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An Investigation of Digital Microprocessor Algorithms in Active Suspension Systems

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Abstract

The need for knowledge and understanding of these systems at the technician level will increase dramatically as other manufacturers develop production systems in the 90's. Current technical literature on the subject of active and semi-active suspension systems does not well explain a major aspect of these systems, the origin and nature of the control algorithms used. A review of 30 articles and technical papers, and a title search of over 300 others discovered only 2 vague presentations of the actual control logic used in such systems. Furthermore, the control information that was printed was presented in terms of Laplace transforms and transfer functions which are third or forth year electrical engineering topics (Miller, 11 November 1991).

The purpose of this paper was to discover the nature of the key control algorithms found in active and semi-active suspension systems and explain them in terms a technician can understand and utilize when investigating such systems.

AN INVESTIGATION OF

DIGITAL MICROPROCESSOR ALGORITHMS IN

ACTIVE SUSPENSION SYSTEMS

A Research Paper for Presented
to the Graduate Faculty
of the
Department of Industrial Technology
University of Northern Iowa

In Partial Fulfillment of the Requirements for

the Non-Thesis Master of Arts Degree

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CHAPTER I

INTRODUCTION

Background

Active suspension systems are new to production automobiles and off-road equipment. Active and semi-active systems improve operator ride and comfort by controlling suspension actuators with microprocessors using position and acceleration as inputs. While the only current production system was found on the 1991 Nissan Infiniti (Csere, 1991), the concept of active and semi-active suspensions was not new to the engineering literature. In the 1970's Thompson (1976) reported on "active suspension with optimal linear state feedback". As far back as the 1960's, Mitschke (1962) reported on the effects of vehicle dimensions and road conditions on body motion and wheel loads.

In the last decade however, interest in these systems has picked up considerably. Claar and Vogel (1989) found no fewer than 79 journal articles and technical papers written since 1980 on the subject. This interest has culminated in the introduction of the Nissan Infiniti system that sets the standard for automobile suspension systems in the 90's.

Need for and Purpose of the Study

The need for knowledge and understanding of these systems at the technician level will increase dramatically as other manufacturers develop production systems in the 90′s. Current technical literature on the subject of active and semi-active suspension systems does not well explain a major aspect of these systems, the origin and nature of the control algorithms used. A review of 30 articles and technical papers, and a title search of over 300 others discovered only 2 vague presentations of the actual control logic used in such systems. Furthermore, the control information that was printed was presented in terms of Laplace transforms and transfer functions which are third or forth year electrical engineering topics (Miller, 11 November 1991).

The purpose of this paper was to discover the nature of the key control algorithms found in active and semi-active suspension systems and explain them in terms a technician can understand and utilize when investigating such systems.

Limitations of the Study

The study was limited to the application of control theory in microprocessor algorithms for active and semi-active electrohydraulic suspension systems.

It concentrated on the processing of inputs to the microprocessor and did not consider control valve theory or specific outputs to the hydraulic system. Special attention was paid to the utilization of acceleration sensor signals and how and why they are useful.

An understanding of certain elements of control theory was necessary to understand active suspension theory. These elements were explained in the context of a simple vertical active system. However, a thorough explanation of control theory was beyond the scope of the paper.

Statement of Procedure

The nature of the key control algorithms found in active and semi-active suspension systems was explained and demonstrated in the following manner:

1. Review the historical development of semi-active and active suspension systems, including advances in theory, prototypes, and production.
2. Review the major aspects of control theory that impact active suspension systems. Explain the major concepts in a manner understandable to technicians.
3. Review the nature of active control system

algorithms in the context of the sky-hook damper concept and absolute velocity of sprung mass.

4. Develop and test by simulation an analog and a digital control algorithm usable with the sky-hook damper concept.

While the research done here was predominantly descriptive or observational, the last section did replicate some work done by previous researchers, and did develop a cause and effect relationship between inputs and outputs to a system.

Definition of Key Terms

The following terms were defined to clarify their use in the context of the study:

Active Suspension System:

A suspension system in which an actuator replaces or acts in parallel with a spring and damper. The actuator works according to a force demand signal generated from a microprocessor on the basis of measured information about acceleration and displacements of the vehicle body (Crolla, 1989).

Algorithm:

A formal procedure or a set of rules for solving a problem in a finite number of steps (Barnhart, 1986).

Controller:

The control module, sometimes built around a Motorola 6805 microprocessor, that receives input from the sensors and decides what output is required to the control valves to achieve the goals of the system (Soltis, 1987).

Passive Suspension System:

A suspension system consisting of spring and damper in which none of the system parameters such as damping factors or spring rates can be adjusted (Soltis, 1987).

Semi-active Suspension System:

A system with the ability to continuously vary the rate of energy dissipation of the system. This is usually done by controlling the orifice size of a hydraulic damper (Corolla, 1989).

CHAPTER II

HISTORICAL BACKGROUND

Literature on the subject of active and semi-active suspension systems consisted of 4 major levels of publication, magazine articles about new production applications, journal articles about prototype applications, technical papers and dissertations about the theory and application to new systems, and text books on pure control theory. A review of this literature showed the spiral of development. First, a technical paper or dissertation proposed the application of a new theory to the problem of rider comfort or vehicle performance. Then journal articles appear showed prototype applications of part of the theory. This was followed by magazine articles on new production applications of part of the theory. Once production models were introduced, enhancements followed the same cycle of theory, prototype, and production. In this manner, the state of the art progressed from passive to semi-active to full active suspension systems.

Theoretical Advances and Early Prototype Work

Mitschke (1962) wrote one of the first papers concerning suspension theory. He wrote about the influence of road and vehicle dimensions on vehicle body motion and wheel forces. Soehne (1965) wrote about vibrations in vehicles, including resonant frequencies as a function of vehicle mass and spring coefficients. Karnopp and Crosby (1974) wrote one of the first papers on semi-active suspension control of vibration. Thompson (1976) modeled an active suspension with optimal linear feedback. His working model, perhaps the first active system, used position sensing of the sprung mass relative to the unsprung mass to control a hydraulic actuator. While he did have one of the first working models, problems with transient motions of the unsprung mass made his model ineffective in practice. Thompson and Pierce (1979) did additional work modeling the road inputs to suspension systems. Krasnicki (1980) did a comparison of analytical and experimental results for a semi-active vibration isolator.

Later, Margolis (1982) did a theoretical study on the response of active and semi-active suspensions to realistic feedback signals. His basic model was a linear mechanical system of springs and dampers simulating the road, tire, spring, and shock absorbers of an automobile suspension system. Using transfer function analysis, he showed how road input accelerations can be sensed and used for suspension control. Figure 1 shows the main features of such a model.

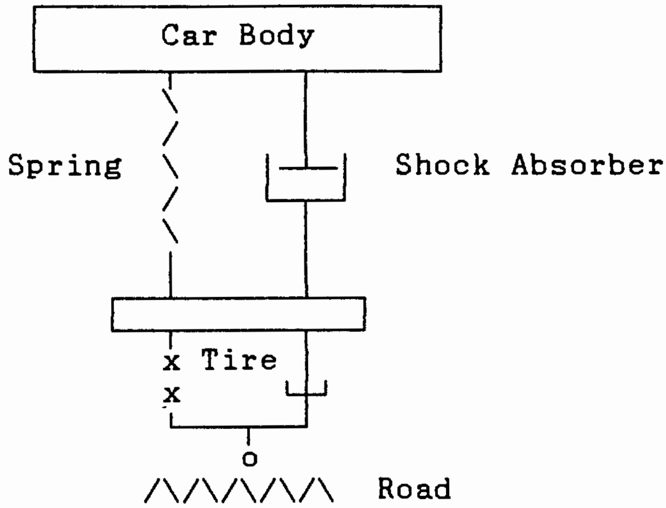


Figure 1. Simple linear mechanical model of road, tire, spring, mass, and damper (Margolis, 1982).

In his study, Margolis defined passive, semi-active and active systems. This study turned out to be a turning point for active systems since it proposed the feedback of the absolute velocity of the sprung mass to an actuator. The Margolis model used modern control theory including Laplace transforms and transfer functions to illustrate system frequency response. Margolis even proposed using the integral of the absolute acceleration of the sprung mass to calculate the absolute velocity.

Oppenheim and Schaeffer (1983) authored Digital Signal Processing, a control theory text book that explained most of the mathematical and physical theory used in control models. This book has many examples and was useful in understanding the various active system models researched.

Low Tech Production Models and High Tech Prototypes

The 1984 Lincoln Mark VII introduced an electronically controlled air suspension system (Berry, 1983). The system used position sensors on each lower control arm to feed information to a control module that activated valves and an air compressor to control the height of the vehicle under varying loads. The system algorithm had several safety features to insure safe operation. Since air was slow acting, and since the position sensors were unable to signal rough road information, the system was a load leveling system only. It was of little help in cornering or braking situations, nor could it compensate for bumpy conditions.

The same year, Mitsubishi Motors reported a near-production electronically controlled suspension system that changed both the spring rate and the damping coefficient depending on sensed conditions (Masaaki & Sunao, 1984). The system utilized sensors for vehicle speed, steering angle and rate of change of steering angle, throttle position, suspension stroke, and sprung mass acceleration. Air springs in parallel with coils and shocks with two position damper orifices were used to change the spring rate and damping coefficients. The control algorithm interpreted road conditions of high speed, acceleration, or cornering to set both springs and shocks to the "hard" position. Otherwise the system was set to "soft" for smooth or bumpy roads. The system was an improvement for cornering, but the slow acting air springs did not handle bumps in corners well.

Dominy of Rolls Royce Aero Division and Bulman of the Royal Military College of Science (1985) proposed a semi-acitve suspension model for Grand Prix racing cars. The model utilized full mathematical analysis of a hydraulic actuator, gas spring, variable control valve system. The most unique aspects of the model and system was the use of hydraulic components and the mathematical modeling of the complete system including fluid movements.

Karnopp (1986) further developed Margolis' model by illustrating the frequency response of a passive system compared to that of an active system. His model focused on the transfer function formed from the ratio of the input velocity to the output velocity.

McElroy (1986) reported on GM's Corvette prototype fully active system. This system was built for the Corvette by the newly acquired Lotus division. It was modeled after the system used on their Grand Prix racer. However, it was judged to consume too much energy and to be too costly to build for production. One of the major cost constraints was the use of five $800 servo valves for the actuators.

Soltis (1987) reported on the 1987 Ford Thunder Bird semi-active suspension system, the first American production car attempt at this technology. The system was not a true semi-active system in that it did not use chassis movement feedback to regulate the shock absorbers. Instead, it used steering wheel movements to infer cornering g-forces. No vertical sensing was employed. Nevertheless, the system did require a network of sensors and actuators controlled by a central processor. This qualified it as a semi-active suspension sysem from an actuator point of view.

<u>Off-road Prototypes and Production Full-active Systems</u>

Miller and Nobles (1988) investigated six variations of active and semi-active suspension systems for a military tank. The proposed control algorithm was a digital filter using a difference equation to calculate the velocity of the sprung mass from accelerometer readings. Lang, Kirk, McCormac, Wilson, and Wilson (1988) explored off-highway active seat suspensions. They explored a mechanical model of the human body and the corresponding resonance points. A table correlating off-road occupations with back injuries was presented. An active hydraulic seat was proposed using accelerometer and position sensor input. The associated algorithm was described as analog filtering followed by some type of microprocessor calculation. Yue, Butsuen, and Hedrick (1989) proposed three alternative control laws for active automobile suspension systems. The laws were full state feedback, absolute velocity feedback, and suspension deflection feedback. All three methods balanced the key performance measures of suspension travel, ride quality, and road holding ability. All three methods used the concept of the sky-hook damper which will be explored below. The authors concluded that absolute velocity feedback had the advantages of the others without high frequency harshness. Akatsu, Fukoshima, Takahashi, Satoh, and Kawarazaki (1990), all of Nissan Motor, described their active suspension system that used pressure control valves for the first time.

Previous to this time, all such active systems used servo valves, a much more expensive alternative. The paper concentrated on the hydraulic aspects of the system with only a reference to a sky-hook damper to infer the algorithm in the microprocessor. Aoyama, Kawabata, Hasegawa, Kobari, Sato, and Tsuruta (1990) further described the Nissan active suspension system of Akatsu et al. This paper demonstrated graphically the improvements in performance and explained the electronic and hydraulic system layout. The algorithm used in the controller was described as integrated accelerometer signals fed back to the hydraulic actuator using sky-hook damper theory for vertical attenuation. Figure 2 shows an abbreviated block diagram of the acceleration control.

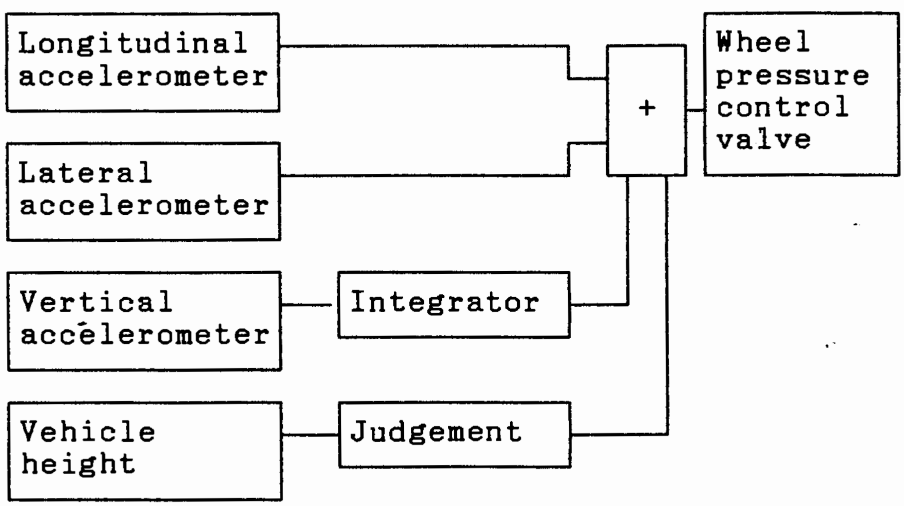


Figure 2. Block diagram Nissan control logic (Aoyama et al, 1990).

Lateral and longitudinal accelerometer signals were joined with the integrated vertical signals in summing junctions at each wheel. A current was sent to the individual control valves proportional to the summed value of the acceleration signals. A judgement circuit monitored chassis height using position sensors to maintain the proper height of the vehicle.

The Nissan breakthrough was proven real by their introduction of the 1991 Nissan Infiniti Q45 with an active suspension option. The performance improvements claimed by Aoyama et al. were confirmed by Csere (1991) in a November 91 Car and Driver article. Csere's tests compared a Q45 with active control to one without active control, using level sensors on all four wheels. Results showed considerable improvement in body roll, pitch, single bump, and rough road performance with the active system. Table 1 shows the percentage improvement of the active system over the passive system.

Table 1.
Percent Improvement Active vs. Passive.
From Csere (1991).

TEST	METRIC	PERCENT
Slalom	Degrees	1%
Bump	Inches	50%
Roll	Degrees	50%
Pitch	Degrees	400%
Rough Road	Inches	66%

The active suspension option was expensive at $5200 cost to the buyer. However, the author estimated 11% of buyers have made the purchase.

It was obvious that other auto manufacturers must offer this type of suspension in the near future to stay close on the Nissan trail. In addition, the off-road needs for these applications has been shown and as soon as the technology is readily available, systems will appear in production.

CHAPTER III

PHYSICS MODELS AND TRANSFER FUNCTIONS

The study of active suspensions and their control algorithms required some knowledge of control theory. The study of control theory required an understanding of the physics of linear models and the concept of transforms and transfer functions.

Physical Principles

Control theory started with the development of simple linear models. Linear models were simplified mechanical equivalents of physical systems such as suspension systems. Figure 3 shows a simple mechanical model of a spring, a mass, and a damper.

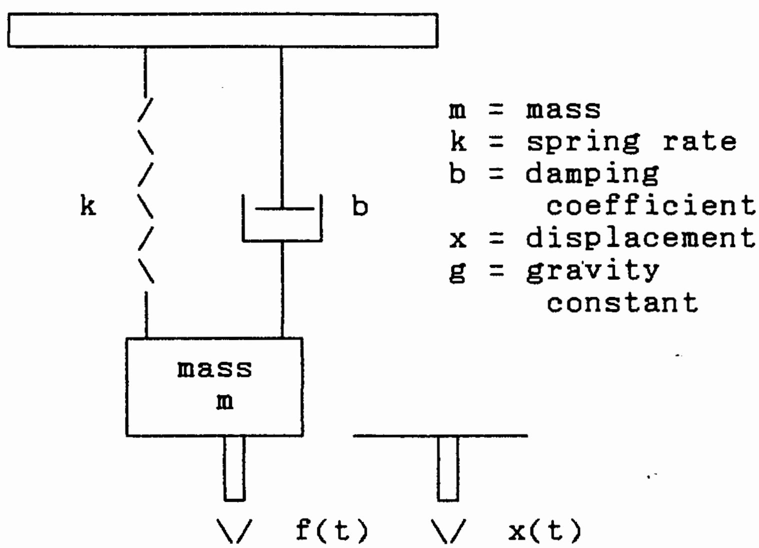


Figure 3. Simple Linear Mechanical Model of Spring, Mass, and Damper (Margolis, 1982).

The mass may be acted upon by an outside force. The relationships between the forces generated by the system and their variables was all linear, hence a "linear model".

The dynamics of the system was described with a series of force equations:

1.1. $f_m = ma = m \frac{d^2x}{dt^2} = m\ddot{x}$ where a, the acceleration of m is equal to the second derivative of x, the position of m.

1.2. $f_d = bv_m = b \frac{dx}{dt} = b\dot{x}$ where v, the velocity of m is equal to the first derivative of x.

1.3. $f_s = kx$

1.4. $f_g = mg = 0$ when balanced by the spring force.

Combining these equations using summation of forces yielded the state equations of the system in the time domain (differential equations).

1.5. $f_m + f_d + f_s = mg + f(t)$ ==> (for $mg = 0$)

1.6. $mx + bx + kx = f(t)$ in terms of x and its derivatives.

The Laplace Transform

Equation 1.6 fully described the system at any time t, as it moved in space and time. However, any oscillating system like this can be viewed more completely in the frequency domain. Because the system was dependent on the constants that describe it, m, d, k, and g, and on the magnitude and frequency of the input force f(t), different input force frequencies resulted in different system behavior. These different behaviors can be viewed in one picture by plotting ratios of magnitudes vs. input frequency.

The transfer of an equation from the time domain to the frequency domain was accomplished by applying the Laplace transform to the terms of the equation (Oppenheim & Schaeffer, 1983, pp. 7-21). The Laplace transform F(s) of a function f(t) was defined by:

$$1.7. \quad F(s) = L[f(t)] = \int_0^\infty f(t)e^{-st}dt$$

Laplace transform theory was extensive, and the proofs of its theorems were somewhat involved. However, the theory was well developed and formula tables existed to transform most common time domain terms to the frequency domain. The two main theorems used here were:

> Any sum of time domain terms can be transformed into the sum of their corresponding Laplace transform terms.
>
> ie, L[A + B] transforms into L[A] + L[B].
>
> The Laplace transform of the nth derivative of a time domain variable is s^n times the Laplace transform of the time variable.
>
> ie, $\frac{d^n f(t)}{dt^n}$ transforms into $s^n * L[f(t)] = s^n F(s)$
>
> See Oppenheim & Schaeffer (1983) p. 13.

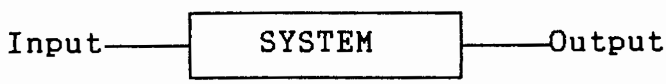
Applying the theorems to equation 1.6, we get:

$$1.8 \quad L[mx + bx + kx] = ms^2 + bs + k$$

To use the Laplace transform, we must think of the spring, mass, damper system in terms of an input f(t) and an output x(t). That is, an input force results in an output displacement. This idea is formalized in the concept of a transfer function.

<u>Transfer Functions</u>

Figure 4 is called a block diagram. It illustrated the concept of an input to a system being operated upon by the physics of the system and producing an output.



<u>Figure 4</u>. Block diagram of input to a system yielding an output (Oppenheim & Schaeffer, 1983).

The transfer function of a system was defined as the ratio of the Laplace transforms of its output and input (Oppenheim & Schaeffer, 1983, p. 12). Stated another way, the input of a system in frequency domain form multiplied by the transfer function of the system equaled the output of the system in frequency domain form. Then for our system, if f(t) was the input and x(t) was the output, and L[f(t)] = F(s) and L[x(t)] = X(s), we have using equation 1.7:

1.8. $X(s)/F(s) = 1/(ms^2 + bs +)k ==>$

1.9 $X(s) = F(s)*1/(ms^2 + bs + k)$

So, for any input force in frequency domain format, we can calculate the output displacement in frequency domain format, thus seeing how the system reacts to dynamic input oscillating forces.

This concept was typically put to use in "Bode plots" (Oppenheim & Schaeffer, 1982, p. 205). Figure 5 illustrates the use of the Bode plot. It is a plot of the frequency response of a seat suspension from .1 to 10 radians/second or .016 to 1.6 Hz or cycles/second. The transfer function was defined in this case in terms of the seat velocity divided by the input force velocity.

Above the 0 line was amplification of the input, and below the line was attenuation. Notice that this suspension did not begin to attenuate until after about 1.5 radians/second input, or about .24 Hz (14 cycles per minute). After that frequency the seat attenuated very well as shown by the steep slope of the curve.

Notice also the resonant peak at about 1 radian/second. This peak was a function of the natural frequency of the system and was a negative feature of all passive suspension system. Natural frequencies were a function of sprung mass, spring rate, and damping coefficient.

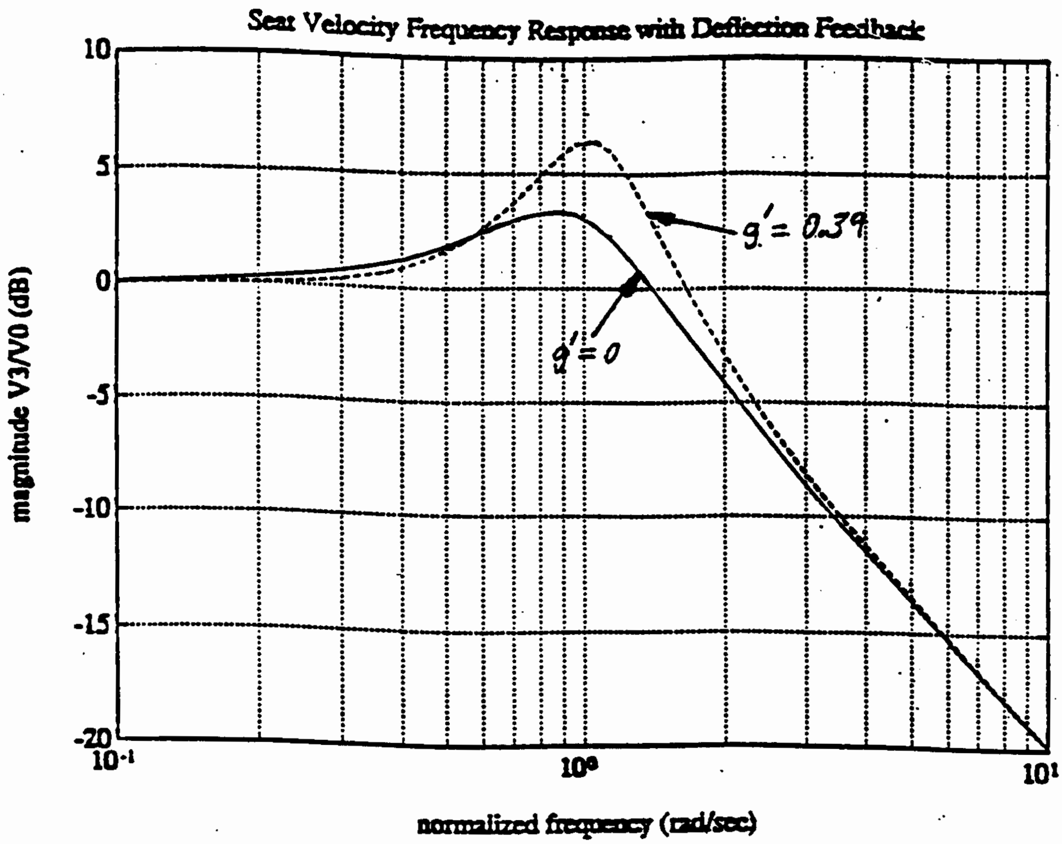


Figure 5. Bode plot of ratio of seat velocity to input force velocity vs. input force frequency. See Miller (1991, 6 September). Printed with permission of B. Miller of JDPEC.

Summary

Suspension systems weremodeled in terms of linear mechanical systems using basic physical laws in differential equation format.

Transfer functions were derived using the Laplace transform. These transfer functions were used to calculate the expected output of a system resulting from any input to the system.

Bode plots utilized transfer functions to plot the behavior (frequency response) of a system across all possible input frequencies. The Bode plot showd in one picture how the system reacted to all the possible frequencies of input force. The most interesting things shown in Bode plots were the resonant peaks of a system and the amount of attenuation at various input frequencies.

The main challenge of active suspension systems was to eliminate the resonant peaks and attenuate input (below the 0 line) through out the frequency range.

CHAPTER IV

SKY-HOOK DAMPERS AND ACTIVE SUSPENSIONS

Skyhook Dampers

We now know how to measure the performance of a suspension system by looking at its Bode plot. But how was it possible to remove the resonant peak and attenuate throughout the frequency range?

Two keys to this puzzle were discovered more than a decade ago. The first key was that if one could attach the sprung mass to the inertial reference system, instead of between components that move relative to each other, one would have a system without resonant peaks and with attenuation through out the frequency range. See for example Margolis, 1982, pp 269-275. Margolis called such a damper a "sky hook" damper.

But how does one go about attaching a damper to the "sky"? To do so required "sky" information, that is, information about the sprung mass relative to the inertial system, not relative to some point on the vehicle or the ground. But, such information was available from an accelerometer attached to the sprung mass. This was true because the seismic mass inside an accelerometer was displaced an amount proportional to the absolute acceleration (relative to the inertial system). The voltage of the output signal was proportional to the displacement of the seismic mass, and was therefore "sky hook" information.

This lead us to the second piece of key information. If the suspension system shock absorber was replaced by a hydraulic actuator and the actuator produced a force opposite in direction but proportional to the absolute velocity of the sprung mass, a sky-hook damper was achieved (Margolis, 1982, p. 270). This fact was expressed in the following equation:

2.1 $f(s) = k * v_m$ where k is some constant gain.

<u>Active Suspensions</u>

Fully active systems combined the two keys described above. By using the accelerometer to attach the suspension system to the "sky", the active system had information about the motion of the sprung mass. This information was analyzed by the system and used to alter that motion hydraulically if the analysis said it should be. The result was completely attenuated input forces at all frequencies using the integral of the absolute acceleration of the sprung mass, which is equal to the absolute velocity of the sprung mass, to calculate the counter forces. Figure 6 shows a skyhook active system with accelerometer feedback to a hydraulic actuator.

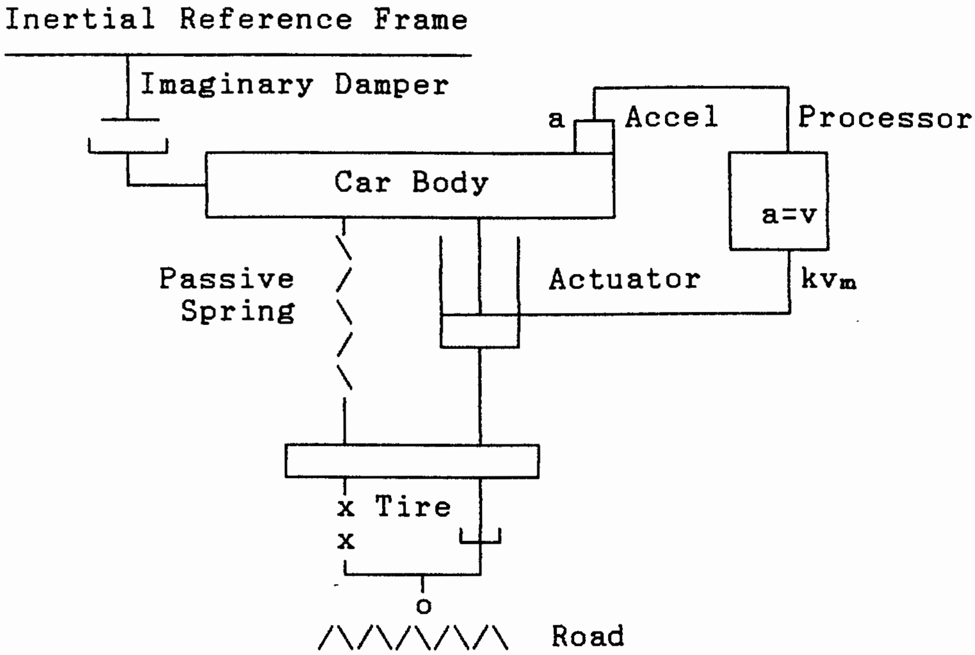


Figure 6. Skyhook damper with integrated accelerometer signal and actuator forcing proportional to sprung mass velocity (Margolis, 1982).

Summary

Complete attenuation of input forces in a suspension system was achieved by a sky-hook damper. Such a damper was achieved by finding the absolute velocity of the sprung mass and providing a force in a hydraulic actuator opposite in direction but proportional to that sprung mass velocity. Figure 7 is a bode plot showing the frequency responses of a passive seat suspension compared to that of an active seat suspension. Notice that the active suspension attenuated throughout the frequency range.

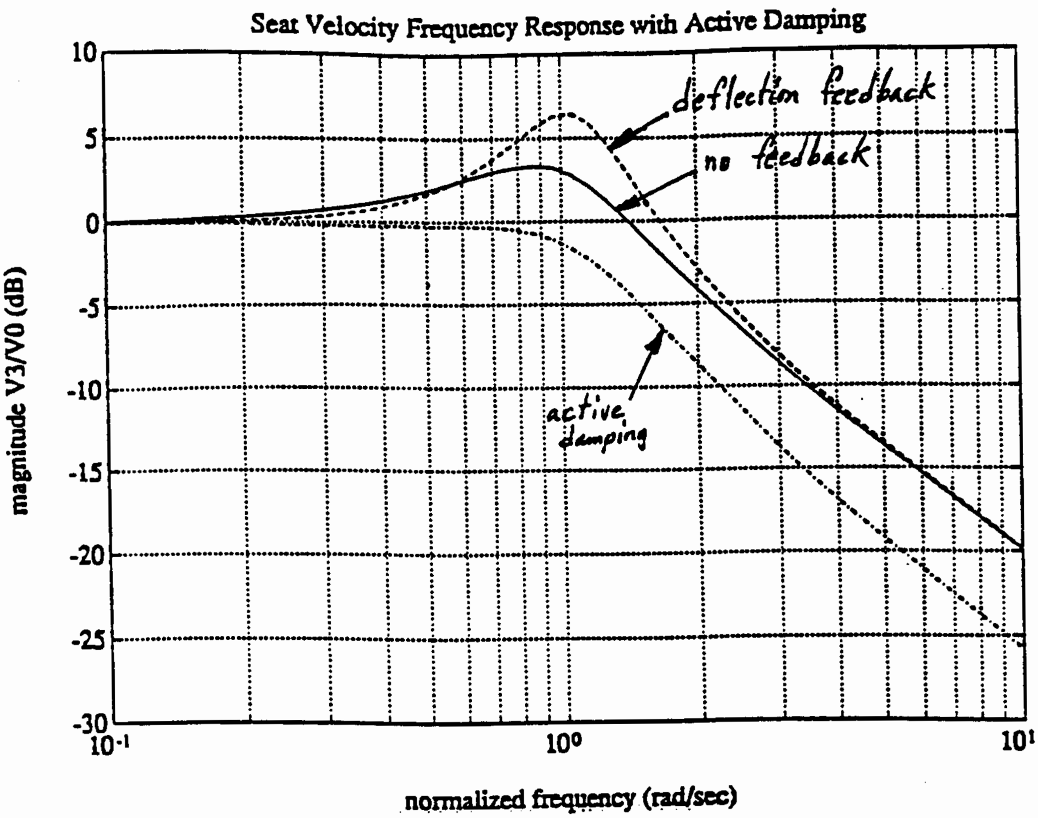


Figure 7. Bode plot of active vs. passive seat suspensions. See Miller (1991, 6 September). Printed with permission of B. Miller of JDPEC.

The active suspension system attenuates throughout the frequency range, is everywhere better than the passives system, and has no resonant peak. The absolute velocity was determined by integrating the signal of an accelerometer attached to the sprung mass. The accelerometer signal was received by the microprocessor and integrated to calculate the absolute velocity of the sprung mass. The microprocessor then sent a signal to the hydraulic valve that was proportional to the calculated velocity. The proportional hydraulic valve then increased the flow to the actuator producing a pressure proportional to the absolute velocity. Since pressure in the actuator multiplied by the area of the actuator piston equaled force, the force was proportional to the absolute velocity and the sky-hook was achieved.

CHAPTER V

ACTIVE SUSPENSION CONTROL ALGORITHM

The final point to be considered in this paper was the nature of the control algorithm used in the digital controllers of active suspension systems. The most interesting and unexplained portion of those algorithms was the integration of the vertical accelerometer signal to obtain the absolute vertical velocity. The melding of other signals in the overall control of the system was more straightforward and was explored in the literature in some detail (Aoyama et al., 1990). The vertical integration logic was common to all such systems but was obscure in the literature and was the portion discussed below.

Analog Integration

The integration of accelerometer signals can be done by a low pass analog filter. One analog integrator that works has a transfer function of the form 1/1+Ts where s is the frequency of the input and T is the break frequency at which the output begins to follow the output linearly (Oppenheim & Schaeffer, 1983, pp. 200-205). The success of a any filter as an integrator can be tested by inputting a sine(t) signal and testing the output for a -cosine(t) signal (since the integral of the sine is minus the cosine). Figure 8 is a computer simulation plot showing the sine wave input, the theoretical perfect -cosine output and the actual output for the filter above. Notice that after about 2 full cycles, the filter is almost 100% efficient.

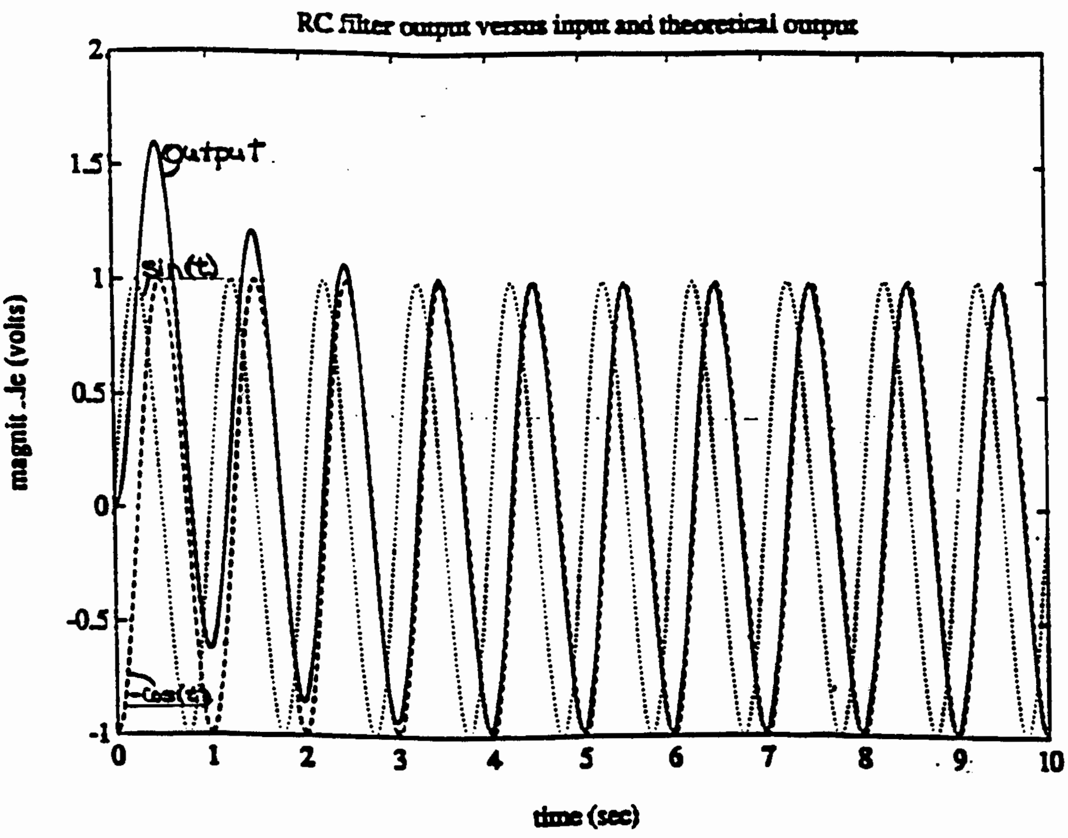


Figure 8. Analog integrator with theoretical vs. actual output. See Miller (1991, 6 September). Printed with permission of B. Miller of JDPEC.

Digital Integration

While analog integrators work in theory, they do not work well when they must be summed across a junction of several inputs. Nissan for instance had six accelerometer signals to sum at a given junction. They used a digital filter in the microprocessor (Aoyama, et al. 1990, pp. 83). While considerable higher level logic was described for their processor, no details of the integrator were given. However, Oppenheim & Schaeffer (1983, p. 243) suggested the optimal digital algorithm utilizes the trapezoidal rule. The equation is as follows:

3.1 $$Y_n = Y_{n-1} + T/2 * (X_{n-1} + X_n)$$

where: Y_n is the output voltage to be multiplied by a gain and sent to the hydraulic valve, Y_{n-1} is the previous calculated value, X_n and X_{n-1} are the current and previous accelerometer readings, and T is the processor sampling interval.

The control loop then worked as follows. Sample the accelerometer signal by going to the digital register that received the signal from the A/D converter board. Add this digital value to the value in the register holding the previous accelerometer value. Multiply the results by one half the time between sampling. This interval is a constant function of the processor. Add this result to the digital value of the previous output. Multiply the final result by a proportional gain and send the signal to the proportional hydraulic valve. Move the current output value to the previous output value register, move the current accelerometer reading to the previous input value register. Repeat the cycle.

The integration equation was demonstrated and tested by building a spread sheet and graphing the appropriate columns (similar to the method used on the analog algorithm). Appendix A showed a part of the spread sheet used in the demonstration. Figure 9 is a graph from the spreadsheet demonstrating the effectiveness of equation 3.1.

Notice that the curve labeled "digital integrator" was in proper phase with the "theoretical output" (-cosine), but the amplitude was not correct. Equation 3.1 was adjusted to better fit the ideal output. The curve labeled "adjusted digital integrator" fit the theoretical curve very closely after the first iteration. This is the type of customizing that is necessary to properly use this algorithm. Equation 3.2 was the adjusted equation for the improved digital integrator.

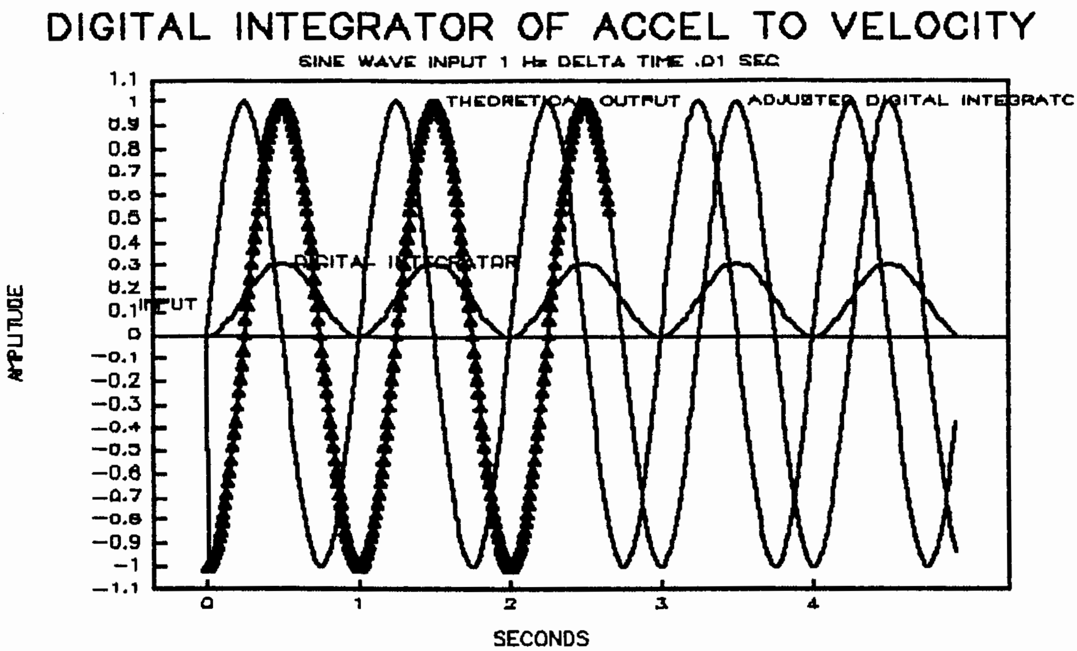


Figure 9. Digital integrator test of sine wave input vs. -cosine ideal output. Figure derived from application of equation 3.1 using Lotus Graphics.

An improved digital integrator is the following equation that employs the input frequency of the road.

3.2 $Y_n = Y_{n-1} + 2*Pi*f*T/2 * (X_{n-1} + X_n)$

where: Y_n is the output voltage to be multiplied by a gain and sent to the hydraulic valve, Y_{n-1} is the previous calculated value, X_n and X_{n-1} are the current and previous accelerometer readings, f is the frequency of the input signal and T is the processor sampling interval.

Notice that the improved version employed the factor "f" that was the input frequency of the sine wave. As long as this frequency was known, the figure showed that the integrator was near perfect. In real life situations, the input frequency may be difficult to determine. One strategy was to infer the frequency from previous input frequencies or develop a compromise value based on the characteristics of the vehicle or the terrain being traveled. This value was most likely a function of the particular vehicle's natural resonant frequency under specific conditions and was adjusted accordingly. For example, for a John Deere Large Row Crop tractor traveling over corn rows, the natural frequency was 2 Hz. This was the compromise "f" value to use for an active seat suspension in this application (Miller, 1991). Other possible methods were the use of ground sensors such as radar or front axle accelerometers to detect the input frequency before it affects the rear axle.

<u>Summary</u>

Accelerometer signals from the sprung mass was integrated by analog or digital means to yield the absolute velocity of the sprung mass. Effective digital filters were developed using the trapezoidal rule. These filters were tuned for the individual application by factoring in the natural frequency of the vehicle.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

Researchers have been working on the concept of active suspension systems since 1962. The decade of the 70's produced some theoretical advances and semi-active prototypes. The 80's completed the theory and produced some production semi-active systems and several fully-active prototypes. The 90's started off with a fully-active production system on the Nissan Infiniti Q45.

Linear mechanical models can be very useful in defining all types of suspension systems. Differential equations derived from system forces define the dynamics of these systems. Transformation of the time based differential equations into the frequency domain was accomplished by the Laplace transform. Transfer functions derived from control theory and application of the Laplace transform were used to predict the performance of systems throughout the frequency range of input forces. Bode plots are plots of transfer function output vs. input frequency. Such plots show the complete performance range of a suspension system in one picture. Bode plots of passive suspension systems show a frequency range of amplification rather than attenuation and a peak at the resonant frequency of the system. Improved systems show attenuation through out the frequency range and no resonant peak.

It has been shown that a theoretical system called a sky-hook damper would provide such performance characteristics. Sky-hook damping is accomplished by damping movement relative to the inertial reference frame, not relative to other system components. A sky-hook damper can be realized by integrating signals from an accelerometer attached to the sprung mass of the system. The integrated accelerometer signal is the absolute velocity of the sprung mass. A force in an attached hydraulic actuator opposite in direction but proportional to the absolute velocity of the sprung mass will produce attenuation through out the frequency range of input forces. Such an arrangement has been successfully utilized in several prototype and one production active suspension system.

The method of integrating the acceleration to achieve the absolute velocity was not well publicized. Analog integration with a low pass filter can be demonstrated. Modern systems use digital integration algorithms. Successful integration can be demonstrated with the following formula derived from the trapezoidal rule:

3.1 $Y_n = Y_{n-1} + T/2 * (X_{n-1} + X_n)$

where: Y_n is the output voltage to be multiplied by a gain and sent to the hydraulic valve, Y_{n-1} is the previous calculated value, X_n an X_{n-1} are the current and previous accelerometer readings, and T is the processor sampling interval.

Improvements to this algorithm can be realized by adjusting the amplitude and shifting the y-axis zero point. One improvement method relies on knowing the input force frequency. Since this frequency is constantly changing, individual system algorithms can be optimized by tuning the algorithm to the natural frequency of the vehicle.

Recommendations

Active systems have been proven possible and are now appearing on production automobiles. These systems will proliferate as costs are reduced. Technicians capable of maintaining these systems will be in ever increasing demand. The ability to diagnose problems requires understanding of the concepts involved. Technicians should be trained in the concepts of basic control theory, inertial reference frames, and digital integrator algorithms.

Technicians should not be intimidated by the apparent complexity of such systems since the principles involved are relatively few. A sufficient understanding of the concepts does not require the ability to prove how they work. Technicians need only have a feel for what a sky-hook damper does, and the knowledge that the force in the actuator is opposite but proportional to the sprung mass velocity.

Technicians should become increasingly aware of control algorithms in other computer controlled systems such as fuel injection systems. This knowledge can be used for perspective in understanding the key integration algorithm in active suspension systems.

Digital integration algorithms using the trapezoidal rule can be improved by tuning the equation to the particular application. System developers should become more aware of what factors affect such tuning.

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APPENDIX A

Spread Sheet of Digital Integrator

APPENDIX A

Spread Sheet of Digital Integrator

DIGITAL INTEGRATOR SIMULATION (9 DEC 91)

SAMPLING TIME (SECONDS) = 0.001
INPUT FREQUENCY (Hz) = 10

TIME t (SEC)	THETA RADIANS	SINE THETA	-COS THETA	DIGITAL FILTER	ADJUSTD DIGITAL FILTER
0	1	0.841470	-0.54030	0.000420	0
0.001	0.062831	0.062790	-0.99802	0.003261	-0.94515
0.002	0.125663	0.125333	-0.99211	0.003852	-0.78916
0.003	0.188495	0.187381	-0.98228	0.004834	-0.74811
0.004	0.251327	0.248689	-0.96858	0.006204	-0.68250
0.005	0.314159	0.309016	-0.95105	0.007957	-0.59261
0.006	0.376991	0.368124	-0.92977	0.010084	-0.47877
0.007	0.439822	0.425779	-0.90482	0.012578	-0.34144
0.008	0.502654	0.481753	-0.87630	0.015429	-0.18116
0.009	0.565486	0.535826	-0.84432	0.018626	0.001435
0.01	0.628318	0.587785	-0.80901	0.022156	0.205629
0.011	0.691150	0.637423	-0.77051	0.026005	0.430613
0.012	0.753982	0.684547	-0.72896	0.030158	0.675499
0.013	0.816814	0.728968	-0.68454	0.034599	0.939322
0.014	0.879645	0.770513	-0.63742	0.039309	1.221039
0.015	0.942477	0.809016	-0.58778	0.044272	1.519540
0.016	1.005309	0.844327	-0.53582	0.049466	1.833646
0.017	1.068141	0.876306	-0.48175	0.054871	2.162117
0.018	1.130973	0.904827	-0.42577	0.060467	2.503657
0.019	1.193805	0.929776	-0.36812	0.066231	2.856919
0.02	1.256637	0.951056	-0.30901	0.072139	3.220507
0.021	1.319468	0.968583	-0.24868	0.078170	3.592988
0.022	1.382300	0.982287	-0.18738	0.084299	3.972891
0.023	1.445132	0.992114	-0.12533	0.090502	4.358716
0.024	1.507964	0.998026	-0.06279	0.096754	4.748942
0.025	1.570796	1	1.0E-15	0.103031	5.142028
0.026	1.633628	0.998026	0.062790	0.109308	5.536423
0.027	1.696460	0.992114	0.125333	0.115560	5.930570
0.028	1.759291	0.982287	0.187381	0.121763	6.322913
0.029	1.822123	0.968583	0.248689	0.127892	6.711905
0.03	1.884955	0.951056	0.309016	0.133922	7.096011
0.031	1.947787	0.929776	0.368124	0.139831	7.473713