)

[1] Numbers in brackets designate References at end of paper.



A Hydraulic ram area	I_p Pendulum moment of inertia	r_p Pressure feedback moment arm
A_p Pressure feedback area	K_v Valve flow sensitivity	r_v Valve actuating arm
B Vehicle damping	K_s Effective pendulum stiffness	s Laplace transform operator
c_i Pendulum inertial space damping coef.	L Ram moment arm	u Torque applied to pendulum sensor
c_v Pendulum vehicle damping coef.	M Vehicle mass	x Valve opening
D Disturbance torque	m Pendulum mass	$\ddot{y}$ Lateral acceleration of track
g Gravitational constant	p Ram pressure	θ Vehicle-inertial space roll angle
H Vehicle c.g. height	p_s Supply pressure	θ_r Road-inertial space roll disturbance
h Pendulum pivot point height	Q Valve flow rate	τ Pressure feedback time constant
I Vehicle moment of inertia	r Pendulum length	

Fig. 5 Characteristics of an active suspension vehicle

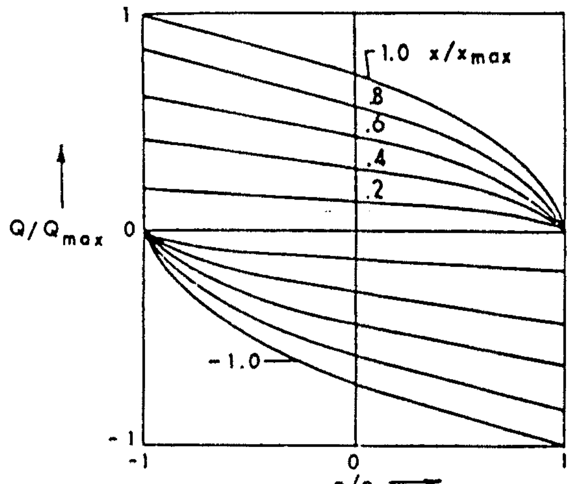


Fig. 6 Static characteristics of valve

is larger than the product of "Mgh." The nonlinearity and additional lag introduced by the valve and sensor present further complications.

For the present experiment, a closed-center spool-type hydraulic valve with a static characteristic as shown in Fig. 6 was used [9]. The performance of the valve as characterized by the curves in Fig. 6 is represented in Fig. 5 by the mathematical model identified in blocks 2 and 3.

In the system in Fig. 5, a pressure feedback scheme (block 6) is shown which is used for the following reasons:

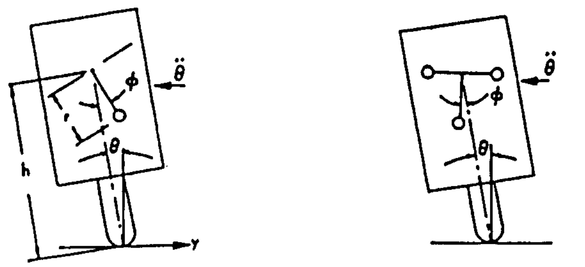
(*a*) As mentioned before, the closed-loop system must have sufficient gain such as KAL > Mgh to keep the system erect, and yet as a general rule for feedback systems the gain must be less than a critical value to prevent oscillation. The nonlinearity of the control valve as shown in Fig. 6 behaves like an amplitude dependent gain and thus makes it difficult to place the gain for different amplitudes within the range bounded by the stability requirement. The use of linear pressure feedback together with high forward loop gain could overcome this difficulty.

(*b*) With the use of pressure feedback it is possible to introduce high gain in block 1 without affecting the total forward loop gain of the overall system. This high gain at block 1 allows the system to minimize the road disturbing effect which is introduced after block 1.

(*c*) With the use of a lag in the pressure feedback loop, it is possible to get an effective lead for the overall system.

Ideally, the apparent vertical sensor identified in block 1 in Fig. 5 should be located at the roll center of the vehicle so that it indicates only the angular deviation of the vehicle from the apparent vertical. Unfortunately, this roll mode rotation center of the vehicle is coincident with the road surface and for practical purposes the pendulum must be installed somewhere else on the vehicle at a distance, *h*, above the ground. At this location the pendulum is sensitive to the sum of the lateral acceleration of the vehicle guided by its track and the lateral acceleration caused by the roll mode angular acceleration of the vehicle multiplied by the factor *h*. In the closed loop, this lateral acceleration produces some undesirable effect as elaborated next.

In the arrangement of Fig. 5, the normal function of the pendulum is to generate a drive pressure to yield a torque of pAL acting against the gravitational pull of Mghθ upon the vehicle in proportion to the gravitational torque of mgrθ upon the pendulum. The lateral acceleration applied to the pendulum due to the roll mode acceleration $\ddot{\theta}$ of the vehicle produced by the drive moment pAL, however, has a tendency to further increase the pendulum torque by a factor of *mrh*, thus rendering the closed-loop system unstable. To overcome this difficulty, the pendulum is "tuned" with the addition of a sufficiently large moment of inertia I_p to compensate for this undesirable effect. An analytical treatment of the dynamic behavior of the compound pendulum subjected to vehicle roll mode motion and lateral accel-



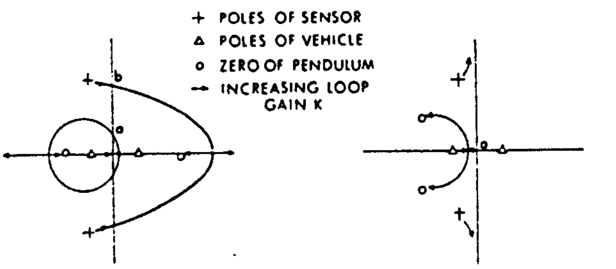
$$\frac{\phi(s)}{\theta(s)} = \frac{(I_p - mrh)s^2 + mgr}{I_p s^2 + c_v s + K_s + mgr} \qquad \text{VEHICLE DAMPING ONLY}$$

$$\frac{\phi(s)}{\theta(s)} = \frac{(I_p - mrh)s^2 + c_i s + mgr}{I_p s^2 + (c_v + c_i)s + K_s + mgr} \qquad \text{INERTIAL AND VEHICLE DAMPING}$$

$$\frac{\phi(s)}{a_y(s)} = \frac{mr}{I_p s^2 + (c_v + c_i)s + K_s + mgr}$$

$\frac{mrh}{I}$ = 1 CRITICALLY TUNED
< 1 SUPER TUNED
> 1 UNDER TUNED

Fig. 7 Transfer functions of the apparent vertical sensors



a) UNDER-TUNED SENSOR WITH 0° PHASE ANGLE ROOT LOCUS PLOT
STABILITY: K > K AT POINT a, < K AT POINT b

b) SUPER-TUNED SENSOR WITH 180° PHASE ANGLE ROOT LOCUS PLOT.
STABILITY: K > K AT POINT a

Fig. 8 Root loci plots for a pendulous apparent vertical sensor

eration is shown in Fig. 7. In this analysis a critically tuned pendulum is defined as the condition when $I_p = mrh$. For example, if the length h is ten times longer than the radius r, and assuming that the additional moment of inertia has a radius of gyration equal to r, then the additional mass required would have to be nine times the pendulum mass for critical tuning. In general, for satisfactory vehicle stability, the pendulum should be supertuned by some 20 percent. Such a condition is illustrated by the root locus analysis shown in Fig. 8(b), where stability can be maintained easily with a gain larger than a critical value as indicated. On the other hand, with an undertuned pendulum, as shown in Fig. 8(a), a very narrow gain margin is available for stable operation. In fact, when other delays and nonlinearities were included in the analog study, no stable region for the system was found with the undertuned sensor.

In actual application, this tuned pendulum is coupled together with the input drive of the hydraulic valve, the pressure feedback linkage, and some suitable damper. The stiffness of all these components is summed up together to form the equivalent stiffness. Similarly, all of the damping factors as well as the masses of the components are summed up together to form the effective damping and mass.

Damping torques for the pendulum sensor can be introduced in two different modes. In the first and most obvious mode, the pendulum is damped with respect to the framework of the vehicle by a suitable damper used in parallel with the unavoidable damping factors which are associated with the valve, the pressure feedback, etc. The second mode of damping can be introduced with respect to inertial space, such as the use of a rate gyro or simply a flywheel.

With analog simulation study for a rigid body system while neglecting the elastic mode it was found that an inertial referenced damper can only provide minor improvement to the stability of the system while the vehicle referenced damper proved to be somewhat detrimental. However, with the experimental vehicle system a vehicle referenced damper was found to be essential for damping out the elastic mode oscillation.

Experimental Realization. In principle, after the general scheme of a system is formulated, the ensuing step would be the optimization of the system performance with the adjustment of the parameters. However, for a complicated system w'th numerous performance specifications to be satisfied by the maneuvering of a large number of adjustments, it would be more sensible to divide the entire problem into major groups where each group is identified by some characterizing parameters so that the complete system can be optimized first by adjusting the characterizing parameters of the major groups. In a subsequent operation, the actual component parameters would then be optimized to satisfy the requirements of each major group.

For the system described in this paper, the major groups involved can be established as the following:

(a) The rigid body vehicle and hydraulic drive system.
(b) The sensor system including the feedback.
(c) The elastic mode of the vehicle.
(d) The human body coupling.

The approach taken in the present stage of development involves the use of an analog simulation of the primary system consisting of groups (a) and (b) listed previously. The optimized characteristic parameters of group (b), including the damping ratio, the natural frequency, the tuning factor, the sensitivity of the sensor system, and the sensitivity of the feedback system were established through the searching process on the analog computer and guided by root locus study. They were then used subsequently to design the parameters of the actual hardware such as the pendulum mass, the lever arms, the spring constants, the characteristics of the pressure sensors, etc., of the controller. The coupling problems of the primary vehicle system with human rider and the elastic mode were later studied heuristically with the experimental model.

Analog Computer Study. For the simulation of the vehicle with active roll mode suspension system, an analog computer layout, as shown in Fig. 9, was used. This layout treats the flow rate of the hydraulic valve as a prime output while the block diagram in Fig. 5 implies control of the vehicle with regulated pressure. Both the block diagram and the computer simulation represent the theoretical model of the vehicle. The block diagram formulation allows better engineering insight into the system while the analog study is adapted to the features of the computer hardware.

With the use of a calculated guess, a set of initial settings were chosen to put the system into reasonably stable operating condition and served as a reference. From there, with the use of transient response criterion, it was possible to scan through the variation of the parameters in different combination to search for a near optimum response. A typical step-function response at the near optimum setting is shown in the middle diagram in Fig. 10. This is surrounded by four other response curves; each has one parameter displaced off the optimum.

The result of Fig. 10 not only supports the basic concept of the original configuration layout in Fig. 5, it also provides a solution for the characteristic parameters of the sensor and feedback group. The design of the sensor and feedback controller also involves some interesting compromises and optimization pro-

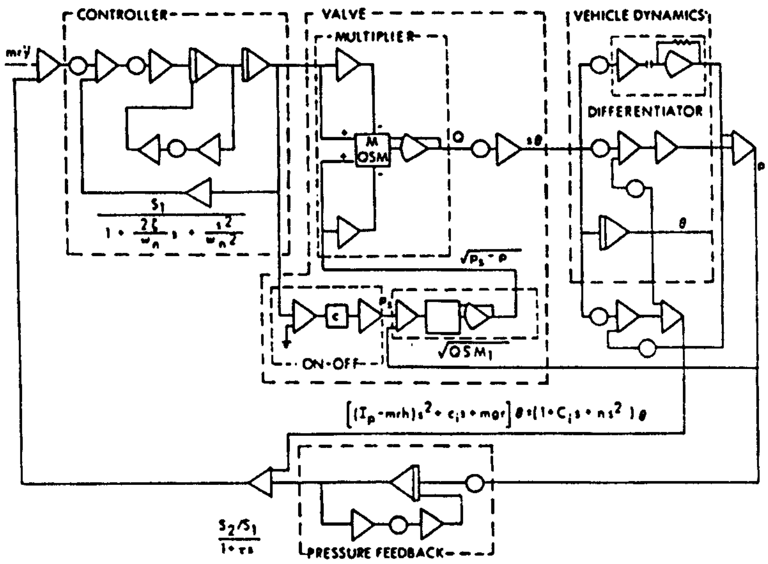


Fig. 9 Analog computer simulation of vehicle

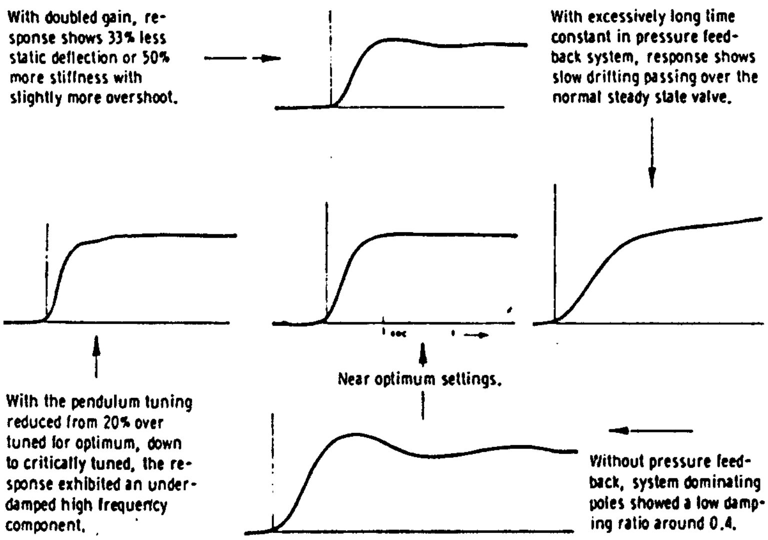


Fig. 10 Simulated system response to a step torque disturbance

cesses. However, detailed discussion of this is beyond the scope of this paper.

Bench Test of the Control System. Fig. 11 shows the general view of the controller of the experimental system. A set of test records of impulse function response of the vehicle controlled by this controller is shown in Fig. 12. Like the arrangement of Fig. 10 the middle diagram shows the response with near-optimum setting which agrees quite well with the behavior indicated by the transient response shown in the middle diagram in Fig. 10. However, stabilized operation with the actual system did not quite materialize after extensive experimenting with various types of dampers needed to eliminate the elastic modes which were not incorporated into the computer setup. By and large, a damper is not essential for the rigid body mode of the vehicle as demonstrated with computer study but is very important during the experimental phase when the pendulum is coupled with the hydraulic valve.

Road Test. The test vehicle illustrated in Fig. 4 is modified from a Yamaha motorcycle with a rebuilt rear section to incorporate a pair of wheels operated by the controller. To further minimize the disturbing effects due to the undulation of the road surface, an unloading valve was installed close to the hydraulic ram to relieve, through a bypass, the excess flow generated when the wheel is going over a bump. A manual control coupled to the pendulum sensor through flexible cables was also incorporated for the rider to roll the vehicle in anticipating a turn.

The vehicle system was not instrumented to make a quantitative road test study. For this reason only qualitative observations were established. The performance of the system can be described for the various modes of operation as follows:

(*a*) Stability while standing still and at low speed: With a rider on the vehicle, the system exhibited a fairly strong coupling between man and machine. If the dominating poles of the vehicle system are somewhat under damped, then with the coupling of the additional elastic mode of the human body, the entire system may exhibit slowly diverging oscillation. Man can exercise some muscle control to stabilize the system. However, even when the complete system is marginally stable, it gives the rider an uneasy feeling, as though the machine has a character of its own. Fortunately, the controller adjustment has enough range to pro-

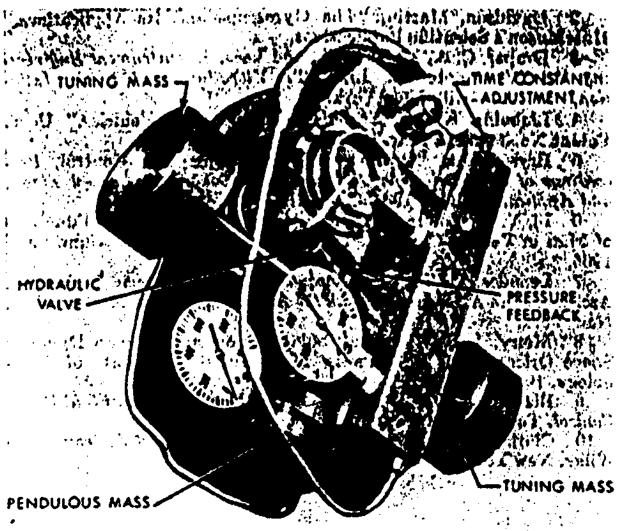


Fig. 11 Experimental hydraulic controller for apparent vertical

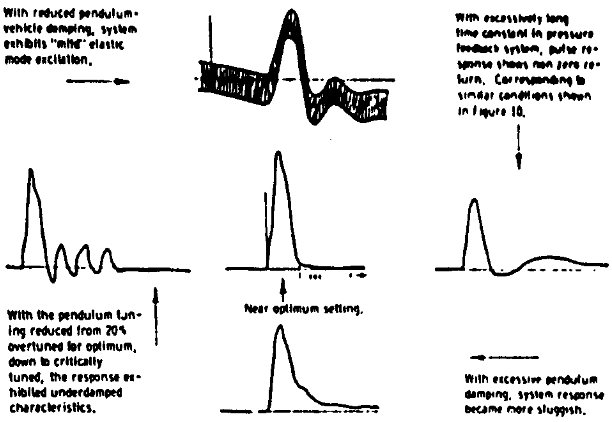


Fig. 12 Response of the experimental vehicle to an impulse function applied to the pendulum sensor

vide sufficient damping characteristics to lend the system insensitive to this coupling effect.

(*b*) Performance at moderate to high forward speed: This is an area worthy of a considerable amount of further research. In the first place, with an experienced motorcycle rider, a new technique must be acquired to fully develop the potential of this new type of vehicle. On the other hand, the vehicle parameters must be optimized to be adapted to the human response. As anticipated, the new vehicle gave the rider the best satisfaction when he had the manual roll mode control and the automatic roll control. Choosing between the two types of control, independently, the manual control gave better satisfaction than the automatic roll mode control.

It is also interesting to note that when the power of the control system was cut off to eliminate both the manual and automatic controls, the vehicle could still be operated like a motorcycle except that it appeared to be somewhat sluggish. On the other hand, even with manual roll mode control, the rider was not able to roll the vehicle readily away from a coordinated turn even though he was consciously trying. This illustrates the result of a deep-rooted reflex of the rider through long periods of training in the proper mode of operation. For this reason, the advantage of the automatic roll control should be explored with a person who has no prior experience with motorcycles. With some modification to improve the safety measures, this vehicle can be a very useful piece of equipment in the study of man-machine control behavior involving multiloop configuration and the learning processes.

(*c*) Roll mode ground disturbance: For the demonstration of the isolation of the roll mode ground disturbance, the vehicle was driven with one rear wheel over a 6-in. ramp at 20 mph. Under these conditions, the wheel base tilted about 10 deg while the observed body tilt was only about 2 deg. This transient response agrees in general with the frequency response in Fig. 2.

Conclusion and Future Potential

Even without extensive instrumented road tests, the performance of the experimental vehicle demonstrated quite clearly the feasibility of an active roll mode suspension system. Further improvement of the dynamic performance presents little problem, and the economic considerations for a production model do not seem to be major obstacles either, because the extra cost of the controller can be balanced by the saving that comes from the simplified front end assembly, the lesser number of doors and seats, etc. The major appeal of this type of vehicle rests upon its greater maneuverability and the pleasure of driving a vehicle with a perfectly coordinated turn and a smoother riding quality. To realize any degree of public acceptance, however, the key issue is the question of safety. This problem can be divided into two parts. First is the consideration of safety on the road, and second is the consideration of safety in design.

For driving safety, one obvious concern about a narrow vehicle is the question of its smaller size which seems to be a disadvantage when in collision with a larger car. However, the narrow width and tapered front end greatly reduce the chance of a head-on collision. The general reduction in the exterior dimension of a narrow vehicle also greatly increases its structural strength, thus providing better body protection.

Another important concern might be the notion that a narrow vehicle would be easier to roll over on treacherous road surfaces while making a turn. In fact, the chance against rolling over in a narrow vehicle with an active roll mode suspension system can be better than that of a wider car with passive suspension system. Since the possibility of overturning depends upon the margin of the distance of the apparent vertical from the edge of the wheel base, for a narrow vehicle, it is usually half of the width of the wheel base under the cg, while for a wider car with a passive suspension system, this margin can be even less than that of the narrow vehicle in a sharp turn. Furthermore, when an ordinary car rolls over, it always rolls toward the outside of the curve exposing the weak top of the car to any obstacle in its sliding path. Also, an ordinary car can continue the tumbling motion quite freely due to its barrel shape. In either situation, it is very dangerous to its occupants. In the case of a narrow vehicle, a roll over would probably happen if the wheels lost their grip while making a sharp turn. As a consequence, it might fall toward the inside of the curve and slide with the bottom out. In this posture, the occupant is better protected than in the situation described earlier for the ordinary car.

In considering the mechanical safety inherent in the system design, one major concern is the occurrence of a power failure. As illustrated earlier, the narrow vehicle can be operated quite adequately like a motorcycle without the power for the control system while running and with hand brake to control the tilt while standing still. In the case of the monorail version, a power failure would allow the side wheels to be locked to the monorail.[3]

In general, with the incorporation of an active roll mode suspension system, the vehicle should still be designed conservatively so that in the event of a failure of either side, all wheels would remain in contact with the ground. The two types of vehicles described in this paper illustrate the extreme cases which can be ben-

[3] For more detailed discussion of monorail adaptation, refer to reference [1].

elitted by the active roll mode suspension system. The general principle introduced here can be applied to a car with any width to improve its riding comfort and maneuverability and to military or farm vehicles to improve their performance over rough terrain.

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Vehicle and controller designs are covered by pending patents filed by Y. T. Li.

References

1 Li, Y. T., "Stability and Controllability of Vehicles for High Speed and High Traffic Permeability." IFAC Tokyo Symposium, 1965; also presented at Society of Automatic Engineers, Automotive Engineering Congress, January 10, 1966.

2 Davidson, Martin, "The Gyroscope and Its Applications," Hutchinson's Scientific Publications.

3 Draper, C. S., McKay, W., and Lees, S., *Instrument Engineering*, Vol. IV, Chapter 35. Li, Y. T. *Vibration and Vibration Isolation*," McGraw-Hill, New York.

4 Linbolm, K. J., "Stabilizing Devices for Vehicles," U. S. Patent No. 3083027, Mar. 1963.

5 Bryson, A., "Optimum Programming and Control, *Proceedings of IBM Scientific Computing Symposium on Control Theory and Applications*, Oct. 1964.

6 Li, Y. T., Young, L. R., and Meiry, J. L., "Adaptive Functions of Man in Vehicle Control," IFAC (Teddington) Symposium, Sept. 1965.

7 Leondes, C. T., ed., *Computer Control Systems Technology*. Aseltine, John A., *Synthesis With the Aid of Computers*, McGraw-Hill, New York.

8 Meiry, J. L., "The Vestibular System and Human Dynamic Space Orientation," ScD thesis, Massachusetts Institute of Technology, 1965.

9 Blackburn, J. F., Reethof, G., and Shearer, J. L., *Fluid Power Control*, Technology Press and Wiley.

10 Clark, R. N. *Introduction to Automatic Control Systems*, John Wiley, New York.

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(Continued on p. 318)