Computer Optimization Of Heavy Truck Suspension Parameters

Douglas W. Blue
Ford Motor Company (previously of The Pennsylvania State University), USA

Bohdan T. Kulakowski
The Pennsylvania State University, USA

ABSTRACT

A method for optimization of heavy truck suspension parameters considering overall vehicle performance is presented and demonstrated. Four performance measures are used to evaluate overall performance: pavement damage, ride comfort, stopping distance, and handling. The method uses numerical optimization methods and a complex heavy truck simulation program which permits consideration of overall vehicle performance. A sensitivity analysis was performed to help choose design variables, and twelve optimization cases were performed to demonstrate the method. The paper shows that a complex heavy truck simulation model can be used for suspension optimization, and that significant improvements in each performance measure can be obtained for a typical tractor semitrailer. Also, a method for implementing roll and yaw stability performance measures as constraints in an optimization problem is demonstrated.

INTRODUCTION

The goal of the designer of a heavy truck suspension system is to obtain a truck that is durable, economical, and provides reasonable comfort and handling properties. Trucks are continually growing larger, more powerful, and more plentiful. These factors, along with a growing awareness of highway safety and economics in recent years are placing more demands on designers. Trucks are expected to continue to become safer, more comfortable and less damaging to roads, and the manufacturers who can achieve these goals in the most economical manner will be the most successful. Computer aided design methods can be used to make the design process more efficient, and have become essential tools for manufacturers in competitive industries. This paper demonstrates the application of one of the computer aided design methods, numerical optimization, to the design of heavy truck suspension systems. A procedure was developed for optimizing the parameters of typical passive suspensions used on heavy duty articulated trucks to obtain improved overall truck performance. A method of this type can result in more efficient selection of the parameters of typical passive suspensions which are commonly used on heavy trucks. The procedure was then demonstrated for a typical tractor-semitrailer truck.

BACKGROUND

Various methods for optimizing road vehicle suspension systems have been used throughout history, the early methods including trial and error and the use of small analytical models. Computer based suspension optimization studies first appeared in 1967, one of the first being by Bender et al. [1]. Several researchers have performed studies since that time, primarily with small quarter or half car models, but the only application of numerical optimal design techniques to articulated vehicle suspension design found in the literature is given by El-Madany [2]. El-Madany optimized suspension damping for a six degree-of-freedom linear tractor semitrailer model with vertical and pitch degrees of freedom. No studies are found relating to the use of optimal design methods for articulated vehicle suspension optimization for overall performance.

Performance Measures

The objective of this study is to demonstrate suspension optimization while considering overall vehicle performance. Therefore, the optimization criteria must provide quantitative measures of overall vehicle performance. While every detail of vehicle performance was not addressed, four factors which broadly cover overall performance as related to suspension design were selected. The aspects of vehicle performance which are considered are: pavement loading, ride comfort, braking efficiency, and handling stability. The first three measures are implemented as objective functions which are to be minimized, while the handling stability measures are implemented as constraints. Two measures of pavement damage and two measures of ride comfort are used, resulting in five objective functions.
1) Pavement damage assuming random peak load location
2) Pavement damage assuming spatial repeatability of peak loads

3) Ride comfort as estimated by absorbed power theory
4) Ride comfort as estimated by RMS vertical floorboard acceleration
5) Stopping distance

The objective functions are used individually in order to learn more about each measure and to keep computing time reasonable. The objective functions could be combined with weighting factors and used simultaneously, but this would require a lot of computer time and was not done for this study. Handling stability includes both roll and yaw stability. The measures used as constraints in this study include the rollover threshold and understeer coefficient. Limits placed on these measures must be met by a design for the design to be considered feasible. In addition to the handling constraints, rattlespace constraints and design variable limits were used. The relative displacement envelope between the body and the axle is referred to as the rattlespace, so a rattlespace constraint is a limitation on the size of this envelope. This is to ensure that the design results in reasonable suspension deflections. Limits were also placed on each design variable to ensure reasonable values. The two pavement loading measures are based on the fourth power pavement damage law [3] and the dynamic load coefficient (DLC), which is the standard deviation of the dynamic load divided by the mean dynamic load.

$$DLC = \frac{\sqrt{\sum\limits_{i=1}^{N} (F_i - \bar{F})^2}}{\bar{F}}$$

where

$$\bar{F} = \frac{\sum\limits_{i=1}^{N} F_i}{N}$$

If random peak load distribution is assumed, then an estimate of pavement damage is [4]

$$DAMAGE1 = (1 + 6\,DLC^2 + 3\,DLC^4)\,\bar{F}^4$$

If peak loads are assumed to be concentrated near particular points, then an estimate of pavement damage is

$$DAMAGE2 = (1 + DLC)^4\,\bar{F}^4$$

It is possible that putting a more "road friendly" suspension on a tractor drive axle can increase the dynamic wheel loads on the attached trailer [5]. This possibility lends more motivation to optimizing the suspension of an entire vehicle to result in a reduction in road damage from the entire vehicle. The pavement damage measure was computed for each wheel and averaged to obtain one value

for the entire truck. The heavier loaded wheels, normally the trailer wheels, are the most damaging and are the dominate terms in the calculation. The two ride comfort measures are absorbed power and RMS vertical acceleration. The absorbed power criterion [6] is based on the hypothesis that ride comfort is related to energy dissipation due to internal damping in the human body. Absorbed power is determined by calculating a weighted integral of the power spectrum of acceleration at the passenger-seat interface and includes acceleration in all three dimensions. The weighting functions, which are functions of frequency, are the mechanical impedance of the human body at the passenger-seat interface, and are higher for frequencies to which the human body is most sensitive. The vertical seat acceleration is calculated using a filter; driver / seat dynamics are assumed to not have any effect on the dynamics of the rest of the truck. The seat is modeled as a spring and damper, and the driver model is a three degree of freedom spring-mass-damper system supported by the seat as in figure 1. RMS
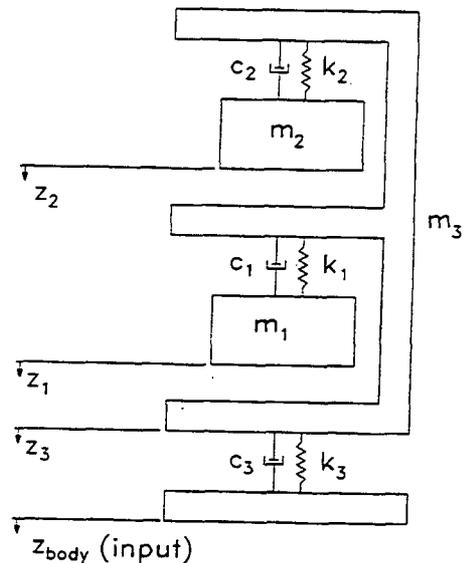
Figure 1. Absorbed Power Model

vertical acceleration is not generally used as a ride comfort criteria in heavy trucks because pitch dynamics also affect ride comfort, but is included because of its ease of use and to determine if the more elaborate criterion yields a different suspension design than the simple criterion. The vertical acceleration is computed at the driver seat baseboard. The handling criteria are drawn from a study by Woodroofe and El-Gindy [7], which is directed at establishing handling requirements for heavy trucks in Canada. The most common method of evaluating the roll stability of heavy trucks is the rollover threshold. Woodrooffe and El-Gindy suggest that the minimum acceptable rollover threshold for safe handling performance is 0.4 g's, and this is the value that is used for suspension optimization in this thesis. The yaw stability criteria used is the Three Point Measure (TPM) suggested by Woodroofe and El-Gindy [7].

1) The understeer coefficient, K , at 0.15 g's, should be between 0.5 (sensitivity boundary) and 2.0 (steerability boundary).
2) The transition from understeer to oversteer should occur at a lateral acceleration of not less than 0.2 g's.
3) The understeer coefficient, K , evaluated at a lateral acceleration of 0.3 g's must be higher than the critical understeer coefficient.

It is sufficient to evaluate understeer only for the tractor because for a tractor semitrailer vehicle, the tractor must be oversteer for any form of dynamic yaw instability to occur. All of these values are evaluated during the same maneuver, a ramp steer at 100 km/hr, which is the prescribed maneuver for evaluating the Three Point Handling Measure. Figure 2 shows understeer coefficient vs. lateral acceleration for the baseline vehicle used for this paper.

COMPUTER PROGRAMS

The computer programs used for this study are written in FORTRAN and include routines for optimization, heavy truck simulation, and performance measure calculation. The optimization routine CONMIN [8], based on the Method of Feasible Directions [9], is used for this study, and the PHASE 4 program [10] is used for heavy truck simulation. Subroutines for performance measure calculation were written specifically for this study. PHASE 4 was written at The University of Michigan Transportation Research Institute and is built around differential equations of motion, developed from Newtonian mechanics, which are numerically integrated. Verification is reported in [11] and [12]. The model is three dimensional and includes four spring and walking beam tandem leaf spring suspension models. Braking can be simulated using conventional or antilock brakes.

Lookup tables are used to define tire, spring, and brake models.

An optimum design problem generally consists of a cost function which is to be minimized, and usually one or more constraints, which are conditions or limits which must always hold for any feasible design. The general form for a constrained optimization problem is [9]

minimize

$$f = f(x)$$

subject to

$$h(x) = 0 \ , i = 1 \text{ to } p \tag{1}$$

$$g(x) \leq 0 \ , i = 1 \text{ to } m$$

$$x \leq x \leq xmax$$

where f(x), h(x), and g(x) are nonlinear functions in general. CONMIN is a flexible optimization routine which can solve the nonlinear problem stated in equation 1. Two important aspects in the use of CONMIN are the methods in which gradients and constraints are calculated. CONMIN, like most optimal design routines, requires calculation of gradients of the objective and constraint functions. The gradient is a vector of first partial derivatives of the function with respect to the design vector. Gradients can be computed using either analytical gradient functions or numerical approximations. The use of analytical gradients is desirable when possible because less function evaluations are necessary, so the optimization process is more efficient. The gradients in this study are calculated using a finite difference method. The size of the truck models, the existence of nonlinearities such as coulomb friction, and the use of look up tables to define tire and spring properties make calculation of analytical gradients difficult or impossible when performing heavy truck suspension optimization. CONMIN requires that constraints be simple inequalities of the form

$$G \leq 0,$$
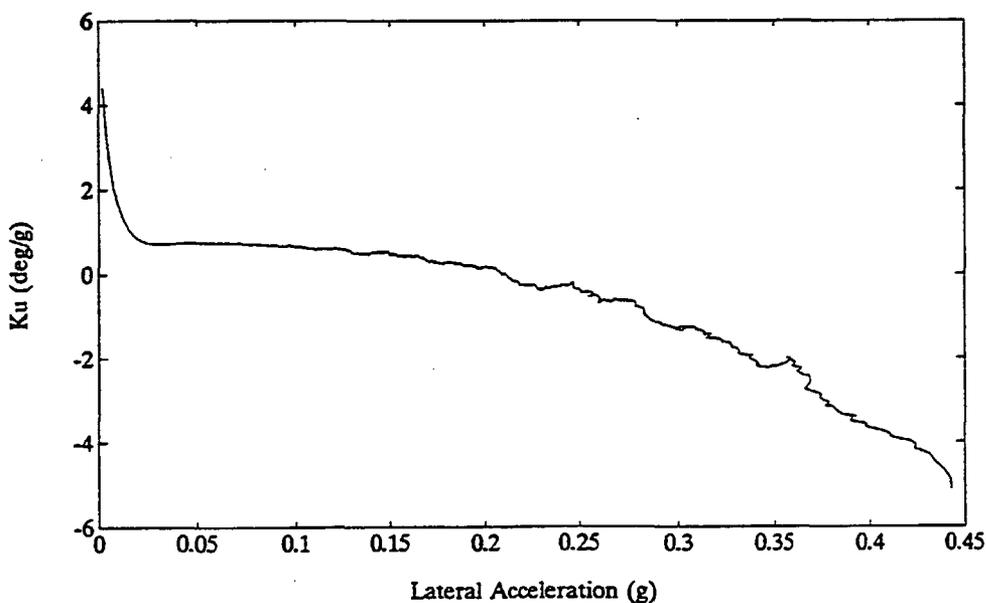
Figure 2. Baseline Vehicle Understeer Coefficient vs. Lateral Acceleration

but a rattlespace constraint is a compound inequality of the form

$$-RMAX \leq R \leq RMAX,$$

so each rattlespace constraint results in two CONMIN constraints. There are ten wheels (dual wheels are considered a single wheel) on the truck, so there are twenty rattlespace constraints.

The goal of the rattlespace constraints is to limit the maximum travel of each axle, so the obvious first choice in choosing rattlespace constraints would be to record the maximum displacement envelope for the run. Calculation of a single maximum value for each run would not be desirable, though, because changing the design to eliminate a violated constraint at one point could result in pushing the displacement envelope out at another point, a process which could possibly keep repeating, preventing efficient optimization. To avoid this problem, it is desired that the entire displacement envelope be considered at once. This can be done by calculating the relative displacement at specified time intervals throughout the run and considering the relative displacement at each point to be a constraint. This approach results in a large number of constraints, but ensures that the entire displacement envelope will be held within bounds.

In order to reduce the number of constraints, a combination of the two approaches is used in the optimization programs for this study. The simulation runs are divided into .10 second intervals, and the maximum relative displacement for each interval is saved as a constraint value at the end of the interval. This reduces the number of constraints while ensuring that a sharp peak is not missed. The ride comfort and braking routines both save constraints over a 5.0 second period for a total of 1000 rattlespace constraints, while the pavement loading routine saves constraints over a 2.5 second period for a total of 500 rattlespace constraints. Braking runs typically take less than 5.0 seconds, so the constraints are initialized to a feasible value and the unused constraints have no effect on the optimization process. The four handling constraints become five CONMIN constraints because of a compound inequality. The first is rollover threshold, the second and third are from the compound inequality for understeer coefficient at .15 g's lateral acceleration, the fourth is the lateral acceleration at the transition from understeer to oversteer, and the fifth is the understeer coefficient at .30 g's lateral acceleration. When no handling constraints are used, just one simulation run is required for each function call. If handling constraints are used, two simulation runs are required for each function call. The first run is a straight run on a rough road for calculation of pavement loading, ride comfort, or braking distance, and a second run is needed to simulate a ramp steer on a smooth road to calculate the rollover threshold and yaw stability criteria.

PROCEDURE

Forty two potential design variable were first chosen, then a sensitivity analysis for these variables was conducted to identify the significance of each variable to each performance measure. A set of design variables was chosen for each performance measure based on the results of the sensitivity analysis. Reducing the number of design variables in this manner allows a more efficient optimization process. The optimization routines were then demonstrated for a typical tractor-semitrailer using twelve different run sets that include all the objective functions with and without handling constraints. In the sensitivity analysis, each of the forty two variables was varied ±5% around a baseline value while calculating the desired objective function or constraint to determine the sensitivity of the objective function or constraint to that variable. The variables were varied one at a time. A different set of design variables was used for each optimization set. Those variables which show to be significant for a particular performance measure in the sensitivity analysis were used when optimizing for that performance measure. Spring rate and viscous damping were optimized for each axle for all runs, even if they were not significant for a particular measure, because they are the most easily and commonly varied parameters. Spring rate and viscous damping can also be varied over a wide range, and so might be used to measurably improve a design even if a variation of 5% in the sensitivity analysis did not have a significant effect. Realistic upper and lower limits on design variables were chosen primarily by considering typical ranges of values currently found on trucks. For example, the tractor is intended to model a Ford LTA-9000 Series Aeromax tractor, so the limits used for wheelbase are the largest and smallest wheelbases currently available from the manufacturer. Typical ranges for many variables may be found in [13]. In practice, limits would often be dictated by packaging constraints or by economic factors such as use of existing tooling or common parts. The optimizations with handling constraints use more design variables because the variables which showed significance for the objective function plus those which were significant for the handling constraints were used. The largest number of design variables in any set used for the optimizations was nineteen for ride comfort with handling constraints. There are many possible optimization studies which could be run with the performance measures programmed for this study.

Some of the design variables, such as unsprung mass and tandem axle static load transfer, are not directly specified by the designer, but can be considered to be subsystem properties. Desired subsystem properties may be found in the total vehicle optimization, and then subsystem component properties may be optimized in an additional study.

Twelve run sets were used which cover all of the basic cases and demonstrate usage of the programs. The sensitivity analysis used two speeds and three road roughnesses, but in order to keep the optimization study tractable, only the worst case conditions were used - high speed and high roughness. Pavement loading and ride comfort runs were all at 55 mph, stopping distance was found from 60 mph, and handling constraints were calculated at 62 mph (100 kph). The twelve run sets were

1) Pavement damage criteria 1, no handling constraints

2) Pavement damage criteria 2, no handling constraints

3) Absorbed power, no handling constraints

4) RMS vertical seat acceleration, no handling constraints

5) Stopping distance, no handling constraints

6) Rollover threshold (unconstrained maximization)

7) Understeer/oversteer transition point (unconstrained maximization)

8) Pavement damage criteria 1 with handling constraints

9) Pavement damage criteria 2 with handling constraints

10) Absorbed Power with handling constraints

11) RMS vertical seat acceleration with handling constraints

12) Stopping distance with handling constraints.

Two types of road inputs were necessary for this study: a straight rough road for pavement loading, ride comfort, and braking, and a smooth, flat turn for handling. Mathematically generated profiles were used in this study to obtain generic profiles with uniform frequency content instead of optimizing for a specific measured profile. The model used for road profile generation, given in [14], is

$$\Phi(n) = \begin{cases} \Phi(n_0) \dfrac{n}{n_0}^{-W_1}, & n \leq n_0 \\ \\ \Phi(n_0) \dfrac{n}{n_0}^{-W_2}, & n \geq n_0 \end{cases} \quad (2)$$

where $\Phi(n_0)$ is the roughness amplitude coefficient, n is the wave number, n_0 is the discontinuity wave number, and the exponents w_1 and w_2 control the frequency characteristics. Values of $\Phi(n_0)$, w_1 and w_2 are given in [14] for different types of roads. MATLAB [15] was used to create profiles with the spectral density given by equation 2. White noise was generated and then filtered in the frequency domain using FFT analysis to obtain the desired power spectral density. The roughness of the profile used for the optimization procedure is

> Left track: 206.9 IPM
> Right track: 213.2 IPM
> where IPM is inches per mile.

Optimization parameters for CONMIN greatly affect the success and efficiency of the optimization process, and selection of the best values for a particular problem requires user experience or trial and error. Each of the case studies without handling constraints, cases 1 through 7, consisted of six computer runs with different optimization parameters and initial designs. These runs were used to find the best designs for each objective function. Cases 8 through 12 then show the effect of enforcing the handling constraints on each performance measure. The number of runs made for each of these cases varied, but several runs were again made for each case with different optimization parameters and initial

designs. The cases without handling constraints were also valuable for gaining experience in selecting optimization parameters, because much less CPU time was required. The runs with handling constraints took much longer to run because more simulations were required for each iteration.

RESULTS

The optimization routines were generally well behaved and were, after initial set-up, very efficient because the choice and number of design variables was easily changed and multiple cases could be run in batch mode. The best results were obtained when the design variables were normalized, but it is possible that a different type of scaling might be even more efficient.

Significant improvements were obtained for each of the individual measures, as shown in table 1. Complete optimization results are in references [16] and [17]. As one would expect, the results of the optimizations show that optimizing a truck suspension for overall performance requires compromises. Short tractor and trailer wheelbases were best for reducing pavement damage, but best ride comfort and shortest stopping distances occured for maximum tractor wheelbase. Locating the fifth wheel at the forward limit was the best design for reducing pavement damage, but it moved back to near the baseline position when handling constraints were enforced. Tractor springs were soft for best ride comfort, but had to be stiffer to meet handling constraints. The highest rollover threshold obtained was 0.413 g's, showing that a minimum threshold requirement of 0.40 g's is feasible but restrictive. Fifth wheel stiffness, rear tractor spring rate, trailer wheelbase and trailer roll center height were all at or near upper limits for the design with the best rollover threshold.

The individual performance measures are valuable as a means of solving particular design problems or for finding a preliminary design. A designer given the task of improving ride comfort, for instance, could use the optimal design for ride comfort as a guide to show where a production design could be improved. If the primary design goal for a new truck was to obtain a "pavement-friendly" vehicle, a preliminary design could be obtained by optimizing for minimum pavement damage while enforcing handling constraints. The design could then be "tweaked" by the designer to improve other performance aspects.

CONCLUSIONS

A computer based method for parameter optimization of conventional heavy truck suspensions for overall vehicle performance has been presented and demonstrated. This type of procedure could be a valuable tool for preliminary design, design modification to improve an existing problem, or custom design to meet a particular customer's needs. Unintended effects can be avoided if overall vehicle performance is considered when optimizing components and subsystems, an important consideration for heavy trucks because of the many combinations of tractors, trailers, and tires which can occur. Improved user interfaces will make this type of tool even more valuable and practical.

Table 1: Summary of Optimization Results

DAMAGE1 without handling	11.4% improvement	Best value: 3.597*10 lb
DAMAGE1 with handling	8.1% improvement	Best value: 3.733*10 lb
DAMAGE2 without handling	12.2% improvement	Best value: 4.870*10 lb
DAMAGE2 with handling	8.3% improvement	Best value: 5.088*10 lb
Absorbed Power without handling	40% improvement	Best value: 7.920 watts
Absorbed Power with handling	48% degradation	Best value: 19.596 watts
RMS Vertical Acceleration without handling	15.7% improvement	Best value: 1.871 ft/sec
RMS Vertical Acceleration with handling	9.5% improvement	Best value: 2.008 ft/sec
Stopping Distance without handling	4.92% improvement	Best value: 201.1 ft
Stopping Distance with handling	3.11% improvement	Best value: 204.9 ft
Rollover Threshold	5.95% improvement	Best value: .413 g's

REFERENCES

1) Bender, E.K., Karnopp, D.C., Paul, I.L., "On the Optimization of Vehicle Suspensions Using Random Process Theory", ASME Paper No. 67-TRAN-12, 1967.

2) ElMadany, M., "A Procedure for Optimization of Truck Suspensions", Vehicle System Dynamics, v16, n5-6, 1987, p297-312.

3) "The ASSHO Road Test, Report 5, Pavement Research", Highway Research Board Special Report 61E, National Academy of Science, National Research Council Publication 954, 1962.

4) Mitchell, C. G. B. and Gyenes, L., "Dynamic Pavement Loads Measured for a Variety of Truck Suspensions", Proceedings of Second International Symposium on Heavy Vehicle Weights and Dimensions, Kelowna, British Columbia, June 1989.

5) Gillespie, Karamihas, Cebon, Sayers, Nasim, Hansen, Ehsan, Effects of Heavy Vehicle Characteristics on Pavement Response and Performance, The University of Michigan Transportation Research Institute, Report No. UMTRI 92-2, 1992.

6) Lee, R.A. and Pradko, F., "Analytical Analysis of Human Vibration", SAE Paper No. 680091, 1968.

7) Woodrooffe, J.H. and El-Gindy, M., "Application of Handling and Roll Stability Performance Measures for Determining Suitable Tractor Wheelbase", Proceedings of the Third International Symposium on Heavy Vehicle Weights and Dimensions, Cambridge, UK, 1992.

8) Vanderplaats, G. N., CONMIN - A Fortran Program for Constrained Function Minimization, User's Manual, NASA-TM-X-62,282, August 1973 (with May 1978 Addendum).

9) Arora, J. S., Introduction to Optimal Design, McGraw-Hill, 1989.

10) MacAdam, C. C., Fancher, P. S., Hu, G. T., Gillespie, T. D., A Computerized Model for Simulating the Braking and Steering Dynamics of Trucks, Tractor Semi-Trailers, Doubles, and Triples Combinations: User's Manual - Phase 4, Highway Safety Research Institute, UM-HSRI-80-58, Ann Arbor, Mich, September 1980.

11) Gillespie, T., MacAdam, G., Hu, G., Bernard, J., Winkler, C., Truck and Tractor-Trailer Dynamic Response Simulation, FHWA-RD-79-124, Volume 2, Technical Report, December 1980.

12) El Gindy, M., Wong, J.Y., "A Comparison of Various Computer Simulation Models for Predicting the Directional Responses of Articulated Vehicles", Vehicle System Dynamics, Vol. 16, No. 5-6, 1987.

13) Fancher, P.S., Ervin, R.D., Winkler, C.B., Gillespie, T.D., A Factbook of the Mechanical Properties of the Components for Single-Unit and Articulated Heavy Trucks, DOT-HS-807-125, 1986.

14) Dobbs, C. J., Robson, J. D., "The Description of Road Surface Roughness", Journal of Sound and Vibration (1973) 31(2), 175-83.

15) 386-MATLAB, The Mathworks, Inc., Natick, MA, 1990.

16) Blue, D.W., Methods for Optimization of Heavy Duty Truck Suspension Design and Control, Ph.D.

Dissertation, The Pennsylvania State University, 1993.

17) Blue, D.W., Programming Details and Results for Optimization of Heavy Duty Truck Suspension Design and Control, PTI 9410, The Pennsylvania Transportation Institute, 1993.