

UC Berkeley

Working Papers

Title

Combined Brake and Steering Actuator for Automatic Vehicle Control

Permalink

<https://escholarship.org/uc/item/9366m38b>

Authors

Prohaska, R.
Devlin, P.

Publication Date

1998-08-01

CALIFORNIA PATH PROGRAM
INSTITUTE OF TRANSPORTATION STUDIES
UNIVERSITY OF CALIFORNIA, BERKELEY

Combined Brake and Steering Actuator for Automatic Vehicle Control

R. Prohaska, P. Devlin

**California PATH Working Paper
UCB-ITS-PWP-98-15**

This work was performed as part of the California PATH Program of the University of California, in cooperation with the State of California Business, Transportation, and Housing Agency, Department of Transportation; and the United States Department Transportation, Federal Highway Administration.

The contents of this report reflect the views of the authors who are responsible for the facts and the accuracy of the data presented herein. The contents do not necessarily reflect the official views or policies of the State of California. This report does not constitute a standard, specification, or regulation.

Report for MOU 331

August 1998

ISSN 1055-1417

Combined Brake and Steering Actuator for Automatic Vehicle Research*

R. Prohaska and P. Devlin[†]
University of California PATH,
1357 S. 46th St. Richmond CA 94804

December 8, 1997

Abstract

A simple, relatively inexpensive combined steering and brake actuator system has been developed for use in automatic vehicle control research. It allows a standard passenger car to switch from normal manual operation to automatic operation and back seamlessly using largely standard parts. The only critical components required are two valve spools which can be made on a small precision lathe. All other parts are either off the shelf or can be fabricated using standard shop tools. The system provides approximately 3 Hz small signal steering bandwidth combined with 90 ms large signal brake pressure risetime.

1 INTRODUCTION

Experimental research on automated highway systems is presently somewhat constrained by the limited availability of vehicles equipped to be controlled by automatic means. We present here a simple method for actuating brakes and steering on a standard passenger car. The apparatus described should not be employed on public roads. It is for experimental purposes only.

Throttle actuators (in the form of cruise control) have been available for some years now, and high performance throttle actuators have been developed. Mechanisms to allow automatic application of brakes and steering are comparatively scarce, since appreciable power has to be delivered and relatively short response time (.1 sec or less) is required for precise control of vehicle speed and intervehicle spacing.

The most direct approach to delivering and controlling the substantial forces required for steering and brake actuation is the use of hydraulic power assist. Modern power steering systems already contain most of the required elements

*This work was supported by the California Department of Transportation

[†]Permanent Address: 1384 Jefferson Avenue, Akron Ohio 44313

and it happens that hydraulic brake assist is commercially available and easily modified for automatic control.

The use of existing ABS is appealing, but most ABS systems are carefully optimized for cost under the assumption that activity will be infrequent and brief. The demands of routine braking, and the consequences of controller misbehavior, are almost certain to exceed the design capacity of an ABS system.

Obviously high performance servomechanisms developed for aircraft and industrial applications are available but their cost readily exceeds that of a modern automobile. Usual high-performance actuators are not easily integrated with the steering and brake systems of a production vehicle since they usually rely on constant pressure hydraulic control. In constant pressure systems the control valves idle in the closed position and an accumulator stores hydraulic energy which can be applied to move masses quickly. In the inactive state there is appreciable power dissipation from valve leakage and some sort of pressure regulation system is required.

In contrast, standard hydraulic power assist schemes for automotive steering and brakes use constant flow designs; they offer acceptable performance at minimum cost. In a constant flow system the control valves idle in the open position with fluid circulating freely through the pump and valves. The parts count is minimal and by careful design the idle state power dissipation can be made thermally insignificant. However there is no accumulator, limiting the peak power available and therefore making the system response slow.

We chose to implement a constant flow actuator system. This minimizes the mechanical disruption to the vehicle and also allows a smooth transition between automatic and human-controlled modes of operation. Figure 1 gives an overall view of the engine compartment and makes clear that no substantial rearrangement is required.

For the PATH steering and brake actuator system we have used a standard hydraulic power assist unit (Bendix Hydro-Boost for the brake, the stock power steering box for the steering) and the existing power steering pump. The normal hydraulic passages between the integral control valves and actuator cylinders are blocked, and the lines brought out to external manifold blocks containing directional valves (to switch between normal and automatic mode) and proportional valves (which do the controlling in automatic mode).

The proportional valves are operated by analog electronic circuits delivering a few watts of electrical power. Closed loop control is achieved by adding a pressure sensor on the brake system and a position sensor on the steering system. The use of analog electronics on the inner loop may seem primitive at first, but it greatly simplifies initial setup and testing. The analog feedback circuit can be easily replaced by a digital one if more sophisticated control is desired.

The constant flow control valves use standard constant-pressure spool valve bodies but custom spools. Valve bodies are essentially impossible to craft by non-specialists but an acceptable spool can be made by a skilled machinist with little more than a good lathe and some patience.



Figure 1: View of the left side of the engine compartment showing minimal disruption of the stock configuration. The only changes are removal of the cold air intake duct and removal of the original vacuum operated cruise control.

2 Technical Description.

The complete hydraulic diagram is shown in Figure 2. All valves are in positions corresponding to human control, with no command applied. There is a free path from the pump, through the brake booster into the steering gear, whence the fluid is returned to the tank. The filter and cooler are omitted for clarity.

The descriptions begin with normal (human) control of the vehicle and then elaborate on changes required to permit automatic control.

2.1 Brake Servo and Actuator

Brake actuation is achieved through use of a Bendix Corporation Hydro-Boost power brake servo, modified for external control. The Bendix Hydro-Boost is a constant flow hydraulic brake servo designed for placement upstream of standard power steering. Since it was meant to be substituted in place of vacuum boosters in existing vehicle designs it is relatively easy to adapt using bolt plates, spacers and pedal rods to a variety of cars. Some cars which are equipped with Hydro-Boost from the factory include late '70s Cadillac Eldorados, mid '80s Lincoln Mark 7 LSCs and mid to late '80s GM diesels. The units cost between \$300 and

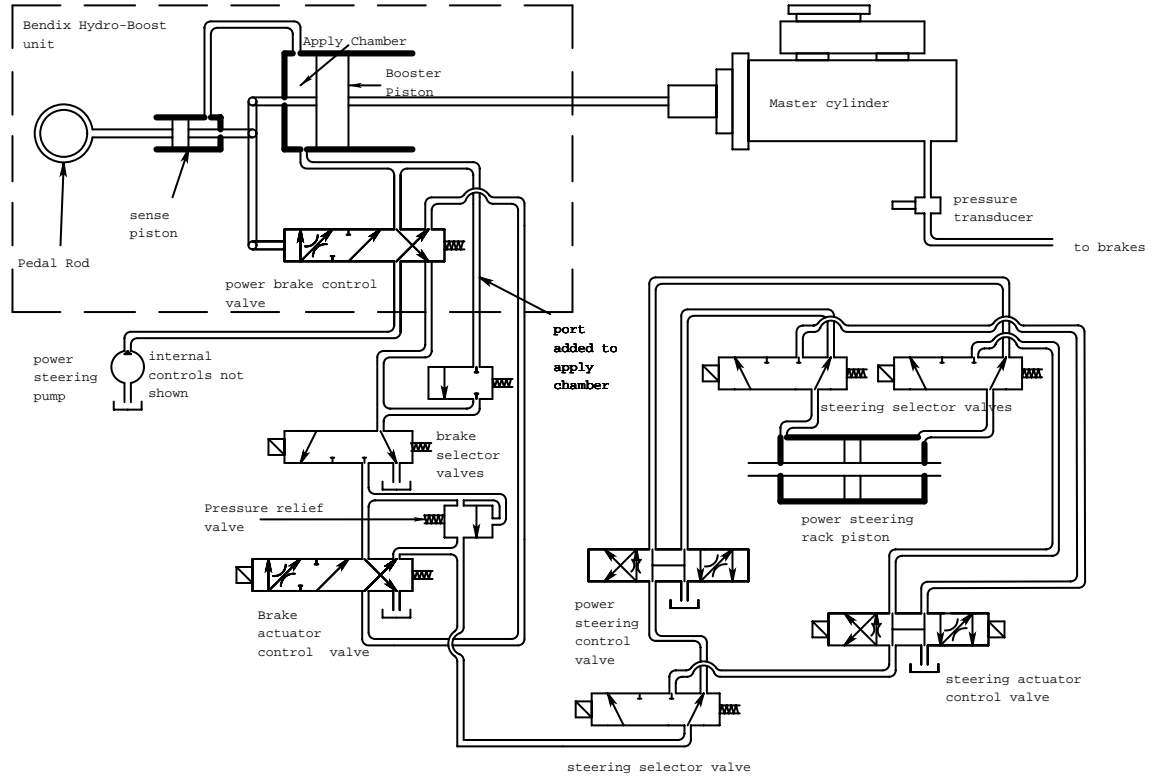


Figure 2: Complete hydraulic diagram for the combined actuators. See the spool construction discussion for a more intuitive representation of the control valves.

\$500 as replacement parts.

In manual operation, the Hydro-Boost receives flow from the power steering pump. With no pedal effort, fluid travels freely through the Hydro-Boost control valve and enters the steering system. The apply chamber is vented to the tank, and an isolation valve allows the steering system to develop normal operating pressure without inadvertently applying brakes. When the driver applies the brakes the power brake control valve senses displacement of the brake pedal closing the vent, opening the isolation passage and restricting fluid flow from the pump to develop pressure in the apply chamber. The booster piston applies force to the brake master cylinder, while at the same time a sense piston on the pedal input rod opposes the driver's foot. When the hydraulic force developed on the pedal overcomes the driver's effort, the power brake control valve returns to neutral position, halting the pressure rise and giving the driver tactile feedback. When the driver releases the pedal, a relatively soft spring returns the valve to the vent position, closing the isolation valve and releasing pressure in the apply chamber. The hydraulic force gain is simply the ratio of apply piston



Figure 3: Figure 3. The brake manifold block, looking toward the rear of the car.

area to sense piston area, typically about 10. The pedal rod and apply piston are linked by a pushrod, providing manual brake application in the event of hydraulic failure. The sense piston corrects for back pressure generated by the steering system, which would otherwise appear if the driver applied the brakes lightly and then turned the steering wheel.

To apply the brakes via an external control valve one must block the vent, bypass the isolation valve and restrict pump flow to develop pressure. Releasing the brakes is accomplished by restoring isolation and venting. All three functions are accomplished within the manifold block illustrated in Figure 3.

The function of the sense piston on the pedal rod is taken over by an electronic pressure transducer mounted on the brake master cylinder output. It's visible in the photo between the manifold block and the brake fluid reservoir. It detects the actual pressure developed in the brake system and generates a feedback signal used to control the actuator spool valve position. Excess pressure whether from control valve overshoot or from steering system operation is detected and relieved.

The only point of dissatisfaction is that driver power assist is lost once automatic mode is invoked: The apply chamber is vented unless the actuator is trying to apply the brake. As a practical matter, a driver of normal strength cannot rapidly stop the test car: gradual stops are possible but generous space



Figure 4: The steering actuator control valve, viewed from the front of the car. The manifold block extends about 4 inches below the control valve.

should be allowed for initial testing.

2.2 Steering Servo and Actuator

In manual operation hydraulic fluid enters the power steering control valve and is immediately vented to tank, with both sides of the power steering rack piston at tank (atmospheric) pressure. When the driver applies steering effort, deformation of a torsion bar in the power steering control valve results in one side of the rack piston being preferentially connected to pump output, the other remains connected to tank, whereupon further steering effort restricts pump flow, elevating pressure and generating net force on the steering system. There is no feedback to the driver, so steering feel is highly dependent upon valve profile and torsion bar stiffness.

To invoke automatic control of steering the normal flow of hydraulic fluid through the power steering control valve mounted on the steering input shaft is diverted to the steering actuator control valve. Pressure is delivered to the rack piston through selector valves via external hydraulic lines. The steering selector valves isolate the actuator system from the power steering system when the car is in manual operation. Figure 4 gives some sense of the layout; the control valve is being pointed to, the three black solids with hex nuts on top are



Figure 5: Steering gearbox showing hoses added to rack piston cylinder.

the tops of the directional valves. Figure 5 illustrates the sort of modification required on the steering gearbox; fittings are simply tapped into the casting.

A steering position sensor (in this case, a linear potentiometer mounted on the idler arm) provided position feedback which is compared to the command signal and the difference is used to generate signals to drive the solenoids which operated in push-pull mode.

If power fails to the selector valves the car reverts to normal power steering. Power failure to the actuator control valve is equivalent to manual steering with no power assist. The only ill-behaved power failure mode is to the position sensor: the car turns to one side, in this case to the right.

In automatic mode the driver retains mechanical steering connection but no power assist. As a practical matter, that means the driver has no steering: the torque of the actuator exceeds that of any normal driver. In the event of actuator (or software) malfunction, the only recourse is to shut off the actuator and recover control of the car using normal power assist. The transition between actuator and normal power assist is smooth; the only difficulty is in remembering to shut off the actuator.

3 Construction Details

The vehicle used to test our design was a 1989 Ford Crown Victoria. It was equipped with factory power steering and a vacuum brake booster. The vacuum brake booster was replaced with a Hydro-Boost unit modified with a larger pedal rod eye to fit the existing pedal assembly, mounted by means of a bolt plate which allowed the somewhat strange Hydro-Boost bolt pattern to fit the firewall. Internal passages between control valve and apply chamber were blocked and the hydraulic signals were brought out to a separate manifold block containing directional valves (which select between manual and automatic operation) and the actuator control valve.

A relief valve was included to ensure that hydraulic flow to the steering cannot be cut off by a saturated brake actuator. The relief pressure is lower than that of the hydraulic pump's internal relief valve and ensures that an errant brake command cannot disable the steering.

The steering gearbox was modified by blocking the normal internal passages between the control valve and rack piston. External ports were added to divert the output from the power steering control valve to the manifold block holding the control and selector valves. Directional valves then connected the rack piston to either the actuator control valve or to the power steering control valve.

The tank connections of both the steering actuator control valve and the power steering control valve were brought out to a common connection for routing through a filter and cooler, whence the fluid returned to the tank. The filter was simply a precaution but the cooler was necessary: continuous application of brake (whose flow then transits the steering system) could easily raise fluid temperatures to destructive levels.

Most plumbing external to the existing components was constructed in two manifold blocks, one for steering and one for brakes. The Hydro-Boost unit's need for an external tank connection was satisfied by replacing the Ford's stock power steering fluid reservoir with a reservoir from a Hydro-Boost equipped Lincoln Continental Mark 7 LSC, which contained the required connection.

3.1 Spool Construction

The fabrication of the valve spools is a critical step in the project. Manufacturers do not supply complete valves suitable for this kind of service, and most are relatively uncooperative at providing things like internal dimensions and spool blanks. A vendor willing to provide such help is worth cultivating if found.

The valve bodies we used are standard Vickers parts: KTG4V-3 for the brake, and KDG4V-3 for the steering. The valves differ most notably in that the former is a unidirectional valve, while the latter is a symmetric design for push-pull application.

Unfortunately Vickers declined to provide specifications on the internal dimensions, so they had to be measured manually. This was done by using a vernier caliper as a depth gauge and measuring to small blocks pressed against the lands inside the valve cavities. The resulting accuracy was adequate when

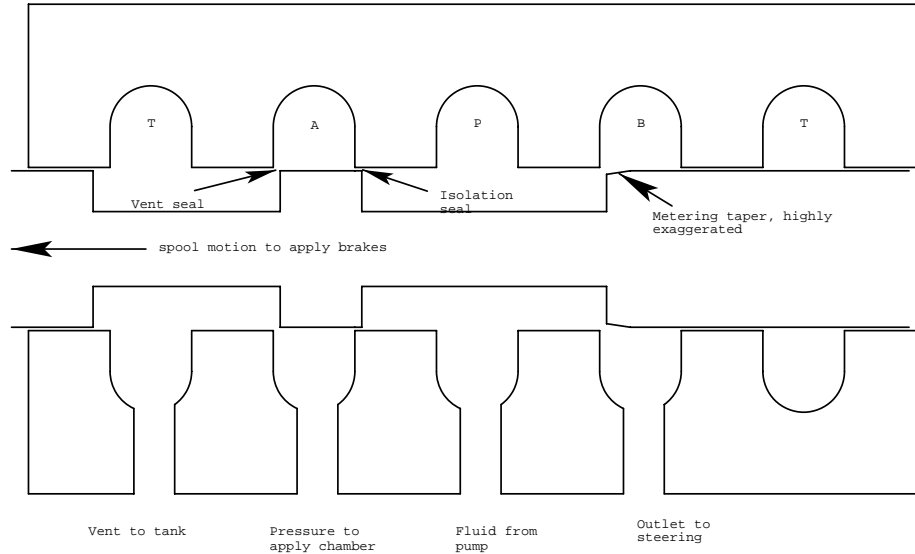


Figure 6: Stylized representation of the brake actuator control valve. Radial clearances are highly exaggerated

the measurements were repeated by different people and averaged. The brake valve spool and body are illustrated in a stylized way in Figure 6.

In the relaxed state fluid enters from the pump and exits at port B to the steering gear. The land occupying port A isolates pressure developed downstream by the steering system and vents the apply chamber to tank. As the spool is moved to the reader's left the land in port A first closes the vent, then connects the apply chamber to the pump at port B. As the leftward displacement increases the metering taper enters the passage between P and B, raising pressure while the fluid continues out port B. That pressure is communicated to port A where it applies the brake.

The steering valve shown in figure 7 differs only in that the spool is symmetric and the isolation valve seal becomes bidirectional to select the direction in which steering motion occurs. When idle, fluid enters from the supply and is immediately vented to tank symmetrically on both ends of the spool. When the spool moves left, the central land first closes the left rack piston port A leaving port B (the right rack piston port) connected to the pump. Further leftward movement drives the metering taper into the right hand tank connection to develop pressure which communicates to the steering rack via port B.

The central switching land deserves special attention: One would like the valve to offer minimal flow impedance when the valve is idle, but also to require minimal displacement to generate threshold pressure (minimum deadband). The establishment of a suitable compromise is to some extent a matter of trial and error. If the land is too wide the valve will exhibit excessive pressure drop

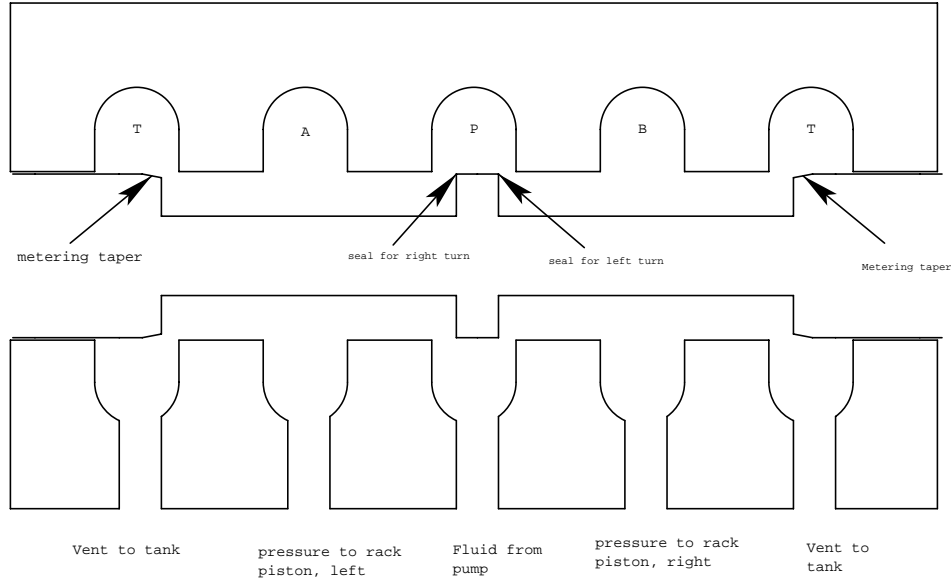


Figure 7: Stylized representation of the steering actuator control valve. Radial clearances are highly exaggerated

in the idle state, if it's too narrow excessive displacement will be required to get any response at all and that response is apt to be too big, since the metering taper will already be engaged. One strategy is to make the land too wide initially, then trim it until the idling pressure is acceptable (ie., the fluid does not overheat). If the minimum pressure is too high, the metering taper is then too close to the seal. That defect can be remedied by further machining of the taper, at the cost of requiring more spool travel to achieve maximum pressure. When the spool has been machined to the point that maximum pressure cannot be achieved the spool must be discarded and a new one fabricated.

The radial clearance and metering taper are highly exaggerated in the figures. The metering taper amounts to no more than 0.010" over 0.050" of length. On the scale of the figure such clearance is invisible.

The metering taper is computed from the known flow rate of the pump, in this case about 6 liters per minute, the maximum displacement of the spool with full coil drive and the desired maximum pressure. The desired taper is one which gives a hydraulic pressure proportional to solenoid current which we have approximated using 3 conical sections. The profile is constructed by first cutting a 1 degree taper on the nominal spool diameter (0.470") with a length of 0.050" measured from the small end. Next a 3.5 degree taper is cut to a length of about 0.020" from the small end. Finally, a 14.5 degree taper is cut to a length of 0.012" from the small end.

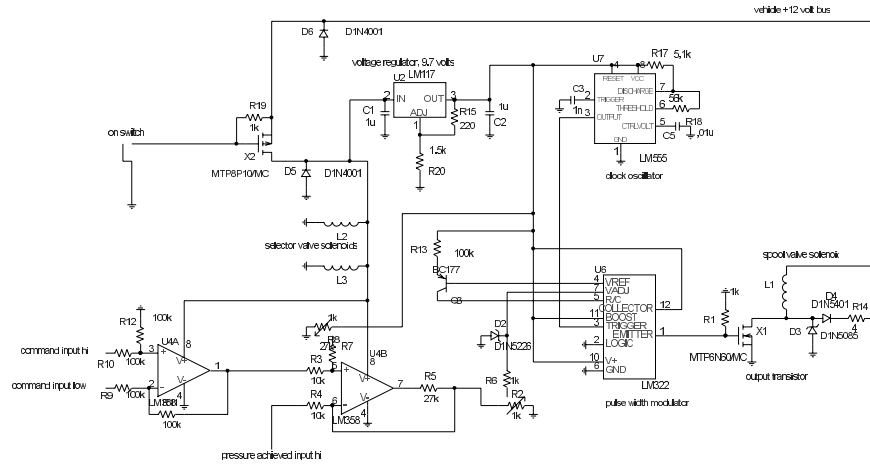


Figure 8: Brake electronic circuit, showing error amp, oscillator and pulse width modulator. External connections are not shown

3.2 Electronics

The electronic circuits used are the simplest possible which offer reasonable performance. Both brake and steering circuits are housed in small boxes hidden inside the front of the passenger compartment to keep the electronics away from engine heat. Both brake and steering control circuits implement a simple P-control, in which actuator drive is proportional to the difference between command input and response.

The brake control circuit is the most intuitive and is illustrated in Figure 8. The circuit consists of an error amplifier, clock oscillator and pulse width modulator. The modulated pulse width is delivered to the gate of a power MOSFET which controls the brake solenoid current and hence the brake pressure.

The steering circuit is a bit less obvious in operation since it does not use a fixed frequency clock and does not employ an explicit pulse width modulator. The circuit compares position to command at U3B. The difference, relative to midrange (5 volts) is delivered to U7B and U4A, which form a phase splitter, determining which solenoid will be driven and how hard. The comparators at U5A and U5B function as latches, to ensure that only one solenoid is on at a time.

The amplified error is delivered to U6A and U7A, where it is compared to the current already flowing in the solenoid coil. If the current in the coil is lower than desired transistors X1 or X2 are turned on.

The circuit looks like, and in principle is, a linear amplifier. The pulse width

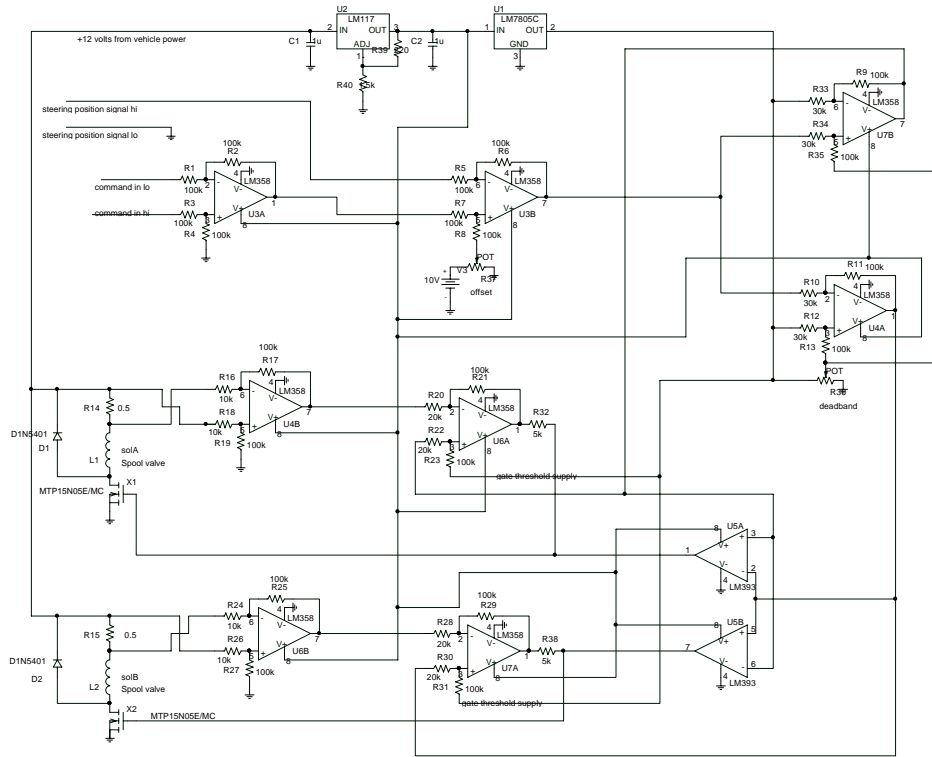


Figure 9: Steering actuator control circuit showing error amp, coil current sense amps and direction latching comparators.

modulation arises from the gain of the op amps and high transconductance of the FET output stage combined with delays associated with coil inductance and mechanical inertia. The transistors are always either full on or full off, with the result that power dissipation in the transistors is nil. The resulting oscillation is filtered out by the mass of the mechanical parts. It might fairly be said that the valve spool is hammered into position by the electronics.

R36 sets a voltage which must be exceeded by the error (deadband) before any corrective action is taken. It is this feature which forces the oscillations to halt when the desired position is achieved.

4 Performance

The small-signal performance of the steering system is illustrated in Figure 10. The vertical scale is in volts, with .5 volt (one division) corresponding to about 6 degrees of steering knuckle angle. The vehicle was stopped, so the turning forces required were larger than would prevail under normal operation. At

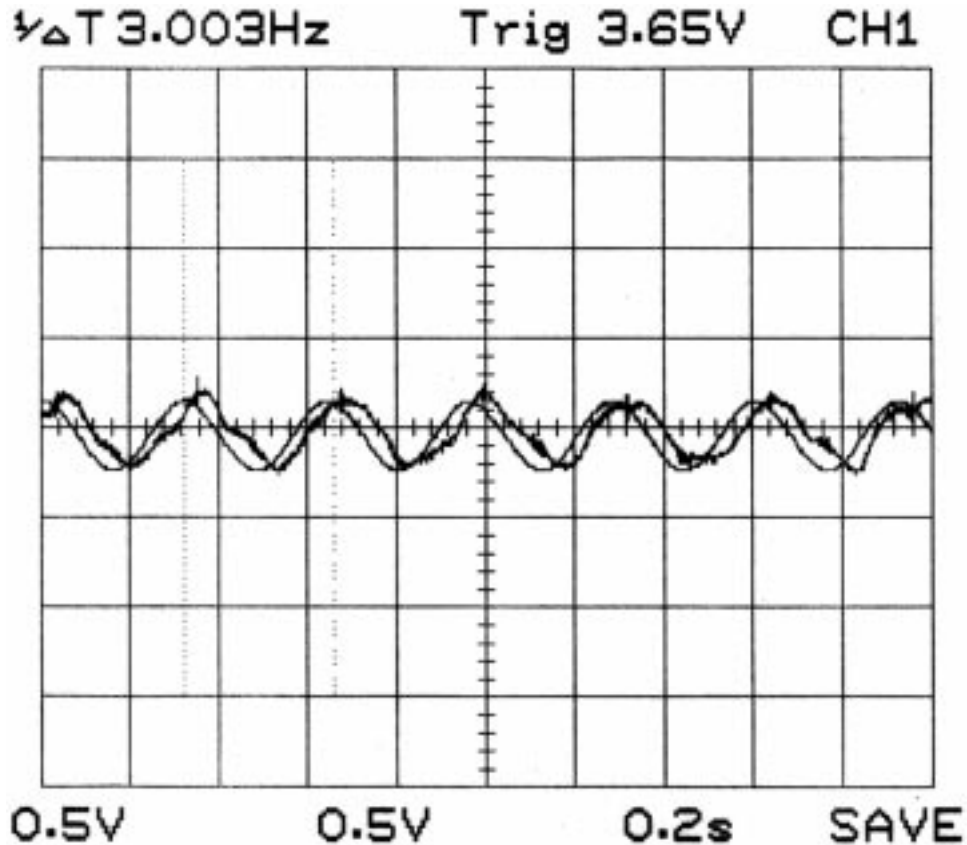


Figure 10: Figure 10. Small-signal steering response. The smooth curve is the command, with one vertical division corresponding to about 6 degrees of steering steering knuckle angle. The timescale is 200 milliseconds per division.

3 hertz the lag is noticeable, at about 4 hertz internal modes take over and command following ceases.

Figure 11 illustrates steering behavior at larger amplitudes. Here the limitation is slew rate, about 80 degrees per second measured at the steering knuckle, as evidenced by the nearly triangular position plot.

This can be helped considerably by increasing the flow rate in the hydraulic system, but it's already some 20% higher than normal (achieved by drilling out the flow control orifice) and further increase invites pump trouble.

Large signal brake response is illustrated in figure 12. Here the command corresponds to slightly over 800 psi with the response settling within about 120 milliseconds

Small signal brake behavior is shown in figure 13. The step input corresponds to about 20 psi command input, and the response can be seen to settle

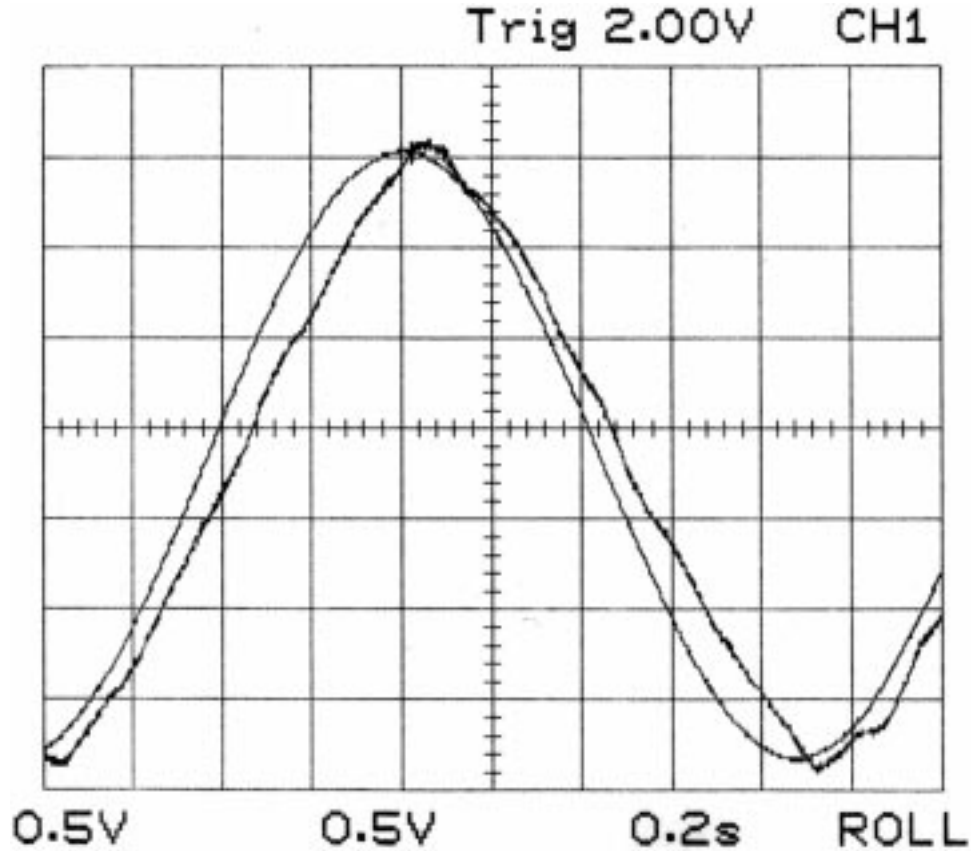


Figure 11: Figure 11. Large signal steering response, showing slew rate limitation.

within about 100 milliseconds. but the offset becomes comparable to the command. This offset is an artifact of the electronics and pressure sensor, not of the actuator itself. It happens that at about 30 psi the brakes go from "off" to "on" as tested by hand, so the offset isn't large.

5 Lessons Learned

In general terms, the brake actuator is satisfactory, the steering actuator somewhat less so.

The brakes offer reasonable time response and more than sufficient maximum amplitude; indeed, it's no problem to lock the wheels. In terms of finesse they are acceptable but only barely so. Smooth operation at low speeds demands considerable delicacy of braking, and the combination of offsets and thresholds exhibited by these brakes are not capable of rivaling the performance of a good

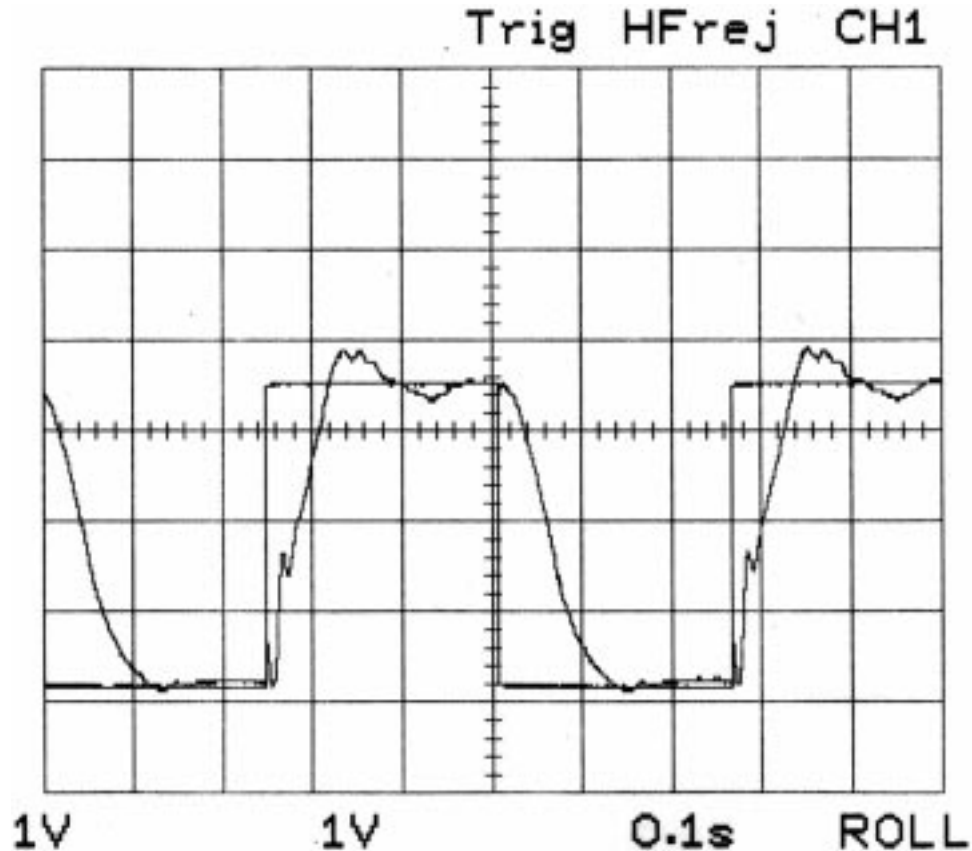


Figure 12: Large signal response of the brake actuator. The input steps are about 800 psi.

human driver. The remedies possible are two: reduce seal friction or add a direct torque measurements at the caliper. The seals are many, and mostly rather specialized; it's not likely that low friction versions of caliper seals and booster piston seal can be obtained. Even if they could, the behavior of the brake pads remains outside the control loop. If one has the choice, a direct measurement of brake torque is probably the best solution. With actual brake torque in the control loop this actuator would be satisfactory for any practical application.

The steering system is both poorer in performance relative to what's needed and harder to improve. One would like the steering to be faster than the normal modes of the car's suspension. This system is not. One would like the steering system to have a smooth transient response, this one does not. As it stands, the steering is adequate to study lanekeeping at speeds up to 50 miles per hour [1], but only barely so. A more clever inner loop would help, but can't really

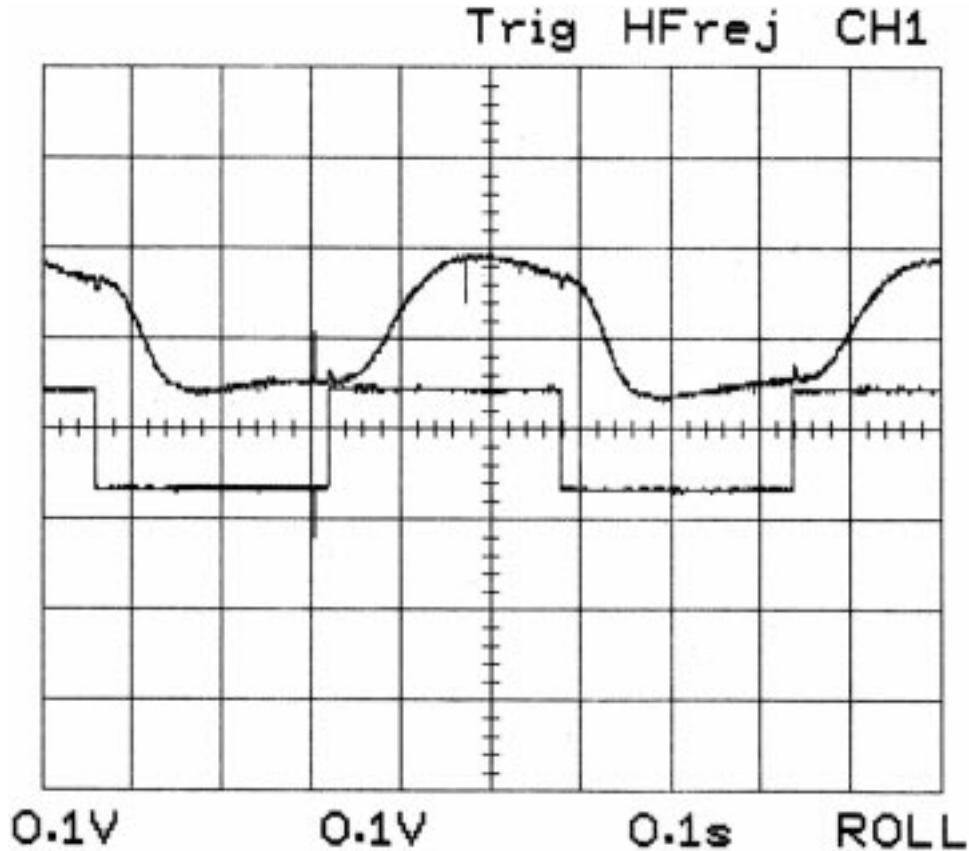


Figure 13: Small signal brake response. The input corresponds to only 20 psi.

solve the problems.

The limitations come from a combination of slow control valves and limited slew rate in the steering rack piston. The mechanical resonance of the steering wheel on the power steering control valve torsion bar is an unhelpful influence but performance didn't improve markedly when the steering wheel was removed (ie., a column clutch won't help). The remedies possible are: Increase pump flow rate, increase electrical drive to the control valves, or go to a more conventional hydraulic system using a constant pressure design and servo valves.

Of the three the last will probably be the most satisfactory. Using the standard power steering pump to charge an accumulator isn't difficult but constant pressure operation presents an entertaining problem of considerable import: How does one get back to human (constant flow) operation when the automatic controls are in a state of irretrievable disarray? The return to human control mode must be graceful and quick, a subject for future investigation.

6 Acknowledgements

The aid, counsel and graphic talents of PATH Publications Department staff (Gerald Stone, Esther Kerkman and Bill Stone) are gratefully acknowledged. Doug Wood of the RFS Maintenance Department constructed the manifold blocks from less than perfect drawings. Dan Marvin of Norman Racing in Berkeley, California constructed the valve spools (from somewhat more nearly perfect drawings).

References

- [1] Proceedings of the American Control Conference, Albuquerque, New Mexico, June 1997, pp. 1603 - 1607.