2002-01-1598

An Experimental Assessment of the Online Tuning of Active Suspension Controller Gains

Brent A. Clark and Ramavarapu Sreenivas
Department of General Engineering, University of Illinois at Urbana-Champaign

ABSTRACT

A model of an active suspension is developed, and a corresponding test stand is designed and constructed. The active suspension test stand is then subjected to a series of experiments to determine the feasibility of utilizing an online control scheme that is capable of automatically tuning itself for optimal performance. The experiments are designed to evaluate the control schemes utilizing both proportional and proportional plus integral plus derivative controller gain parameters. The methodology for online tuning of the gains proves to be useful with a considerable increase in ride quality for the scenarios tested.

INTRODUCTION

Since the fifteenth century when the first passenger coach was suspended from a frame by belts and chains [1], vehicles have been equipped with a means to cushion passengers from bumps and other roadway irregularities. The classic means of isolating the passenger from these disturbances has been a passive suspension, which is typically composed of some combination of springs and dampers. (The springs and dampers store and dissipate energy, respectively, but do not add any external energy to the system, hence the term “passive”). More recently, components have been added to passive suspensions that can provide external energy to reject road disturbances. Generally, this type of suspension is termed “active.”

One of the main objectives of an active suspension is to provide a better quality of ride, but an active suspension is accompanied by an increase in the degree of complexity over its passive counterpart; and, if improperly designed, the addition of active components could conceivably worsen ride quality.

BACKGROUND

The concept of a passive suspension can be found in an introductory text on controls or vehicle dynamics; however, an active suspension is the subject of this paper, so a brief overview of some common types of active suspensions is in order.

Some variations of active suspensions include self-leveling, semi-active, slow-active, and fully active [2]. From fully active to passive, the following categories of suspensions are defined in order of decreasing capabilities [2, 3]:

- Fully active – A hydraulic servo with a high frequency response provides control input to the system at all times. A fully active suspension is the focus of this paper.
- Slow-active – An actuator operates in series with a spring and provides control within the limits of its frequency response. Outside its limitations, it becomes rigid and no longer provides input to the system (the passive elements take over at this point).
- Semi-active – A semi-active suspension is capable of dissipating power but not supplying it. In essence, the actuator becomes a continuously variable damper that acts to absorb energy when appropriate.
- Switchable damper – The switchable damper is a relatively simple (but effective) method of control. The dampers are switched between two discrete states (“hard” and “soft”) dependent upon vertical wheel and vehicle velocities.
- Self-leveling suspension – This type of suspension implements a lift component (typically utilizing compressed air) to compensate for changes in load or operating conditions.
- Passive suspension – The typical automobile suspension—equipped with springs and dampers.

Although a passive suspension can be designed to meet certain performance criteria, the design objectives compete with one another, so there are tradeoffs between them. Alternatively, an active suspension has the capacity to enhance and control various performance aspects simultaneously (depending on the nature of the active system) [4]:

- Ride quality – A passenger perceives ride quality any time the vehicle is in motion—particularly when the ride quality is poor. Ride quality is the focus of this paper.

- Height control – Height control will maintain a "level" vehicle when heavy loads or braking cause the vehicle to pitch (see Figure 1). This will help to maintain braking performance and fuel economy (by maintaining a lower aerodynamic drag). It is also possible to reduce drag by lowering the vehicle at high speeds. Another possibility is to raise a vehicle when traveling on rough roads to increase ground clearance and suspension travel.

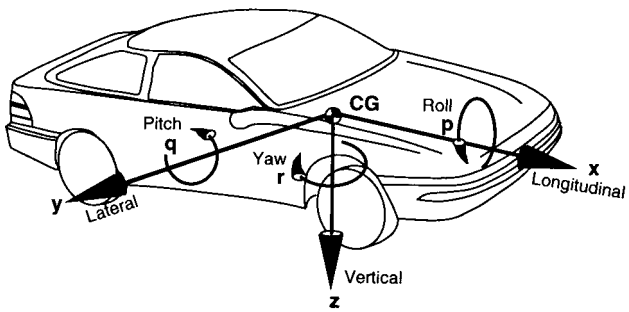


Figure 1: Society of Automotive Engineers vehicle axis system defining the yaw, pitch and roll directions [2].

- Roll control (see Figure 1) – This mode of control improves cornering performance by reducing load transfer in the roll direction.

- Squat/dive control – This type of control improves acceleration and braking performance by reducing forward weight transfer (dive, or a negative pitch) and rearward weight transfer (squat, or positive pitch).

The tradeoffs between the preceding performance criteria with a passive suspension can be overcome with an active suspension, but the economic trade-offs must be considered. The implementation of a fully active suspension requires the addition of expensive components (an electronic control unit, several sensors, high frequency response hydraulic servos, and an adequate power source to manage the system), so the technology currently only appears on expensive luxury and sports cars. However, there is generally a "trickle down" effect as technology advances, and standard passenger cars may be equipped with fully active suspensions in the near future.

OBJECTIVES

The purpose of this paper is to study the feasibility of the automatic tuning control parameters of the active suspension in real-time. This important feature will allow ride quality to be maintained throughout the life of the vehicle and suspension components.

As a vehicle's suspension ages, the suspension parameters change, and the control parameters must be adjusted. For instance, as a shock absorber ages, it will not be able to dissipate energy as effectively; and the control parameters must compensate for this. Also, as suspension bushings age, the system compliance will also change.

Although shock absorbers can be replaced rather easily, extending their usable life would have a positive economic impact on the end user, and ride quality could be maintained for a longer period of time. Other suspension components, such as the suspension bushings, can also change the system and ride quality will also be affected. Since aging bushings are not as easily replaced as an old shock absorber, automatic tuning of control parameters will allow these components to remain on the vehicle for a longer period of time. Automatic tuning will allow an active suspension to maintain optimal performance over the life of the vehicle—even as suspension parameters change.

To study the feasibility of automatically tuning the suspension in real-time, an active suspension testing apparatus was constructed and instrumented. The test stand was interfaced with an IBM-compatible PC and operated by VenturCom's RTX software (a real-time environment for Windows) in conjunction with a graphical user interface (GUI). The integration of RTX, a control algorithm (programmed in C), and the GUI (programmed in Microsoft Visual Basic) results in real-time control and automatic tuning algorithm that is straightforward to operate.

MATHEMATICAL MODEL

A quarter-car single degree of freedom model (see Figure 2 and Equations 1-2) is utilized to study the real-time tuning of an active suspension. Input is provided at the wheel (or unsprung mass—the mass that is not supported by the suspension, including one-half the mass of the suspension), and the output of interest is the displacement of the vehicle body (or sprung mass—the mass supported by the suspension plus one-half the mass of the suspension).

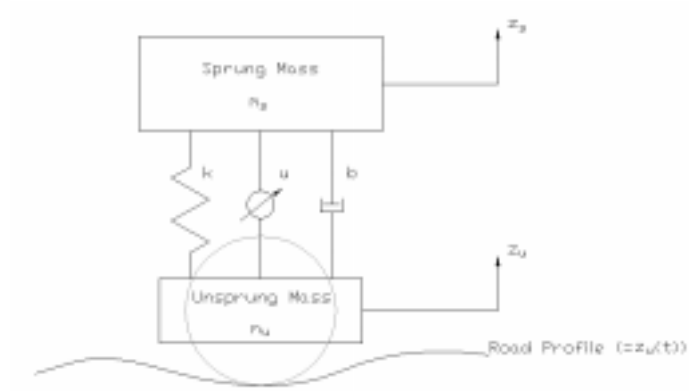


Figure 2: A quarter-car model with an active element.

Writing the equation of motion of the system (Figure 2):

$$m_s \ddot{z}_s = -k(z_s - z_u) - b(\dot{z}_s - \dot{z}_u) + u \qquad (1)$$

The input is provided to the system through the vertical velocity of the road input ($\dot{z}_u$), control effort is provided through *u*, and the output is the displacement of the sprung mass, (z_s).

ASSUMPTIONS

A few simplifying assumptions were made when modeling/designing the apparatus. The results will nonetheless translate to full-scale implementation on a vehicle if the proof-of-concept proves to be effective.

The usage of the quarter-car model is the first simplifying assumption, as it does not account for vehicle pitch (refer to Figure 1). In other words, only motion in the vertical plane (bounce) is considered. It is assumed that pitch modes (as well as the other performance capabilities outlined in the Introduction) can be controlled by the active suspension without difficulty.

Another simplifying assumption is the use of the single degree of freedom model, since it does not account for the nonlinear dynamics of the tire. Although tires also contribute to ride quality, the stiffness of the tire is great enough that tire dynamics can be neglected at lower frequencies (~1-3 Hz) [1]. Since the apparatus is subjected to no more than 4 Hz, the tire dynamics can be neglected, and the single degree of freedom model is suitable.

The test apparatus is subjected to only a relatively low range of frequencies, and not the entire range of frequencies that a typical automobile would encounter during normal driving conditions. The reason for the low range of frequencies is twofold: 1) it is difficult to safely and economically provide such high frequency system input in a laboratory environment; and (also due to economic factors) 2) the test apparatus is equipped with a DC servo motor with limited frequency response in comparison to a high frequency response hydraulic servo that is equipped on a vehicle with an active suspension.

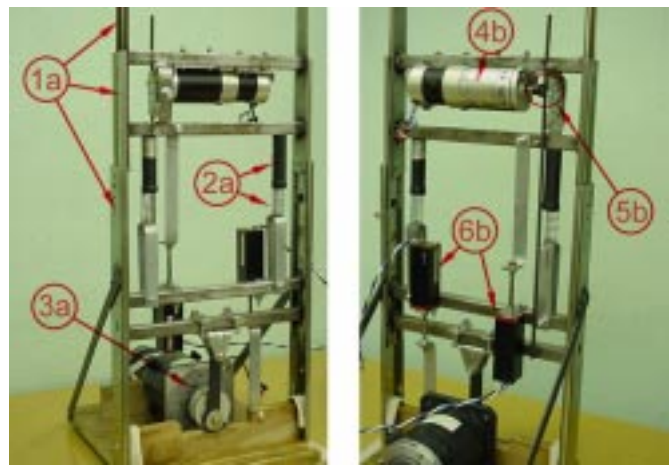
It is also assumed that the position of the sprung mass is known with respect to some fixed inertial frame of reference. This assumption allows the self-tuning concept to be evaluated without the addition of extraneous sensors and software algorithms.

The preceding assumptions merely act to simplify the model for feasibility assessment. A successful experiment on the proof-of-concept test apparatus will therefore translate to full-scale implementation.

PHYSICAL SYSTEM

An active suspension test stand was constructed (Figure 3) and instrumented for the automatic tuning of the proportional, integral, and derivative (PID) controller gains. The test stand was subject to the following requirements:

- It must meet budgetary constraints.
- It must support the weight of the sprung and unsprung masses while retaining them in a vertical plane.
- The support structure must allow the sprung and unsprung masses to move vertically with little friction.
- It must possess a means of periodically disturbing the system.
- There must be a means of measuring/computing the inputs and outputs of the system.
- It must be capable of rejecting inputs to the system.



(a) (b)

Figure 3: (a) A frontal view of the active suspension test stand indicating: (1a) ball bearing drawer slides, (2a) springs and dampers, and (3a) the input motor. (b) A rear view of the active suspension test stand indicating: (4b) the control motor, (5b) the rack and pinion, and (6b) the linear potentiometers.

The design consists of several main components (indicated in Figure 3) that satisfy the above requirements. The main components are: a pair of three-stage ball bearing toolbox drawer slides, two springs and dampers, an input motor, a control motor, a rack and pinion for the application of control effort, and two potentiometers. A synopsis of each of the main components follows.

BALL BEARING SUPPORTS

The pair of drawer slides provides a nearly friction-free mechanism and also act to confine motion in the vertical plane. The first stage of the slides is connected to the wooden base, the second stage is connected to the unsprung mass, and the third stage is connected to the sprung mass.

SPRINGS AND DAMPERS

Two springs and dampers are also connected between the unsprung and sprung masses. A pair of each is utilized to provide a symmetric force distribution between the two masses. The spring constants and the damping coefficients sum together, respectively. Hence, the presence of the “extra” spring and damper gives an equivalent system to the one derived earlier.

INPUT MOTOR

A DC motor is connected to the wooden base and provides input to the system via the unsprung mass. Input is accomplished by converting rotational motion into translational motion (see Figure 4). The implementation is similar to an internal combustion engine where the rotational motion of the crankshaft is converted into the translational motion of the piston along the cylinder. Impulses of varying frequency are provided to the system with this motor. The magnitude of each impulse is specified via the user interface and utilizes the road profile measurements.

CONTROL MOTOR

The control motor is a DC servo resting on the sprung mass (and doubling as the sprung mass). Actuation is provided via a rack-and-pinion mechanism. The pinion gear is attached to the control motor, and the rack is connected between the unsprung and sprung masses. So that the sprung mass can move freely along the rack when the system is in passive mode, the lower end of the rack is welded to the unsprung mass, and the upper end is left free. Since the upper end is left free, reaction forces must exist to restrain the rack from movement away from the gear. To provide the necessary reactions, an adjustable roller mechanism is fitted to the sprung mass. The rollers press against the backside of the rack so that it does not “kick out” from the gear (Figure 5). It is also necessary to constrain the rack in the lateral direction, so the roller fixture is attached with long screws that are fitted with spacers (Figure 5).

Figure 4: The input motor disturbs the system by converting rotational into translational motion.

Figure 5: The control motor is a DC servo that rejects disturbances with a rack and pinion.

POTENTIOMETERS

The two linear potentiometers provide data required to control both the input and output. The lower potentiometer measures the displacement of the unsprung mass with respect to earth-fixed inertial coordinates. In other words, the lower potentiometer provides the measurement of the road profile. This measurement is used to control the motor that provides input to the system, and is also used to calculate the earth-fixed position of the sprung mass. Note that this measurement is not available on a vehicle, but is used here to simplify the system (see the section dealing with the mathematical model for the assumptions).

The upper potentiometer is connected between the unsprung and sprung masses and measures the distance between the two. The sum of this distance with the road profile measurement gives the earth-fixed position of the sprung mass and is used to calculate the displacement from equilibrium and the resulting control effort.

SOFTWARE

The active suspension test stand is interfaced with an IBM-compatible PC and operated by VenturCom’s RTX software (a real-time environment for the Windows operating system). The RTX software works in conjunction with a control algorithm written in the C programming language and a graphical user interface (GUI) designed in Visual Basic. The result is the ability to provide real-time control and automatic tuning with an easy-to-use interface. This section details the overall operation of the software. For an in-depth treatment of the control algorithms, refer to the section on control.

REAL-TIME SUB-SYSTEM

The nucleus of the control algorithm and real-time processing is the RTSS file. Compiling and linking the

core C program and several “include” files creates the RTSS file. The RTSS file manages the sending of user input from Visual Basic to C and the sending of control effort and sensor output to Visual Basic from C (the process is executed with a 0.001 second sample period). The C program is the focus of the data acquisition and control routines since it contains the control algorithms utilized in this paper.

GRAPHICAL USER INTERFACE

Visual Basic provides the user with an interface (Figure 6) to VenturCom’s real-time sub-system (RTSS) and allows for easy operation of the test stand in both disturbance mode and automatic tune mode. The use of Visual Basic allows the user to send data to the RTSS process and view the output sent from the RTSS process.

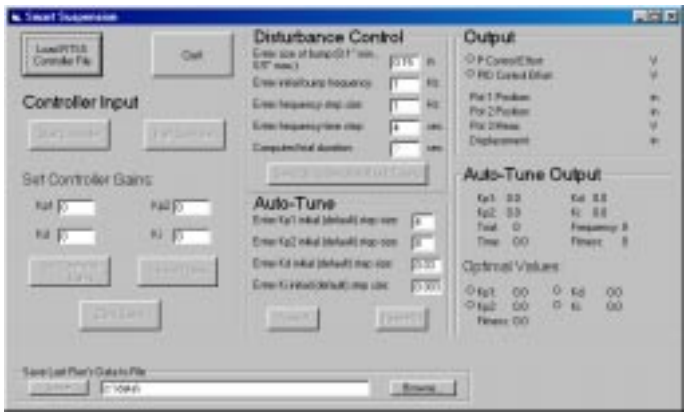


Figure 6: A screen shot of the Visual Basic GUI.

DESCRIPTION OF THE CONTROL ALGORITHM AND EXPERIMENTAL METHODOLOGY

Control was developed for both the input motor and the disturbance rejection motor. Once the control algorithms were in place, a methodology was developed for automatically tuning the controller gains. The following subsection describes the control used for inputting and rejecting disturbances, and subsequent subsections discuss the methodology utilized for automatic tuning of the PID gains.

INPUT CONTROL AND DISTURBANCE REJECTION

An algorithm was developed to provide impulses to the system according to the user-specified inputs (see the software section). Disturbance rejection was provided using one of two methods: proportional (P) or proportional plus integral plus derivative (PID) control. The means of disturbing the system is discussed here, and the method of controlling the system is discussed in following sections.

Input Control

The disturbances or “bumps” are provided as impulses of varying magnitude and frequency. In order to provide the proper bump size, the potentiometer measuring the distance between ground and the unsprung mass is utilized. A constant voltage is applied to the input motor until the appropriate magnitude has been reached. Voltage is then switched off, and the crank returns to its equilibrium position (the crank slightly rotates past bottom dead center—refer back to Figure 4). The voltage is then applied once every $1/f$ seconds.

After a series of preliminary experiments, it was noticed that one of the potentiometers, on occasion, outputs spurious voltages. This was attributed to mechanical causes. In order to maintain consistency in the control algorithm, the input control was modified to check the current status of the lower potentiometer. If the potentiometer reading was outside the range of possible values, the current potentiometer position was estimated. The estimation varies for differing bump sizes and can be determined by running the apparatus at a given bump size and analyzing the data. The system was subjected to a 0.35” bump size, and the method of implementing the position estimation is based on the zones shown in Figure 7. By marking the time of the onset of each bump, the current zone was known, and the current position was calculated using the following logic:

- In Zone 1, the slope of the line (velocity) is used to predict the current position,
- In Zone 2, a constant position is assumed,
- In Zone 3, as with Zone 1, the slope of the line (velocity) is used to predict current position,
- In Zone 4, equilibrium is assumed. Note: The small bump at the beginning of Zone 4 is a result of the way the bumps are applied to the system. Since the crank is allowed to turn in either direction, it rotates slightly in the reverse direction, causing another small bump. For simplicity, this bump is always neglected and therefore has a similar effect on fitness for all experiments.

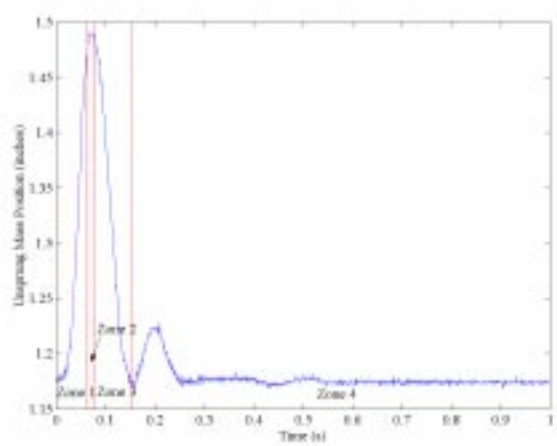


Figure 7: The four zones considered when implementing position approximation.

The position estimation proved to be useful for alleviating the problems caused by the potentiometer. However, this method would not have implemented well on the upper potentiometer (had it posed a problem). Because the input to the system is constant (varies only in frequency), its position is easy to predict with reasonable accuracy. However, once control is applied to the system, the variation in the position of the sprung mass would prove difficult to predict.

Disturbance Rejection

As previously mentioned, disturbance rejection is provided in the form of proportional P control or PID control. Proportional control (see Figure 8) is implemented in a characteristic way by applying a control proportional to the error. The PID control (see Figure 9) is also a characteristic implementation.

When utilizing only P control, the control applied to the disturbance rejection motor is a combination of two proportional gains (Figure 8). Two differing gains proved to be better than a single gain because pushing the motor downward (a negative displacement) is more difficult than raising it (a positive displacement). Utilizing a single gain would inhibit the performance of the controlled system for negative displacement. As seen in Figures 8 & 9, the control effort is negative for positive displacement and positive for negative displacement. This is due mainly to the wiring and mounting of the control motor.

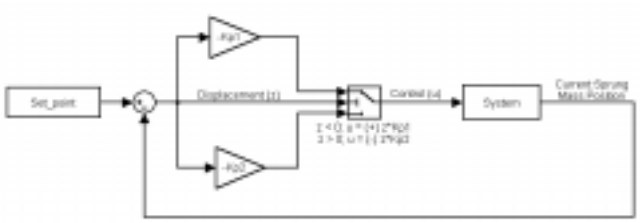


Figure 8: The block diagram of the system with proportional control. The sign on the control effort is due to the way the motor is wired.

The desired position (set point) could be specified as the position of the sprung mass when the controller is activated. This would allow the controller to maintain a position other than the natural equilibrium resting position and would prove useful in the event that weight is added or removed from the sprung mass. However, for the purposes of this experiment, the sprung mass remains constant, so the desired position is set by default to be the equilibrium position of the sprung mass.

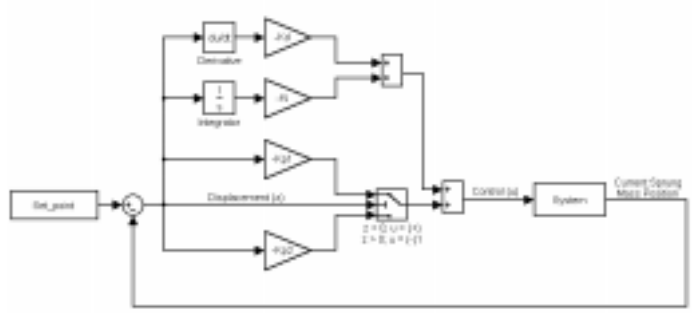


Figure 9: The block diagram of the system with PID control.

EXPERIMENTAL METHODOLOGY

After entering the desired inputs at the Visual Basic interface and selecting "Auto-Tune," the system will begin sending impulses to the system and adjust the controller gains until the tuning is complete. The details of the automatic tuning methodology are presented here.

The first frequency sweep (termed a "trial") is conducted with the provided initial gains to establish a baseline fitness (the fitness is the sum of the square of the displacement of the sprung mass over the entire trial duration). Kp_1 (the gain utilized for negative sprung mass displacement) is then incremented using the indicated step size, and the frequency sweep is conducted once again. The fitness of the current trial is then compared to the previous. There are three possibilities and courses of action at this point:

- If the fitness is within –5% or +2% of the previous value:
  - The value of Kp_1 is assumed to be optimal, and Kp_2 is tuned.
- If the fitness is better by more than 5% of the previous:
  - The gain is increased by the current step size.
- If the fitness is worse by more than 2% of the previous:
  - The step size is reduced to half its previous value.
- If the step size has been reduced 4 times, the search proceeds in the opposite direction using the default step size.
- If the search has been conducted in both directions, the current gain is assumed to be optimal and Kp_2 is tuned.

This method of automatic tuning is then applied to Kp_2 (the gain utilized for positive sprung mass displacement) and then to K_d and K_i if applicable. Because the gains are tuned independently of one another, it is necessary to repeat the process (termed a "run") until the gains

have settled to some optimal values. The collection of runs is termed an "experiment," where the outcome is a set of optimal controller gains.

RESULTS

Several experiments were conducted to determine the feasibility of automatically tuning the proportional gains and proportional, integral, and derivative gains. For purposes of comparison, each experiment was conducted utilizing equivalent bump magnitudes and frequency sweeps. Several experiments were conducted to determine if the gains converge to the same (optimal) values, and the gains are always initialized as zero. The gains are initialized as zero so that the first local minimum is attained. This is desirable because there is less cost involved (i.e., lower power requirements, etc.). However, if the increase in fitness is not substantial enough, then the search would need to be continued by choosing a larger step size or different initial conditions. This proves not to be the case however, as a considerable increase in fitness was obtained utilizing the automatic tuning methodology.

Since the potentiometers occasionally output erratic voltages, steps were taken so that this has a minimal effect on the system, however, after only a few runs, the problem would worsen enough that data became flawed. It was found that allowing the system to rest for a day would restore it to normal operation, so this course of action was taken when necessary.

Selected experimental results are presented here, including those that exhibit the behavior of the system when it receives erratic sensor readings. Note that each frequency sweep is termed a "trial," a number of trials is termed a "run," and a number of runs is termed an "experiment." Also note that numerous runs are essential because the gains are tuned individually, and it is necessary to determine if the gains are dependent upon one another.

TUNING OF PROPORTIONAL GAINS

The first set of experiments was the determination of the optimal values of the two proportional gains. Table 1 presents an example of one proportional gain run. It shows the progression of the gains from one trial to the next, as well as the respective change in fitness. It is seen that Kp_1 is adjusted by its step size until the fitness is not bettered by more than 5%. Kp_2 is then tuned until within 5% of the optimal fitness. It is also seen that the final fitness is reduced by a factor of one hundred as compared to the fitness of the initial (passive) trial.

Table 2 presents the compiled final gain values at the end of each run for three selected experiments (Experiments 1 and 2 are not considered here because they were conducted prior to implementation of the potentiometer position estimates).

Table 1. The progression of Controller Gains for P Tuning - Experiment #3 Run #1

Trial Number	Fitness	Kp1	Kp2
1	776.64	0	0
2	498.46	4	0
3	393.25	8	0
4	362.88	12	0
5	341.17	16	0
6	341.06	20	0
7	149.98	20	8
8	110.84	20	16
9	48.01	20	24
10	29.27	20	32
11	19.41	20	40
12	17.67	20	48
13	14.33	20	56
14	11.12	20	64
15	9.12	20	72
16	8.17	20	80
17	8.97	20	88
18	8.12	20	84

Intermediate gain values similar to those found in Table 1 are omitted for conciseness. The final optimal values of the proportional gains converge to similar values with the exception of Experiment 4. Potentiometer problems were apparent during Experiment 4, so the position estimation detailed in previous sections was relied upon and may explain the difference in gains.

Table 2. Final Controller Gains for Each Run of Selected P Tuning Experiments

Experiment #3				
Run Number	Number of Trials	Fitness	Kp_1	Kp_2
1	18	8.12	20	84
2	16	8.34	19	83
3	13	8.38	21	83
Experiment #4				
Run Number	Number of Trials	Fitness	Kp_1	Kp_2
1	14	15.38	20	64
2	10	16.04	24	63
Experiment #5				
Run Number	Number of Trials	Fitness	Kp_1	Kp_2
1	17	9.09	20	88
2	9	9.38	21	89

The fact that Kp1 and Kp2 converged to distinctly different values supports the usage of two proportional gains. It is also relevant to point out that both gains tune in the first run, insinuating that the proportional gains do not substantially affect one another.

Considering Experiments 3 and 5 only, the values of Kp_1 match exactly, Kp_2 is within 6.7%, and the fitnesses are within 10.7%. A comparison is not made with Experiment 4 since it was affected by potentiometer problems (Experiment 5 was not affected because the potentiometer was allowed ample time between experiments to restore normal operation).

Figures 10-13 present a graphical view of the results of the optimization. It is qualitatively seen that there is a substantial increase in fitness for all four frequencies presented in the figures.

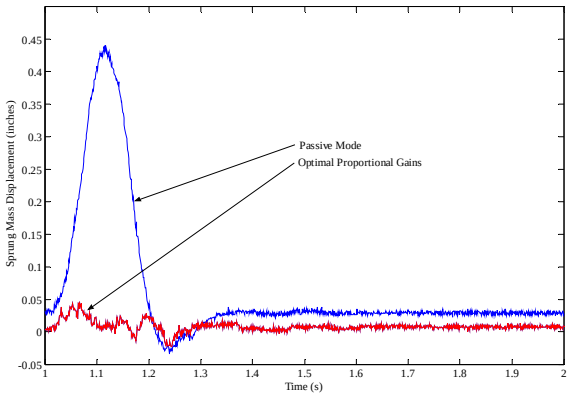


Figure 10: Sprung mass displacement at 1Hz for the passive and active (optimal P control) suspensions.

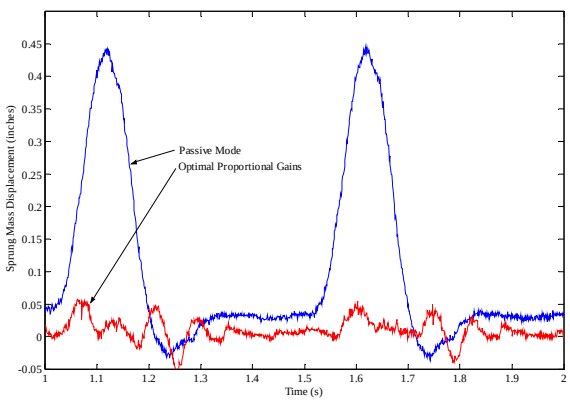


Figure 11: Sprung mass displacement at 2Hz for the passive and active (optimal P control) suspensions.

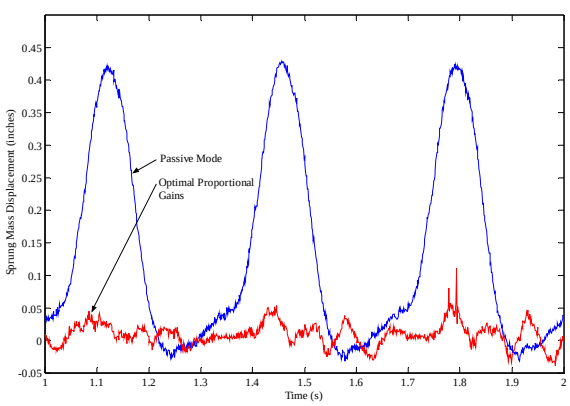


Figure 12: Sprung mass displacement at 3Hz for the passive and active (optimal P control) suspensions.

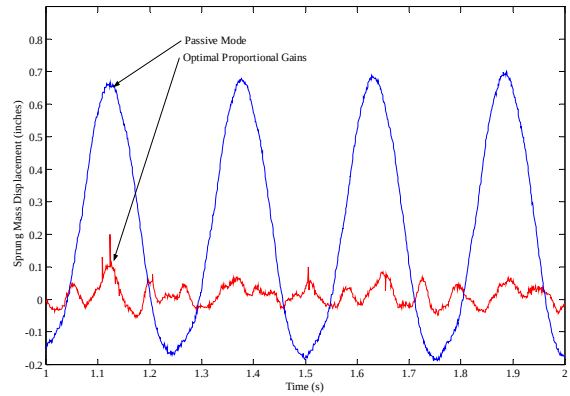


Figure 13: Sprung mass displacement at 4Hz for the passive and active (optimal P control) suspensions.

Figures 10-13 indicate that the sprung mass displacement is controlled well at all frequencies and that the magnitude of the displacement is highest at the largest frequency of 4 Hz, but the peak displacement is still reduced by approximately 90%.

TUNING OF PROPORTIONAL, INTEGRAL, AND DERIVATIVE GAINS

The second set of experiments was the determination of the optimal values of the two proportional gains in conjunction with the derivative and integral gains. Table 3 presents an example of a single PID tuning run. Similar to Table 1, it shows the change in the gains from one trial to the next, as well as the respective change in fitness. It is seen that each gain is adjusted by its step size until the fitness is not bettered by more than 5%, and then successive gains are tuned in a like manner. It is also seen that the fitness after one run is considerably better than the initial fitness of the initial (passive) trial, but not noticeably better in comparison with the fitness resulting from the optimal proportional gains. Note that in the first run, the proportional gains converge to the optimal gains that were obtained in the proportional-only experiments. This further supports the validity of the proportional gain optimization experiments.

Although the proportional gains converge similarly in the first run, integral and derivative terms are now present and could possibly affect the proportional gains. This turns out to be the case as seen by inspecting the final controller gains for the PID experiments presented in Table 4.

Table 3. The Progression of Controller Gains for PID Tuning - Experiment #1 Run #1					
Trial Number	Fitness	Kp_1	Kp_2	K_d	K_i
1	903.89	0	0	0	0
2	620.34	4	0	0	0
3	470.87	8	0	0	0
4	397.63	12	0	0	0
5	342.14	16	0	0	0
6	328.62	20	0	0	0
7	157.73	20	8	0	0
8	107.15	20	16	0	0
9	50.06	20	24	0	0
10	33.56	20	32	0	0
11	25.83	20	40	0	0
12	19.62	20	48	0	0
13	14.17	20	56	0	0
14	10.36	20	64	0	0
15	8.65	20	72	0	0
16	8.05	20	80	0	0
17	8.00	20	88	0	0
18	7.65	20	88	0.03	0
19	7.18	20	88	0.03	0.0030
20	11.86	20	88	0.03	0.0060
21	7.69	20	88	0.03	0.0045
22	6.42	20	88	0.03	0.0038
23	6.75	20	88	0.03	0.0045
24	6.78	20	88	0.03	0.0008
25	7.48	20	88	0.03	0.0023
26	6.61	20	88	0.03	0.0030

Table 4. Final Controller Gains for Each Run of Selected PID Tuning Experiments						
Experiment #1						
Run Number	Number of Trials	Fitness	Kp_1	Kp_2	K_d	K_i
1	26	6.61	20	88	0.0300	0.00300
2	16	5.34	28	96	0.0225	0.00220
3*	3	39.99	36	96	0.0225	0.00220
4	16	5.15	28	97	0.0375	0.00520
Experiment #2						
Run Number	Number of Trials	Fitness	Kp_1	Kp_2	K_d	K_i
1	24	15.78	24	25	0.0650	0.00200
2*	23	223.77	35	37	0.0850	0.00150
3*	10	52.37	35	45	0.0850	0.00200
4	31	9.15	35	61	0.0850	0.00150
5	27	7.57	35	60	0.0850	0.00150
Experiment #5						
Run Number	Number of Trials	Fitness	Kp_1	Kp_2	K_d	K_i
1	19	7.47	20	88	0.0300	0.00300
2	10	6.50	28	96	0.0375	0.00375

*The relatively large final fitness for each of these runs was observed to be a result of potentiometer problems.

Irregular potentiometer readings were once again present, as shown by the results of Experiment 2. The system was allowed to rest between trials three and four, and a drastic increase in Kp_2 is seen during the next run while the other gains remain relatively the same.

Although the gain values of Experiments 1 and 2 are distinctly different, the fitnesses differ only slightly. This indicates that there is more than one optimum, however, it is desirable to maintain the lower gains of Experiment 1 than the higher ones of Experiment 2 because lower gains translate to lower power consumption and therefore a savings in cost. The gains would not have settled at the local minimum of Experiment 2 barring potentiometer error (consistent with the results of the remaining experiments).

Figures 14-17 indicate that the sprung mass displacement is controlled well at all frequencies and that the magnitude of the displacement is highest at the largest frequency of 4 Hz. Even at 4 Hz, the peak displacement is reduced by approximately 90%. These results are very similar to those obtained from the proportional gain experiment and indicate that the addition of integral and derivative gains do not contribute to much of a performance increase.

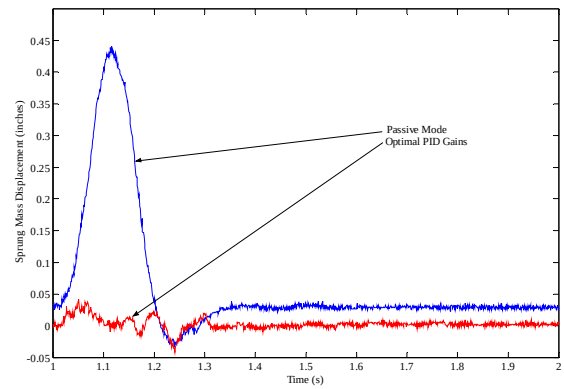


Figure 14: Sprung mass displacement at 1Hz for the passive and active (optimal PID control) suspensions.

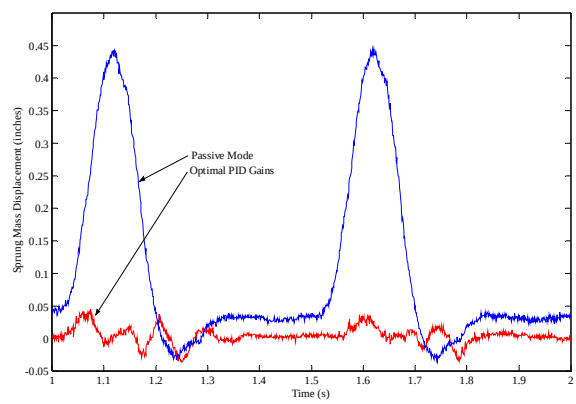


Figure 15: Sprung mass displacement at 2Hz for the passive and active (optimal PID control) suspensions.

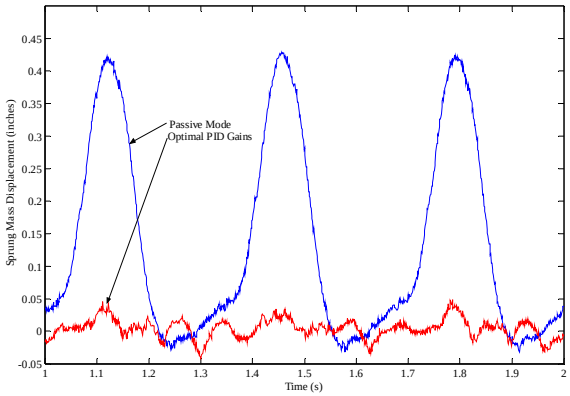


Figure 16: Sprung mass displacement at 2Hz for the passive and active (optimal PID control) suspensions.

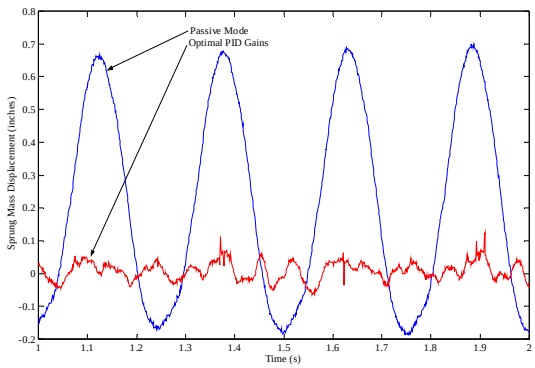


Figure 17: Sprung mass displacement at 4Hz for the passive and active (optimal PID control) suspensions.

Quantitatively, the optimal fitness obtained from the PID experiments (5.15) is approximately 60% better than the optimal fitness obtained from the proportional experiments (8.38). The results are not entirely repeatable, which accounts for some of the difference, but it may not account for the entire difference. Since the numbers are relatively small in magnitude, a small increase will translate to a large percentage. A turn is taken to the qualitative results, which are shown in Figure 18. Figure 18 shows an overlay of the sprung mass displacement for the optimal P and PID gains for 1, 2, 3, and 4 Hz. This assessment verifies that for the scenarios tested in the experiment there is not much benefit from the addition of the integral and derivative gains, as the plots nearly overlap for the 1, 2, and 3 Hz scenarios.

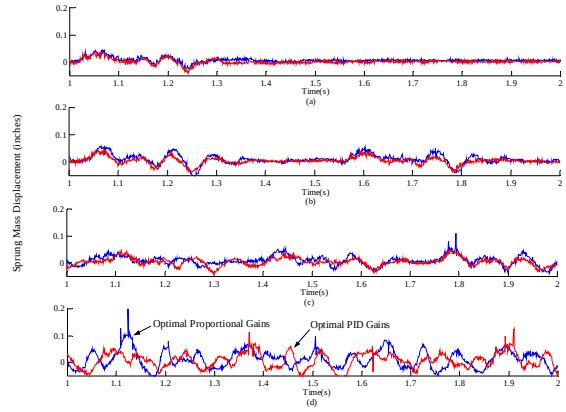


Figure 18: A comparison of the sprung mass displacement for the optimally tuned P and PID controllers at frequencies of 1, 2, 3, and 4Hz.

CONCLUSION

The results indicate that the automatic tuning of controller gains is a feasible means of obtaining and maintaining a better ride quality. This translates to an online, real-time system that is able to adapt to changes in the system. Since the objective of the system would be to minimize power consumption while maximizing ride comfort, minimum control effort is desirable. However, the results do indicate the presence of local minima, meaning that more effort is required before the system can be implemented effectively. Specifically, further investigation into the effects of the search step size and direction are required. By investigating the effects of the step sizes or by implementing an alternative control algorithm on the step sizes, the number of runs required to settle may even be shortened.

Accompanying an investigation into the behavior of the system around the local minima, an analysis on economic feasibility must also be conducted. The computer hardware required to manage the system may prove too complex and costly to implement, as consumers are reluctant to pay for what they perceive as a slight increase in performance. However, as technology advances, the economic impact may reduce dramatically, so this would require periodic analysis into the methodology.

Overall, the methodology was investigated successfully, and it proves to be useful with a substantial increase in fitness in both the proportional and PID cases. The proportional gains settle in a relatively short time and take slightly longer when integral and derivative control is used in conjunction with proportional control, and the system does not seem to benefit from the addition of the integral and derivative terms. However, only ride quality has been considered, and their inclusion may prove useful if other factors such as cornering, accelerating, or

braking performance are taken into account. Additional details of the procedures outlined in this paper can be found in reference [5].

REFERENCES

1. Genta, Giancarlo. Motor Vehicle Dynamics: Modeling and Simulation, 1997. *Series on Advances in Mathematics for Applied Sciences*, Vol. 43. World Scientific, River Edge, NJ.
2. Gillespie, Thomas D. Fundamentals of Vehicle Dynamics, 1992. Society of Automotive Engineers, Inc., Warrendale, PA.
3. Crolla, D. A. "Theoretical Comparisons of Various Active Suspension Systems in Terms of Performance and Power Requirements," IMechE Paper C420/88, 1988, pp. 1-9.
4. Sharp, R.S., and Crolla, D.A. "Road Vehicle Suspension System Design – A Review," *Vehicle Systems Dynamics*, Vol. 16, No. 3, 1987, pp. 167-192.
5. Clark, Brent A. "An Experimental Assessment of the Feasibility of the Online Tuning of Active Suspension Controller Gains." M.S. Thesis, Department of General Engineering, University of Illinois at Urbana-Champaign, May 2001.