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ABSTRACT

This paper presents a methodology for validating vehicle stability and control computer simulations. Validation is defined as showing that, within some specified operating range of the vehicle, a simulation's predictions of a vehicle's responses agree with the actual measured vehicle's responses to within some specified level of accuracy. The method uses repeated experimental runs at each test condition to generate sufficient data for statistical analyses. The acquisition and reduction of experimental data, and the processing path for simulation data, are described. The usefulness of time domain validation for steady state and slowly varying transients is discussed. The importance of frequency domain validation for thoroughly validating a simulation is shown. Both qualitative and quantitative methods for the comparison of the simulation predictions with the actual test measurements are developed.

To illustrate the validation methodology, experimental testing of four different vehicles was performed; with the results being compared with predictions made by two existing vehicle stability and control simulations. Constant steer, braking in a turn, straight line braking, double lane change, and sinusoidal sweep steering maneuvers of varying severity were performed to study simulation validity for a wide range of vehicle operating conditions. Comparisons between the actual test measurements and the simulation predictions are shown.

DURING THE LAST 30 YEARS, substantial effort has gone into the development of many vehicle stability and control computer simulations. Unfortunately, much less effort has gone into the important question of the validity of these simulations. The experimental testing of full scale vehicles, for use in a validation procedure, is quite expensive and time consuming. Because of this, many vehicle dynamics simulations have had little or no validation work performed. Many modified and new simulations have been compared with predictions from existing simulations as the sole check of their validity. Others have been experimentally substantiated only for limited vehicle operating conditions and then assumed to be valid for all other operating conditions.

The National Highway Traffic Safety Administration (NHTSA) desires a vehicle stability and control simulation that can simulate a wide range of light vehicles (passenger cars, pickup trucks, vans, and utility vehicles) in a broad range of cornering and braking maneuvers. NHTSA is studying existing simulations, selecting the most appropriate one for its purposes, and improving it to resolve problems identified during the selection process. The simulation validation methodology and procedure described in this paper was developed as part of this work.

For a particular application, the validation methodology described here can distinguish the most appropriate simulation out of a group of simulations. In addition, it has proven to be very useful for identifying measurement errors in the simulation vehicle parameters, and for finding transducer offset or calibration errors in experimental data channels. In the modification and improvement stages of simulation development, the procedure is helpful when attempting to determine the specific problem areas of a simulation model.

The validation methodology contained in this paper can be used to validate a variety of vehicle dynamics simulations besides the stability and control types discussed herein. These include simulations dealing with tripped or untripped vehicle rollover, ride quality, and dynamic models used in real time man-in-the-loop simulation. This validation methodology also can be extended and used for validating computer simulations of many other types of physical systems.

BACKGROUND

Validation work using full scale vehicle test results has been performed for several of the vehicle stability and control simulations developed during the past two decades. A few examples of past simulation validation work are discussed below.

One simulation involving substantial validation effort is the Highway-Vehicle-Object Simulation Model (HVOSM) (1)*. This simulation was originally developed in the late 1960's and has matured through several versions to its present form. It includes general three-dimensional motions resulting from vehicle control inputs, traversals of terrain irregularities, and collisions with certain types of roadside obstacles.

HVOSM's predictions were compared with experimental results for various handling and accident maneuvers. These included sinusoidal steering, ramp traversal, skidding, bridge rail impact, straight ahead braking, and braking in a turn tests. Although the validation work appears adequate for steady state conditions, little attention appears to have been paid to certain maneuver regimes, especially the transient response region. No validation work in the frequency domain appears to have been performed. Finally, it appears that some adjustment of vehicle parameters to better match simulation predictions to experimental vehicle responses occurred.

Two modified versions of the HVOSM simulation were implemented in the early 1970's at the Applied Physics Laboratory of the Johns Hopkins University (APL/JHU). The principal modifications were the addition of a steering system model and the removal of obstacle impact dynamics. The resulting simulations, named the Hybrid Computer Vehicle Handling Program (HVHP) (2) and the Improved Hybrid Computer Vehicle Handling Program (IHVHP) (3), were installed on a hybrid computer to reduce computing time and cost. In 1979, an all digital version of these simulations was implemented at the University of Michigan as the Improved Digital Simulation, Fully Comprehensive (IDSFC) (4). This simulation was used in the current research.

For validation purposes, the predictions of the HVHP, IHVHP, and IDSFC simulations were compared with vehicle field test data for several handling maneuvers. The validation of these simulations, as with the previous versions of HVOSM, had little attention paid to the transient response regime. No validation work in the frequency domain was performed. As a result, the simulation's poor performance in predicting transient behavior was never identified. Finally, the vehicle parameter measurement techniques that were available to the researchers were not very accurate. As a result, adjustment of vehicle parameters to better match vehicle responses occurred, making the entire validation suspect.

In the mid 1980's Systems Technology, Incorporated (STI) developed the Vehicle Dynamics Analysis Non-Linear (VDANL) simulation for analyzing vehicle lateral/directional control and stability (5). This simulation differs from the preceding ones primarily through its suspension and tire models. It was validated, by comparison withvehicle field test data for several handling maneuvers (6). The maneuver matrix used by STI was not extensive enough to provide simulation validation for the full range of the vehicle operating conditions currently of interest to NHTSA. In addition to doing simulation validation work in the time domain, STI performed validation in the frequency domain. This involved comparison of the experimentally measured yaw rate to handwheel steering angle frequency response function (magnitude and phase angle) with the simulation's predictions. During validation of this simulation, some adjustment of vehicle parameters to better match vehicle responses occurred. The STI simulation was also used in the current research.

The preceding cases are typical of simulation validation work in the literature. While the performers of the work did the best job that they could with their available resources, usually there was insufficient testing to cover all maneuver regimes of interest. Often, there was little attention paid to the transient maneuvers or the frequency domain. Finally, due to parameter measurement problems, there was a strong tendency to adjust the vehicle parameters to make the simulations' predictions match the experimentally measured vehicle responses.

WHAT IS SIMULATION VALIDATION?

A computerized, mathematical model of a physical system, such as a vehicle stability and control simulation, will be considered to be valid if, within some specified operating range of a system, a simulation's predictions of a system's responses of interest to specified input(s) agree with the actual physical system's responses to the same input(s) to within some specified level of accuracy.

The above definition contains several points that need to be explained in greater detail.

First, a simulation's predictions will, in general, only be correct within some portion of the system's operating range. An obvious example of this is that a vehicle dynamics simulation's predictions may be correct for low lateral acceleration maneuvers but become progressively worse as lateral acceleration increases and non-linear effects become more important. A second example is that, due to an incorrect modeling of the effects of steering system compliance, a simulation's predictions may get worse as steer angle increases.

Similarly, a simulation's predictions may only be correct for inputs that predominantly contain (following Fourier decomposition) frequencies within a specified range. Many vehicle dynamics simulations are valid for steady state and slowly varying input conditions but have problems with fast transients that contain high frequencies.

This last point illustrates why simulation validation needs to be performed in both the time and frequency domains. Validation in the time domain is good for demonstrating that the simulation can correctly predict steady state

*Numbers in parenthesis represent references at the end of this paper.

conditions and that non-linear effects are properly modelled. High frequency transient phenomena, however, are very difficult to study in the time domain. The effects of increasing input frequency on the correctness of a simulation's predictions are best determined through frequency domain studies.

A second significant point in the definition of simulation validation is that validity is determined only for specified groups of inputs and outputs. For example, in a vehicle stability and control simulation, simply because the simulation has been shown to be valid for braking and steering control inputs does not imply that the response to a road disturbance (such as a bump in the road) will be correctly predicted. Similarly, a simulation that successfully predicts lateral sprung mass acceleration might fail to predict vertical sprung mass acceleration.

The third significant point is whether a simulation can be considered to be valid depends upon how much the simulation's predictions can acceptably vary from the actual test results at a given operating point. The degree of accuracy required to classify a simulation as valid, depends on the intended uses of the simulation and level of accuracy believed to be attainable. If only the trends of the response of a physical system are to be simulated, with little interest in predicting values, much less accuracy is required than when trying to predict exact values.

Every experimental measurement contains random error superimposed onto the signal. Random errors are defined here as transducer measurement noise, unaccounted for variations disturbing the system's inputs, and random minor changes in the system. For vehicle testing of the type covered in this paper, random error would include, for example, the effects of wind gusts, road roughness, tire non-uniformity, and changes in the brakes from test to test. Note that there are other sources of experimental non-repeatability including, for example, variability in the inputs.

Some of the disagreement between a simulation's predictions and experimental measurements will be due to this random error. Since simulations cannot predict random error, if a simution's predictions agree with experimental measurements to within the experimental random error level, the simulation should be considered valid.

Conversely, while simulations whose predictions are less accurate than the experimental random error level are useable for many tasks, the "best" simulation (for a specified operating regime and input/output set) of a system is one whose predictions agree with experimental measurements to within the random error level. It is recognized that practical considerations prevent the development of a "best" simulation for many physical systems. However, this level of accuracy should be used as the benchmark in the development of simulations.

The easiest way to determine the experimental random error level present in data is to repeat all experimental runs multiple times. Given data from several tests, statistical procedures can be used to calculate the random error levels.

Finally, two important points about simulation validation that are not in the above definition are:

1. The parameters used to describe the physical system to a simulation must be measured independently (<u>not</u> from the experiments that obtain simulation validation data).

2. While validating a simulation, the parameters describing the system to a simulation must not be varied from their independently measured values to improve the accuracy of a simulation's predictions.

If either of these conditions are not met, then a simulation is not actually being validated. Instead, all the researcher is doing is showing that, by adjusting one or more parameters, curves can be generated to match experimental data. This process is not always unique; there may be several ways to change parameters to make simulation predictions match experimental measurements. This type of validation work tells nothing about a simulation's ability to predict system performance when experimental data is not present. Note that a simulation for which this sort of validation has been performed is useful for studying vehicles for which experimental data is available.

SIMULATION VALIDATION METHODOLOGY - GENERAL CASE

Figure 1 is a block diagram of the basic steps in our methodology. Shown is the flow of information through two processes, designated the experimental and the simulation processes. These steps illustrate the general stages required to validate a computer simulation of a physical system. However, the steps shown in this diagram are not entirely mandatory, as specific details of data processing are unique to individual research programs.

The start of the experimental process is obtaining, via experimental testing, reliable measurements of the behavior of the physical system. As was explained in the preceding section, repeat runs of the same experimental test should be made. This gives a measure of the random error present in the experiment and also improves the reliability of the experimental results.

The physical system responses of interest must be appropriately measured and recorded. Usually, the inputs, or forcing functions, used to excite the physical system are also recorded. These inputs are then used to drive the simulation during the validation process. For certain types of physical systems, the inputs are predetermined and sufficiently well defined that they need not be measured. However, in most cases, the physical system's driving functions are not accurately known but must be measured during system testing in order to precisely define their characteristics.

Typically, the experimental test data requires a data reduction stage along its processing route. This stage usually includes transforming measured electrical signals into engineering units. Also, this stage of data reduction usually includes

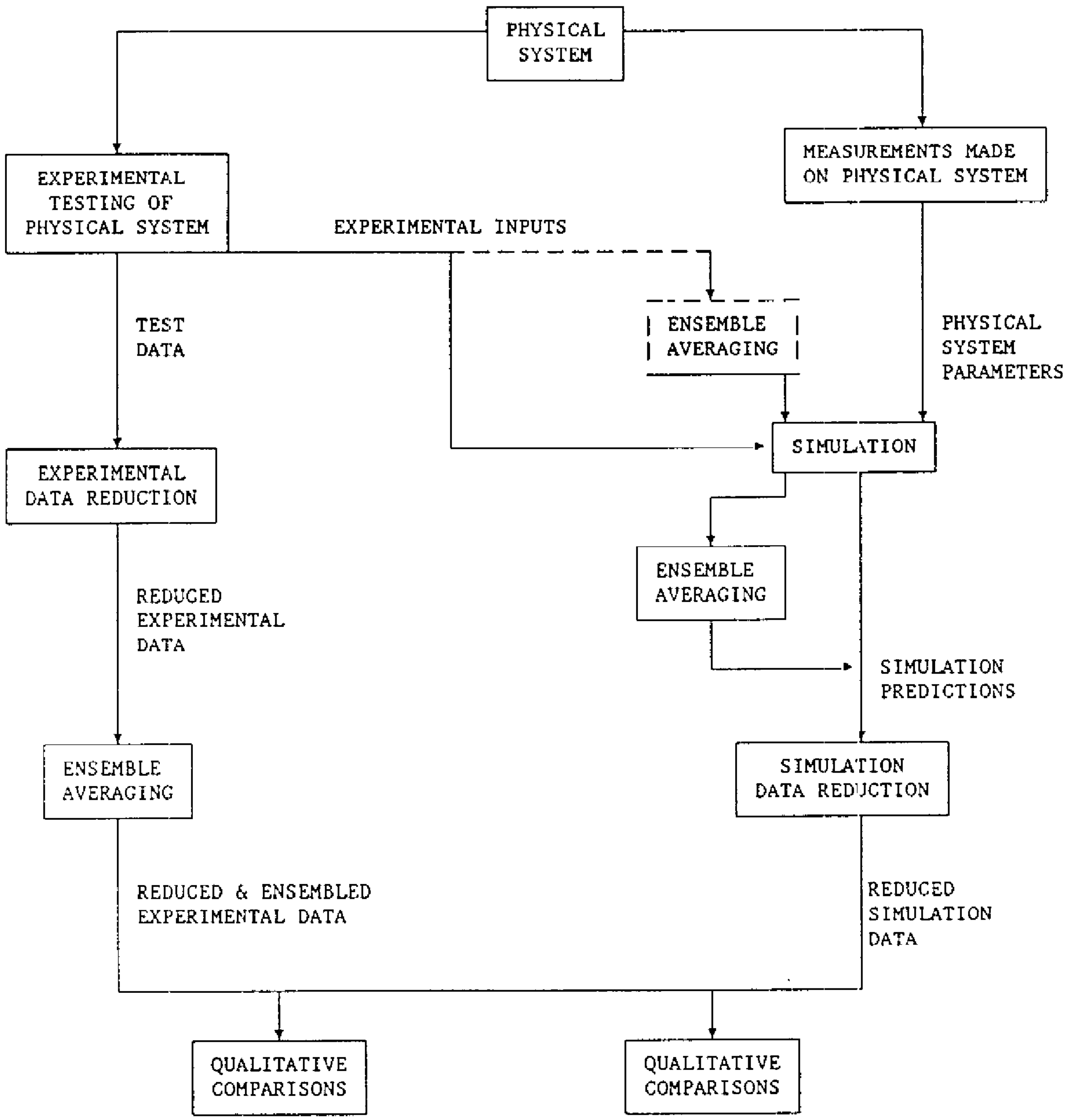

Fig. 1 - Simulation validation process data flow

digital filtering, to reduce spurious noise in the measured data. Other data reduction operations that might be performed at this point include the Fourier transformation of data into the frequency domain, the computation of experimental quantities that cannot be measured directly, and the subsampling of the data to reduce the size of data files.

The next step in the flow of experimental data is the ensemble averaging of the repeated test runs. This involves computing the mean values as a function of the independent variable (time or frequency) for each of the responses of interest. It also allows the use of statistical methods to compute a measure of experimental repeatability, the 95 percent confidence interval of the mean value. The result of this step in the block diagram is the reduced and ensembled data (mean values and confidence limits) from the experimental testing of the physical system.

Next, consider the flow of information through the simulation process. A simulation is an analytical model of the behavior of a physical system. This model includes equations that require parameters that represent the physical system. These physical system parameters, such as mass/inertia, damping, compliance, geometry, etc., must be measured (or estimated when necessary) and supplied to the simulation model.

For a simulation to accurately predict a specific physical behavior, as in simulation validation, it must be driven with the same inputs as the physical system. These driving inputs must be known (or measured) for each physical system response to be simulated.

The validation methodology presented here is based on the analysis of data from repeated experimental runs. In general, a simulation should be driven by the separate, measured, inputs from each run, followed by the ensemble averaging of a simulation's predictions. This corresponds to the case shown in the block diagram with solid lines.

From a practical point of view the above procedure is time consuming and expensive. Although the correct procedure is to ensemble average following simulation, if the individual inputs from the repeated experimental runs are sufficiently alike and a simulation is not highly nonlinear, the inputs may be ensemble averaged before they are feed into a simulation. This corresponds to the case shown in the block diagram with dotted lines.

Provided both of the above conditions are met, the predictions obtained by feeding the mean of the ensemble averaged inputs into a simulation should closely approximate the mean obtained by ensemble averaging simulation predictions from the separate runs. For this situation, multiple runs of the simulation have been reduced to a single run.

Once the simulation predictions have been obtained, regardless of when the ensemble averaging was performed, simulation data reduction may be necessary. This stage may include Fourier transforming data to the frequency domain or subsampling the data to reduce data file size. The result of this stage of the simulation process is reduced simulation data.

Upon completion of the above steps, the experimentally measured, reduced and ensembled averaged, test data is compared both qualitatively and quantitatively to the reduced simulation data. Qualitative and quantitative comparison schemes will vary depending on the nature of the physical system and simulation. Some simulations and experiments may readily provide results that are best suited for qualitative comparison. In other instances, quantitative comparison may furnish the best validation mechanism. Usually, some combination of both qualitative and quantitative comparison, as is the case for validation of the vehicle stability and control simulations discussed in this paper, provides for the most thorough validation.

The qualitative comparison scheme used in the current research consists of overlaying plots of the experimental mean and the simulated mean versus an independent variable of either time or frequency. The 95 percent confidence limits of the experimental mean are also plotted to indicate the level of experimental variability. Researchers can then observe the level of agreement between a simulation's prediction and the actual, measured, data.

Quantitative comparison schemes presented include the comparison of computed steady state gains, response times, peak response times and percent overshoots from simulated and experimental time domain data. Frequency domain comparisons include peak frequency, peak amplitude ratio, bandwidth, and phase angle metrics. Other methods, which are being explored by the authors, involve the use of statistical methods for comparing simulation predictions with experimental mean values and confidence intervals.

VALIDATION OF VEHICLE STABILITY AND CONTROL SIMULATIONS

As was previously mentioned, a goal of the current research is to develop a vehicle stability and control simulation that can simulate a wide range of vehicles, and be valid for a broad range of crash avoidance maneuvers. The first step in the research was to study two existing vehicle stability and control simulations, namely the STI developed VDANL simulation and the University of Michigan developed IDSFC simulation. The simulation validation methodology presented here was developed for use during this study. The simulation validation procedure is being used to find areas of disparity between these simulations and field test results. Once problem areas have been identified, simulation model improvements can be implemented to reduce simulation disagreement. This work is expected to result in a more accurate (for the operating conditions and maneuvers of interest) vehicle stability and control simulation than any developed for NHTSA to date.

Four vehicles were used in this vehicle stability and control simulation validation research. Each vehicle has been processed through all the steps shown in Figure 1. The following sections describe in greater detail how the experimental and simulation processes have been performed for the four vehicles. This is followed by sections showing typical qualitative and quantitative comparison results.

The four vehicles used in this research are significantly different in size, shape, and design function. The test vehicles included a small and mid-size passenger car, a small utility vehicle, and a standard size van. The four vehicles tested were:

1. A 1987 Ford E-150 van
2. A 1987 Ford Thunderbird (mid-size) passenger car
3. A 1987 Hyundai Excel (small) passenger car
4. A 1988 Suzuki Samurai (small) utility vehicle

EXPERIMENTAL DATA COLLECTION AND PROCESSING

Figure 2 shows, in somewhat greater detail, the experimental data collection and processing portion of Figure 1 as was performed for the four test vehicles.

The first step was the experimental testing of each vehicle. For each vehicle, several types of vehicle maneuvers were performed to cover a broad range of crash avoidance situations. Five major types of vehicle maneuvers were studied. These were:

1. Constant Speed, J-Turns
2. Straight Line Braking
3. Braking in a Turn
4. Double Lane Changes
5. Sinusoidal Sweep Steering

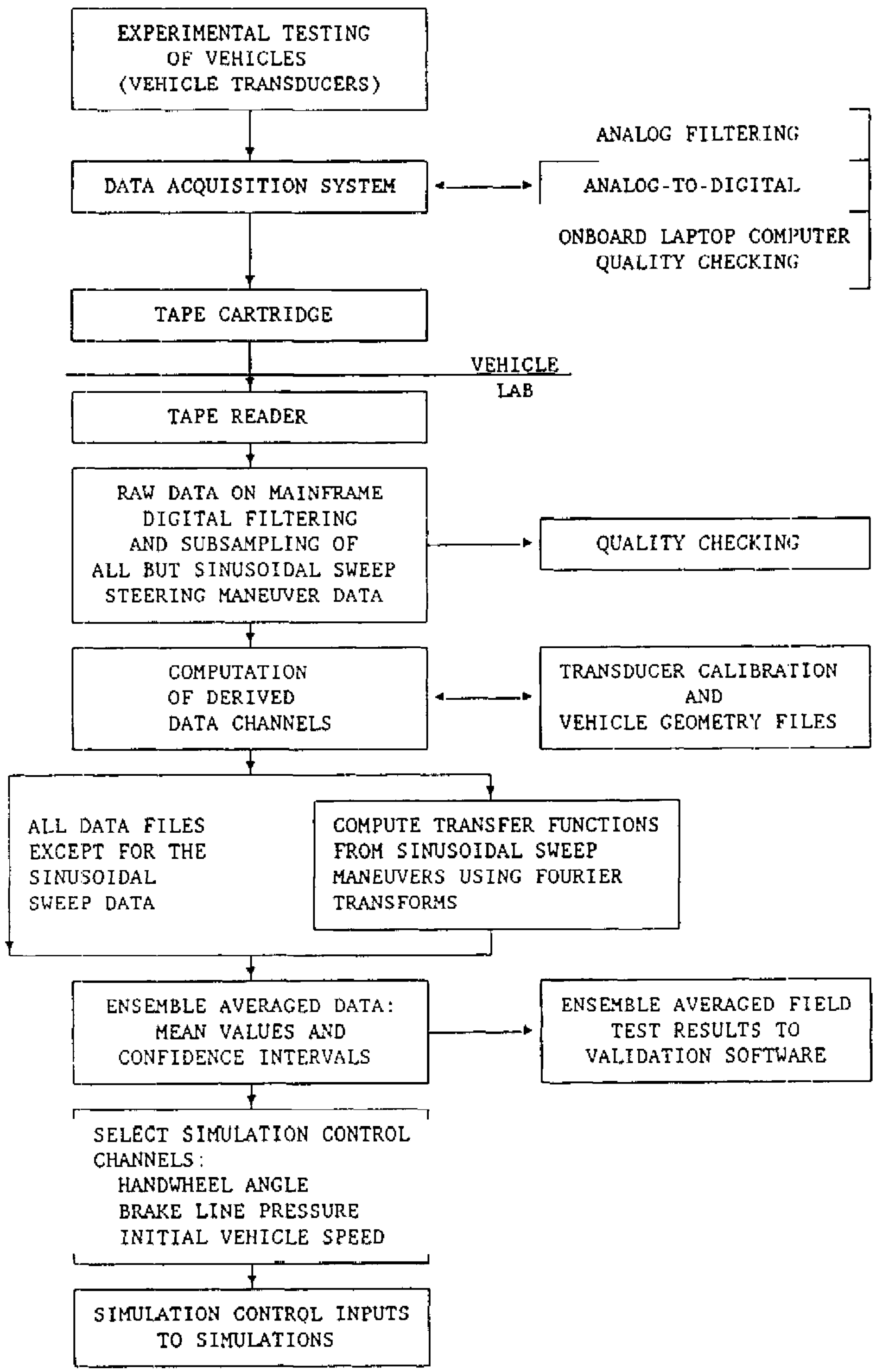

Fig. 2 - Experimental (vehicle field test) data flow

The constant speed J-turn maneuver provides information on the simulation's ability to predict lateral direction attributes of the vehicle, the straight line braking maneuver allows for the study of longitudinal attributes, and the braking in a turn combines both lateral and longitudinalresponse characteristics. Double lane change maneuvers, which can be considered representative of some crash avoidance scenarios, are also useful for studying lateral vehicle response. The last maneuver, sinusoidal sweep steering was used to study the frequency domain response of each vehicle.

For each of the above maneuver types, 2 to 18 specific test cases were run. Each case was run at a different severity level or speed. For example, for straight line braking tests, cases were run with nominal deceleration levels of 0.2 g., 0.4 g., 0.6 g., and at the maximum deceleration achievable by the vehicle without wheel lockup. In addition, for maneuvers involving turns, cases were run with both right turns (positive lateral acceleration) and left turns (negative lateral acceleration).

Ten repeat runs were made for each test case. Later, six of these runs were selected for further analysis. Runs were selected for analysis to minimize the range in the vehicles' test speed at the start of each run. For example, for a straight line braking test case, the vehicle velocity used for selecting particular test runs was the speed at the instant of brake application. The six test runs with actual initial velocities closest to the test case's desired initial velocity were selected. This run selection criterion was developed so test runs with too high or low initial velocities, thus increasing the random error level of the results, would be excluded.

A total of approximately 40 cases, with approximately 400 individual tests runs were run for each of the four test vehicles.

Each test vehicle was instrumented to measure all control inputs and vehicle outputs that were expected to provide useful comparisons with simulation predictions. This resulted in the installation of 29 transducers per vehicle. The transducers used included linear and rotary potentiometers, tachometers, pressure gages, angle and rate gyros, accelerometers, and torque dynamometers. These devices were mounted to provide direct measurement of such quantities as: vehicle sprung mass lateral, longitudinal, and vertical accelerations, vehicle yaw rate, sprung mass pitch and roll angles, vehicle speed, front and rear brake line pressures, handwheel steering angle, handwheel steering torque, suspension motions, roadwheel steering angles, and roadwheel angular velocities.

The output signals from the in-vehicle transducers are collected by a Megadac 2210C digital data acquisition system. The Megadac performs four main functions.

First, it conditions the incoming signals. This consists of amplifying the transducer signals to maximize analog-to-digital converter resolution and analog filtering them to reduce power supply and mechanical noise. The analog filtering performed varies from channel to channel; most channels are passed through a 20 Hertz, 4 pole active filter.

Second, it converts data from analog to digital form. Data is digitized by a 16 bit analog-to-digital converter at a rate of 100 samples per second per channel.

Third, the Megadac stores the digitized data onto a tape cartridge for later analysis.

Fourth, a laptop computer in the test vehicle, working in conjunction with the Megadac, is used by the test driver to read data from the tape and check its quality. The driver can use the on-board laptop to check for dead transducer channels and for proper transducer calibration values.

Upon completion of a series of test runs, the Megadac tape cartridge is taken from the vehicle into the lab. There the data is read by a second Megadac and then transferred to a VAX 11/750 computer. The VAX was used for all subsequent processing of the experimental data.

Once the data has been transferred to the VAX the data reduction stage begins. The data is converted into engineering units and additional quality checking of the data is performed, looking for transducer or data collection errors. Digital filtering and decimation are also part of this stage, to further reduce high frequency noise and to reduce the size of the data files.

For the vehicle dynamics studies considered in this research, the highest frequency of interest is about 3 Hz. The vehicles' lateral acceleration, yaw rate, and roll angle fundamental frequencies in response to a steering input are all below 1.5 Hz. Based on this, all data that was to be analyzed only in the time domain was digitally filtered using a low pass, 12 pole, phaseless[1] Butterworth filter with a cutoff frequency (-3 dB down amplitude) at 3 Hz.[2] and then subsampled to 25 samples per second per channel. Since the original data collection rate was 100 samples per second per channel, subsampling reduces the amount of data to be analyzed by three-fourths. Data which was to be analyzed in the frequency domain, i.e., data from the sinusoidal sweep steering maneuver was neither subsampled nor filtered.

As shown in Figure 2, the next step in the data reduction stage is the computation from the original data channels of additional experimental (derived) data channels. The derived data channels are system responses of interest that cannot be measured directly by the transducers mounted in the vehicle but must instead be computed from the original, measured, data channels. For example, the experimental longitudinal wheel slips are not

[1]Phaseless digital filtering is performed by first processing the data with a 3 pole filter while going forward in time, then applying another 3 pole filter while going backwards in time. This process is repeated twice for a 12 pole filter. Note that phaseless filtering cannot be performed in real-time.

[2]Frequency chosen by authors by examining the filtered and unfiltered data. Filtering at this frequency removes unwanted ride and engine vibration effects while only minimally affecting the responses of interest.

easy to directly measure. They can, however, be computed from the wheel angular velocities and the forward velocity of the vehicle.

The next step in the data reduction stage is the computation of vehicle frequency response functions.[3] As was previously mentioned, knowledge of experimental vehicle response in the frequency domain is vital to thoroughly validate a simulation.

For this research, all frequency response functions are computed from sinusoidal sweep steering maneuvers. Sinusoidal sweep steering maneuvers are performed by sweeping the steering input in a smooth manner from the lowest to the highest frequency physically attainable by the driver while maintaining a generally straight vehicle path. Any transient steering input can be used to compute the frequency response functions using appropriate Fourier transformations. However, sinusoidal sweep steering inputs typically result in a flatter power spectral density over the frequency range of interest.

Along with the frequency response function, the coherence function (a function that indicates system noise and/or nonlinear system behavior) was also computed. For the frequency response functions found during the current research, the coherence function was very near unity (a value of unity indicates noise-free, linear, system behavior) except in frequency ranges where the amplitude ratios exhibited very low valleys. In these ranges, poor coherence is to be expected since very little system response is present and even a small amount of random error dominates the measured signal (7). Although non-linearities are accounted for in the simulation models, the high coherence values show that, during the sinusoidal sweep steer maneuvers used to generate the frequency domain results, the vehicles were being operated in a mostly linear manner.

Shown in Figures 1 and 2, the next stage in the experimental data processing is ensemble averaging the reduced test data. As was previously discussed, for the purpose of simulation validation, it is imperative that the repeatability of the experimental measurements be known. This is achieved by making repeated experimental runs of each test case and using ensemble averaging and statistical methods to analyze data from the repeated runs.

As has been stated, ten repeat runs were made for each test case. Up to this point, data from all ten runs has been analyzed. However, at the beginning of the ensemble averaging stage, six of these runs were selected for further analysis. Runs were selected for analysis to minimize the range in the vehicles' test speed at the start of each run.

For test results that are being analyzed in the time domain, each data channel's mean values are computed for each increment of time. Figure 3 shows typical measured lateral acceleration data.

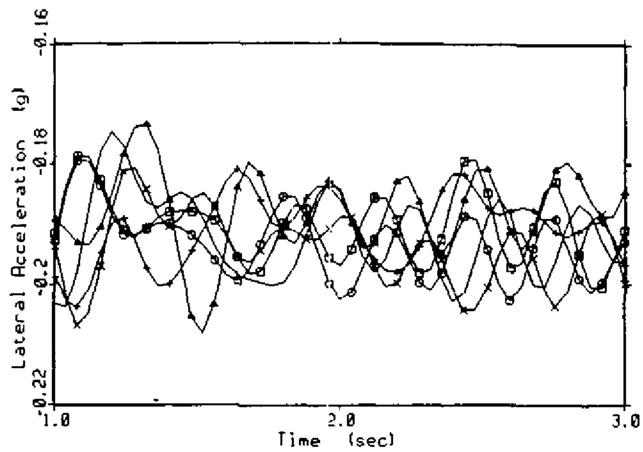

Fig. 3 - Steady state lateral acceleration from six repeat vehicle test runs (nominally a 25 mph -0.2 g lateral acceleration maneuver)

This data is from the six selected runs of a -0.2 g. lateral acceleration constant speed, J-Turn test case performed at a nominal speed of 25 mph. For each time increment, it is assumed that the distribution of the channel variables, about some population mean value, is a normal or Gaussian distribution.

Shown in Figure 4 is the mean value, computed at each time increment (every 0.04 seconds), of the six curves shown in Figure 3. The sample mean is computed by simple averaging. As can be seen, computing the mean from the six repeated runs has a "smoothing" effect on the experimental data since random spikes, caused by such things as electrical noise or road surface irregularities, become less pronounced after averaging.

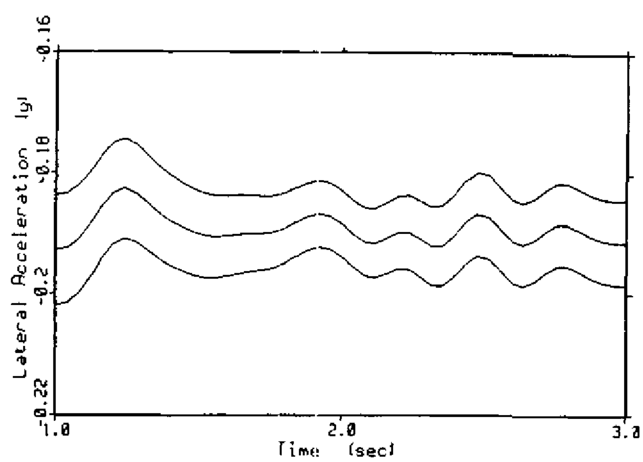

Fig 4 - Mean value and 95% confidence intervals for mean value for the six lateral acceleration curves of Figure 3

[3]Frequency response functions are similar to transfer functions except no assumption is made that the system is linear.

Since the distribution of the individual run values is assumed to be Gaussian, the sample standard deviation of the six curve values, s, is also computed at each time increment. Equation 1 shows the formula used:

$$s = \left(\frac{\sum_{i=1}^{n} (x_i - \bar{x})^2}{n - 1} \right)^{1/2} \qquad (1)$$

where the x_i values are the individual sample values, $\bar{x}$ is the sample mean value, and n is the number of samples.

From the standard deviation information, confidence intervals for the population mean values are computed. The 95 percent confidence intervals for the population means at each time increment are shown on Figure 4. These confidence intervals represent the range into which there is 95 percent confidence that the true mean values of the measured data fall. They are calculated using the equation:

$$\varepsilon = t_{\alpha/2,n-1} \cdot \frac{s}{(n)^{1/2}} \qquad (2)$$

where ε is the distance from the population mean to the upper (or lower) boundary of the confidence interval and $t_{\alpha/2,n-1}$ defines the spread of the Gaussian distribution (the t-distribution is to be used to approximate a Gaussian distribution for small sample sizes) about its mean value. Available t-distribution tables give values for $t_{\alpha/2,n-1}$ as a function of n for selected $100*(1-\alpha)$ percent confidence intervals.

For data that is being analyzed in the frequency domain, similar ensemble averaging calculations of experimental mean values and 95 percent confidence intervals are made at each frequency interval of 0.01 Hz.

At the completion of ensemble averaging, the experimental data has been processed to the point where it can be used for simulation validation. Two of the ensembled averaged channels, the front brake line pressure and the handwheel angle, along with the average value for the initial vehicle speed, are used as inputs to the simulations.

SIMULATION DATA PROCESSING

Figure 5 is a block diagram of the simulation data processing portion of Figure 1 showing this work as it was actually performed for the four test vehicles.

The first stage in the simulation data processing is vehicle parameter measurement. All vehicle simulations must be supplied with physical system parameters that describe the particular vehicle being simulated. These parameters vary from simulation to simulation but typically include vehicle geometric and inertial properties, vehicle

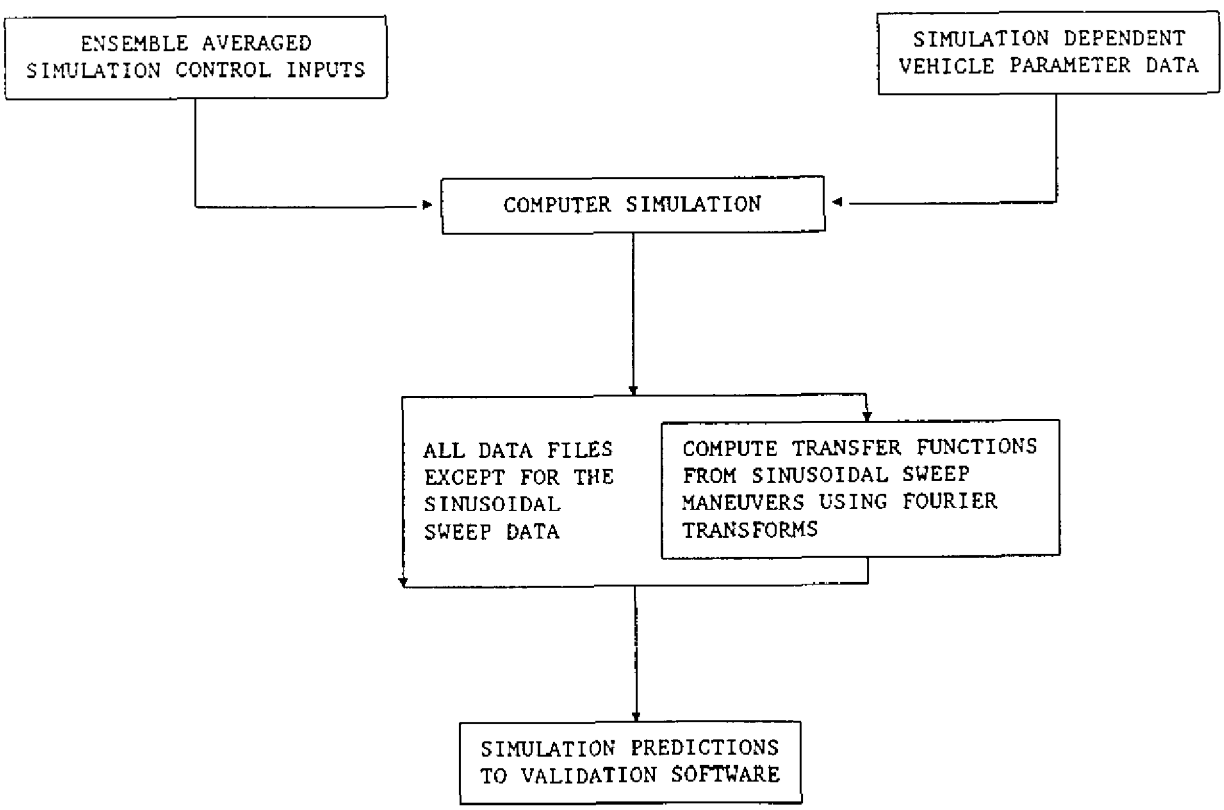

Fig. 5 - Simulation data flow

component inertial properties, suspension kinematic, damping, and stiffness properties, tire force generation properties, and brake torque generation properties.

Measuring vehicle parameters is a time consuming and sometimes difficult task. Although vehicle parameter estimation schemes are available for many parameters, it is best, especially for simulation validation work, to directly measure all parameters.

When validating a vehicle dynamics simulation, it is sometimes tempting to adjust one or more parameters to improve simulation predictions. This type of parameter adjustment is ill-advised if the goal of the validation work is truly to determine the predictive capability of the simulation. Rather, the validation process should be used to identify possible errors in the parameter measurement techniques, the measurement problems resolved, and the suspect parameters independently remeasured.

Several specialized test machines were used to measure, for each vehicle, the parameters required to describe the vehicles to the two simulations. NHTSA's Inertia Parameter Measurement Device (8) and Small Parts Inertia Rig were used to measure the inertial properties of each entire vehicle and of selected vehicle components, respectively. The Suspension Parameter Measurement Device (9), was used to measure vehicles suspension's kinematic, compliance, and Coulomb damping parameters. Data obtained from manufacturers was used to find shock absorber properties. The vehicle's braking characteristics were measured using NHTSA's Road Transducer Plate (10). Finally, the simulation's empirical tire models are dependent on measured tire parameters, and these tire parameters were measured by the Calspan Corporation on their flat belt tire tester, TIRF (11).

As shown in Figure 5, following completion of the vehicle parameter measurement stage, the actual simulation runs were made. The inputs required by the simulations were the vehicle parameters and control inputs from the experimental program. To minimize the run time and data handling effort, the control inputs were ensemble averaged before being feed into each simulation. Only the mean values of each input were feed into the simulations.

As has been previously mentioned, the two simulations used in this research are the IDSFC and the VDANL simulations. IDSFC runs on the VAX and using it presented no particular problems. VDANL runs on IBM PC-AT (or compatible) computers thereby requiring that simulation inputs be transferred from the VAX to the personal computer and the simulation run. The simulation output was then transferred back to the VAX where the experimental data and comparison software resides.

Both simulations were run in open-loop control mode (no driver feedback) with control inputs being read from the ensemble averaged mean value channels mentioned above. The one exception to this was when the sinusoidal sweep steering maneuvers were performed. It was decided that mathematically generated control inputs would be feed into the simulation instead of using experimental time histories. The frequency response curves generated from these mathematically generated inputs show no appreciable differences when compared with the curves produced from the experimental inputs. This was done to save time and to insure that sufficient input signal power is present throughout the input frequency range of interest.

The data reduction stage of the simulation data processing was mainly performed via modifications that were made to the actual simulation codes. The problems here were:

1. The actual in-vehicle accelerometers were unstabilized (i.e., vehicle pitch and roll affected their outputs) while both simulations output stabilized sprung mass accelerations.

2. The actual in-vehicle accelerometers were located at each vehicle's center of gravity while the VDANL simulation assumes that the accelerometers are mounted at each vehicle's roll axis directly below the center of gravity.

The equations relating the measured accelerations to the original simulation output channels were derived and implemented as modifications to the two simulations thereby providing output sprung mass acceleration channels that were directly comparable with the in-vehicle transducer measurements.

For the sinusoidal sweep steering maneuvers, one other data reduction operation was performed; computation of the simulated frequency response functions from time domain simulation output.

Since the simulations were driven by the mean of the ensemble averaged control inputs, no further ensemble averaging was necessary. At this point, the simulation predictions are now ready for comparison with the reduced and ensembled experimental data.

QUALITATIVE SIMULATION VALIDATION

The qualitative validation methods discussed here use graphs overlaying results from both simulations (VDANL and IDSFC) and experimental data to determine simulation validity. Graphs are prepared showing simulation predictions overlaid with the mean and 95 percent confidence level of the mean of the experimental data for the channel to be compared. The accuracy of the simulation predictions are then compared visually with the experimental data. This is done for both time and frequency domain data.

The following discussion is presented to provide a few examples of the types of the topics that should be considered during simulation validation or simulation development.

As stated previously, for a simulation to be considered valid, it must predict, to the desired level of accuracy, both the steady state and transient responses of the test vehicle. The first priority in validating a simulation is to check its ability to predict the vehicle's basic steady state gains and transient behavior during simple vehicle maneuvers. Once this has been satisfactorily completed, a simulation needs to be checked against more complex maneuvers, in order to duplicate "real world" driving scenarios (lane change, brake in a turn, etc.). Both time and frequency domain data will be used to "see" the full range of vehicle responses.

The first area addressed is the simulation's ability to predict the vehicle's steady state response over the desired operating range. The lateral and longitudinal accelerations, yaw rate and body roll angle steady state values are the simulation outputs studied in this research project. Each of these vehicle responses depends on a number of vehicle characteristics and therefore can lend insight into modeling and/or parameter specification problems. For example, in order for lateral acceleration to be accurately predicted, tire side force coefficients, steering system gain and compliances, front to rear roll stiffness distribution and suspension kinematics and compliances all must be accurately modeled and have accurate parameters supplied to the simulation. Longitudinal acceleration gives a good view of the front to rear brake proportioning, and for limit straight line braking maneuvers, the longitudinal tire model. The roll angle prediction shows the accuracy of the vehicle roll stiffness measurement and the simulation's roll moment calculation.

The graphs of constant speed J-Turn maneuvers provide a good way to check the steady state values of vehicle response. Figure 6 shows the vehicle's lateral acceleration at its center of gravity and sprung mass roll angle. Comparison of the simulation output with the experimental data should be made after the vehicle transient behavior has died out, in this case after approximately three seconds. Visual inspection of the lateral acceleration graph shows the VDANL simulation to be doing an excellent job for this vehicle and maneuver. The IDSFC steady state value is slightly low for this test, although still doing a good job of predicting steady state lateral acceleration.

The graph of sprung mass roll angle in Figure 6 shows the predictions of both simulations to have considerable errors. Since the vehicle's roll moment is driven by lateral acceleration, which has been predicted accurately, and the roll stiffness values for both simulations were computed from the same vehicle measurement data, the roll moment modeling of both simulations appears to be in error. (Note that the vehicle fixed axis lateral acceleration is dependent on roll angle but not to a large extent, as the measured roll angle is only about three degrees.)

When validating a simulation's ability to predict vehicle behavior, it is important to avoid chasing "cross talk" between vehicle responses. That is, the value of a particular response may be strongly influenced by the value of other responses. For example, as discussed previously, the force input for the sprung mass roll mode is lateral acceleration. If the simulation is not doing a good job with lateral acceleration in a particular frequency range, this will directly effect the roll dynamics. Therefore, there is no point looking at roll mode correlation until the lateral acceleration problems are resolved.

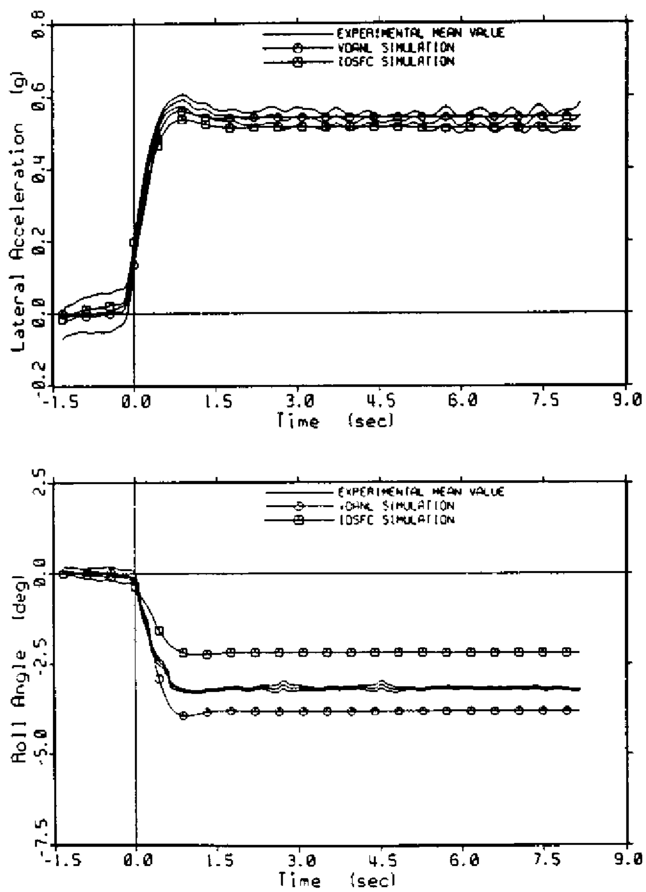

Fig. 6 - Lateral acceleration at vehicle C.G. and sprung mass roll angle for a 50 mph constant speed J-turn (1987 Hyundai Excel)

This same approach of visually comparing the time history graphs can be carried out for longitudinal acceleration and/or other channels of interest. It is important that the comparison be made over the entire operating range of interest. A simulation may show good agreement in the vehicle's linear operating range (below 0.3 g's) but have problems for the limit maneuver case. This suggests more work is necessary in the modeling or parameter measurement of the tire and/or suspension nonlinearities.

Once the steady state responses are being accurately predicted, the simulation's performance predicting the vehicle's transient responses can be checked. A good view of the simulation's ability to predict the vehicle's transient behavior is given by plotting frequency response curves. Figure 7 shows the lateral acceleration frequency response (magnitude and phase angle) to steering wheel angle inputs for a 50 mph sinusoidal sweep steering maneuver. When analyzing the frequency domain graphs, it is helpful to divide the frequency range into low, middle and high frequency windows to better understand possible causes of simulation problems.

The low frequency response data, below approximately 0.5 Hz (3.14 rad/sec) for most light vehicles, is influenced primarily by the vehicle's

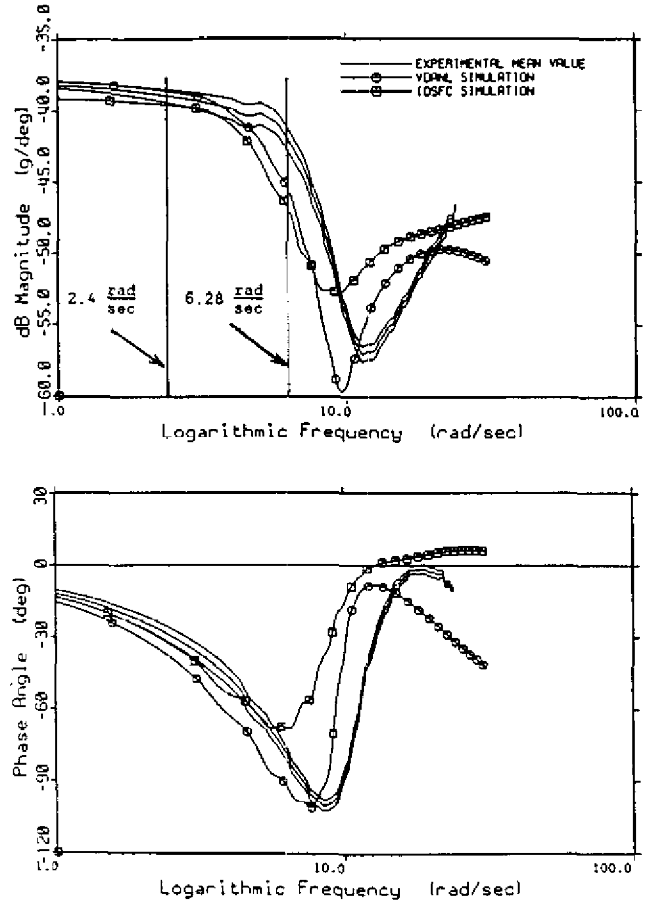

Fig. 7 - Lateral acceleration frequency response (magnitude and phase) for a 50 mph sinusoidal sweep steering maneuver (1987 Hyundai Excel)

steady state response. In fact, for the graphs presented, the 0 Hz data is taken from a constant speed J-Turn test. Data in this range can be treated in the same way the constant speed J-Turn test data was in the previous discussion. The middle frequency response data, between approximately 0.5 and 1.5 Hz (9.42 rad/sec), depends upon the basic sprung mass dynamics along with the low frequency effects of the steering system dynamics (12,13). The high frequency response data, above approximately 1.5 Hz, is influenced by the lag of the tire forces and steering dynamics.

The dynamic characteristics of road vehicles are very complicated with complex interactions between each of the vehicle subsystems (steering, tires, sprung mass/suspension system). Trying to find sources of discrepancy between simulation and experiment by looking at the vehicle as one unit can become very difficult. One way to simplify this problem is to look at each vehicle subsystem individually before combining them to see the complete vehicle response characteristics. As an example, when studying yaw rate to handwheel angle frequency response, the effects of the steering system can be eliminated by first looking at the yaw rate to roadwheel angle frequency response. Then, to study the steering system separately study the roadwheel angle to handwheel angle frequency response. This approach was shown by Smith (12) and Segel (13).

Once the steady state and frequency response predictions of the simulation have been validated, the simulation can be checked against maneuvers designed to simulate "real world" driving conditions. The following discussion will use a lane change maneuver as an example, although other maneuvers could be substituted. This example points out the problems of trying to infer validity of a simulation at one operating condition from its performance at another.

Figure 8 shows vehicle lateral acceleration at its center of gravity for a 50 mph lane change maneuver. As can be seen, both simulations do a good job of predicting lateral acceleration for this maneuver. The predominant input handwheel frequency for this test is approximately 2.4 rad/sec (0.4 Hz). This lane change is, therefore, a low frequency maneuver. As was previously pointed out, the ability to simulate vehicle response in this frequency range is influenced primarily by the steady state predictions. Looking at the steady state predictions in Figure 6 and the frequency response predictions at this frequency in Figure 7, it follows that the simulation predictions for this lane change maneuver should be

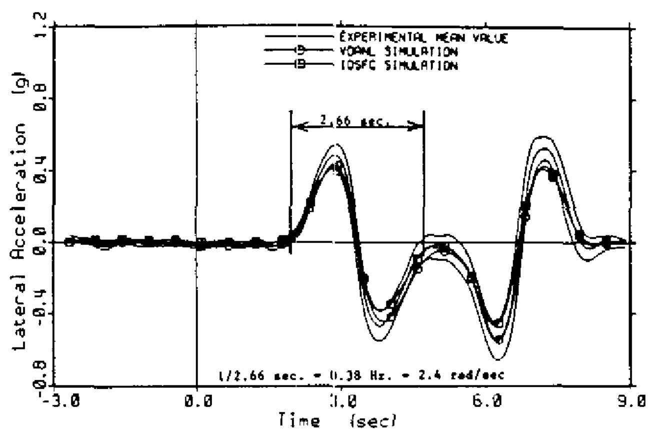

Fig. 8 - Lateral Acceleration at vehicle C.G. for a 50 mph right lane change maneuver (1987 Hyundai Excel)

good. The lane change maneuver could have been designed differently and required a handwheel input frequency of 6.28 rad/sec (1 Hz). Looking at Figure 7 at this higher frequency shows that the simulated response would not agree nearly as well as at the lower frequency. For a simulation to be valid for crash avoidance research (as is desire here), it should simulate, with reasonable accuracy, maneuvers that are likely to occur in "real world" crash avoidance situations. This means that the simulation should accurately predict vehicle frequency response for any input that actual drivers can generate.

Figure 9 shows the Hyundai's handwheel input and yaw rate response for a constant speed (50 mph) J-Turn maneuver. These tests are "pseudo-step steer" tests where the driver, after achieving constant speed straight running, turns the handwheel rapidly against a preset steer-stop thus simulating a step input. However, it is possible to supply a true step steer as input to the simulations. Supplying simulations with true steering step inputs excite higher frequencies than those that are physically excited during a rapid step-type steering maneuver. Therefore, a true step steer input to the simulation may produce simulation predictions that are not comparable with the experimental results. This is why it is imperative to drive the simulations with inputs analogous to those used during the experimental testing phase of the validation process.

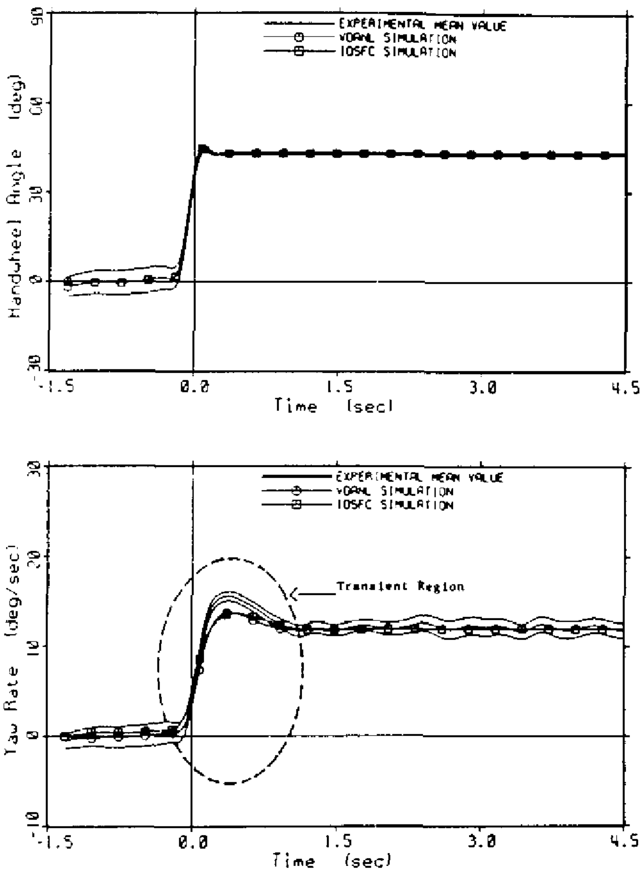

Fig. 9 - Handwheel angle input and yaw rate response for a constant speed J-turn maneuver 50 mph (1987 Hyundai Excel)

Figure 9 indicates that both simulations appear to do a reasonable job of predicting yaw response for this vehicle and this maneuver. The simulations' steady state responses are excellent, while examination of the transient region reveals discrepancies. Examination of the Hyundai's yaw rate frequency response curves (Figure 10) detail the differences in the predicted and measured yaw

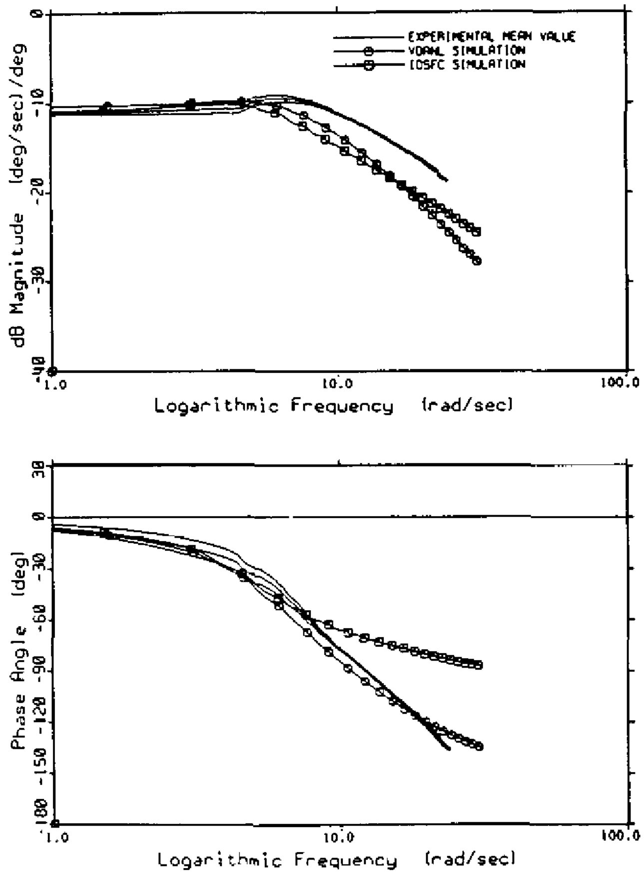

Fig. 10 - Yaw rate frequency response (magnitude and phase) for a 50 mph sinusoidal sweep maneuver (1987 Hyundai Excel)

rate response. The frequency where the peak magnitude in the frequency response occurs, and the magnitude of the peak (which is predominantly governed by effective yaw damping), highlight the discrepancies that were not as apparent when examining the time domain responses. Both simulations predict low values for the peak yaw rate frequency and both simulations exhibit more damping than is experimentally measured.

The yaw rate frequency responses at 50 mph are shown in Figures 11, 12, and 13 for the other three vehicles tested. Figure 11 shows the yaw rate frequency response for the Ford Thunderbird, Figure 12 for the Ford E-150 Van, and Figure 13 for the Suzuki Samurai. Notice that the Suzuki Samurai (Figure 13) shows a very large peak in the yaw rate magnitude at the yaw rate peak frequency. If this peak, both magnitude and frequency, is not predicted properly, then the transient response of the vehicle cannot be simulated accurately.

Too often, researchers have overlooked differences in the transient region of vehicle response by only examining time domain data. Errors in the transient region of time response results may be small for some maneuvers, but large for others, depending on the shape and speed of the steering (or other) input. The yaw rate frequency response curves for the four vehicles reveal

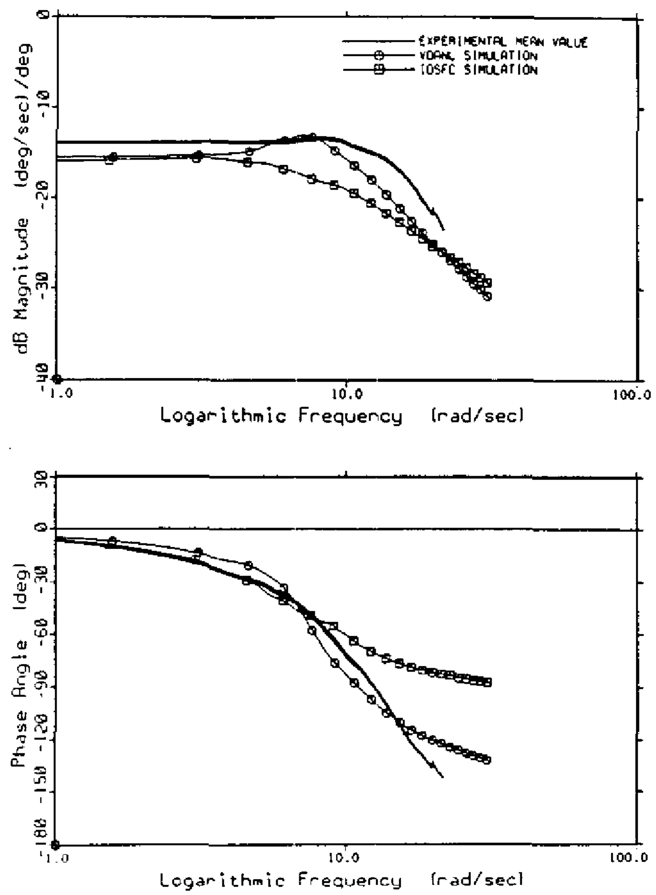

Fig. 11 - Yaw rate frequency response (magnitude and phase) for a 50 mph sinusoidal sweep maneuver (1987 Ford Thunderbird)

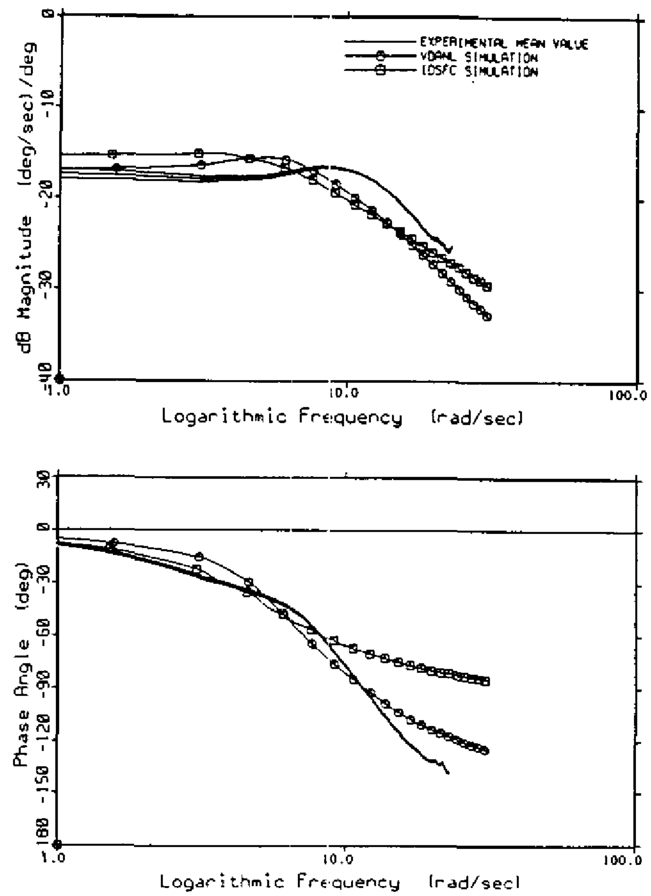

Fig. 12 - Yaw rate frequency response (magnitude and phase) for a 50 mph sinusoidal sweep maneuver (1987 Ford E-150 Van)

discrepancies between the simulation predictions and experimental results. These discrepancies could be difficult to recognize using only time response comparisons. This illustrates again why frequency response curves should be generated, since they provide a great deal of information about system behavior. It also shows the need to study several different types and severities of vehicle maneuvers, and, if possible, several different vehicles, before claiming full-fledged simulation validity.

QUANTITATIVE SIMULATION VALIDATION

To supplement the qualitative validation methods presented, quantitative validation methods provide significant insight into the vehicle modeling process. The method presented here computes values, called metrics, from simulation output and experimental data for direct comparison. Besides aiding the validation process, metrics also can be used to aid in quantifying vehicle performance by providing direct measures of the characteristic vehicle responses. Metrics are obtained from both time and frequency domain data and are helpful, like the qualitative method, in locating and differentiating both parameter and modeling problems.

Quantitative measures are useful when trying to determine the origin of disagreements between simulation and experimental data. In this case it is very helpful to look at data for a variety of vehicles and a variety of maneuvers. When studying many different vehicles/maneuvers, quantitative measures may provide easier to interpret results than a large number of graphs. These measures can show if problems are isolated, random or systematic. Isolated errors, those that occur for only one vehicle, can usually be traced to parameter problems. Random and systematic type errors can be the result of parameter and/or modeling problems. The source of these types of errors are often very difficult to find and require careful examination of results for a variety of vehicles and maneuvers.

The metrics suggested here can all be measured from graphs prepared in the same manner as during the qualitative validation. Alternatively, computer programs can be written to compute metrics from both simulation output and experimental data. This second method is preferable if the validation project is extensive. The metrics suggested here are just a few of many that can be devised. The results presented are for vehicle yaw rate response. These are representative results and by no means encompass a complete regime of validation

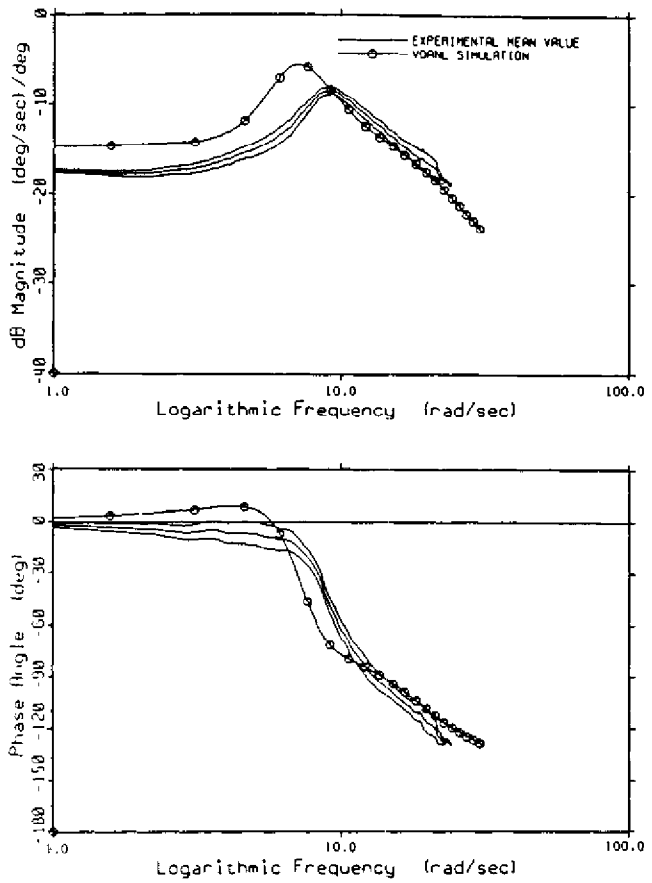

Fig. 13 - **Yaw rate frequency response (magnitude and phase) for a 50 mph sinusoidal sweep maneuver (1988 Suzuki Samurai)**

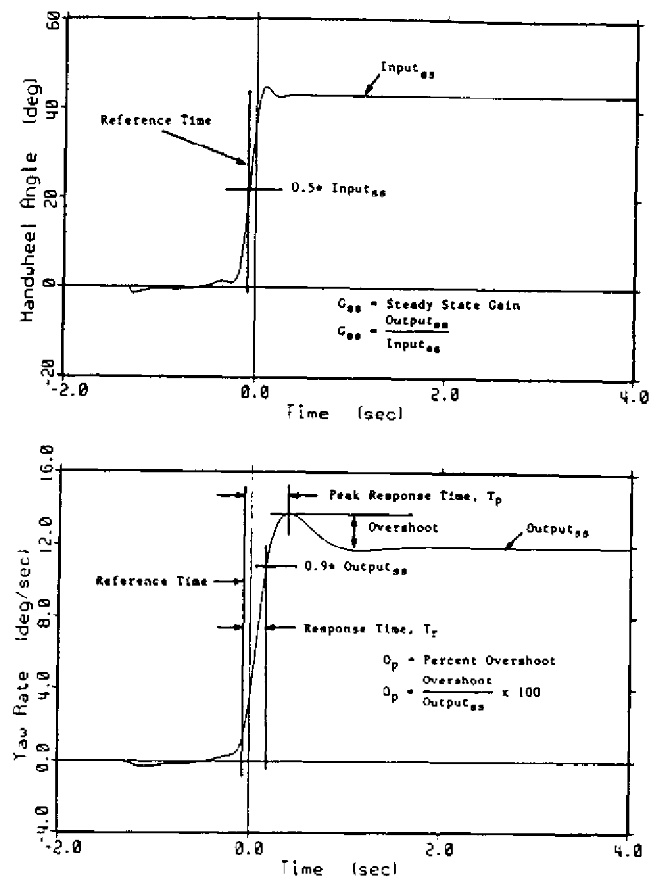

Fig. 14 - Handwheel angle and yaw rate from a simulated 50 mph constant speed J-turn maneuver

measures. Other vehicle response quantities, such as lateral acceleration and roll angle, also provide important measures of simulation performance and are of major interest in studies of vehicle handling and crash avoidance research. When starting a validation program, the intended use of the simulation should be kept in mind and be used to drive the metric selection process.

TIME DOMAIN METRICS - The first set of metrics are derived from the time domain data of "pseudo-step steer", constant speed, J-turn maneuvers. Steady state gain, response time, peak response time and percent overshoot are standard time domain performance specifications (14). These specifications have been presented and discussed previously, as they relate to vehicle stability, by Nisonger and Fancher (15). These metrics can be computed for any of the experimental and simulated vehicle response channels with the selection depending on the simulation's intended application. For this program, lateral acceleration, yaw rate and sprung mass roll angle were the channels chosen.

Figure 14, which will be used in the discussion of time domain metrics, shows typical simulation time history plots for vehicle input and output, handwheel steering angle and yaw rate, respectively, in this example. The first time domain metric, steady state gain, G_{ss}, is defined as the steady state value of the output (yaw rate) divided by the steady state value of the input (handwheel steering angle). For this research, the simulated and experimental steady state gains are analogous to the steady state values of the response, since the simulations were driven with the experimentally measured handwheel angle inputs. In general this may not be the case, so computing the steady state gain provides a means of normalizing the results. Computing gains as a function of the input also allows for a more direct comparison among maneuvers run at different severity levels; that is, different input levels.

As discussed in the qualitative procedure, the utility of the steady state gain metric is to check basic chassis, suspension, steering and tire parameters and show simulation performance for "low frequency" maneuvers.

Table 1 shows the yaw rate to handwheel angle steady state gains for all four vehicles performing a constant speed J-turn at a nominal lateral acceleration level of 0.4 g. Mean values, and corresponding 95% confidence intervals on the mean values, are presented for the experimentally measured gains, as are both the VDANL and IDSFC simulation predicted gain values. Two different test speeds, 25 mph and 50 mph, are shown.

Table 1 illustrates some of the utility of quantitative validation. First notice that both simulations are adequately predicting the yaw rate steady state gains for the Hyundai Excel. This is an indication that both simulations may have suitable models for predicting this steady state quantity.

The gains for the Ford Thunderbird are low for both simulations and at both speeds. Also, both simulations predict similar values. This suggests that a parameter common to both simulations is likely in error. (In this case, the parameter was the steering system compliance. The compliance value used was measured without the power steering operational. Using a revised steering system compliance, measured with the power steering operating, will improve the simulated yaw rate gain values for the Thunderbird.)

The IDSFC simulation satisfactorily predicts the yaw rate steady state gains for the Ford E-150 Van, while the VDANL simulation shows substantial discrepancies at both speeds shown on Table 1. This indicates that a parameter unique to the VDANL simulation may be in error. The cause of these discrepancies is not yet known, however, thorough examination of the likely parameters should reveal an inaccurately measured, computed, or recorded parameter.

The measured yaw rate gains for the Suzuki Samurai show a trend opposite to the other three vehicles. That is, the yaw rate gain decreases with vehicle speed. Analysis of a two degree of freedom, linear vehicle model shows that the steady state yaw rate gain is a nonlinear function of vehicle understeer, and that the yaw rate gain can increase or decrease with vehicle speed depending on the value of the vehicle's understeer (16). As seen on Table 1, the VDANL simulated values do not reflect this trend. (The IDSFC simulation does not model vehicles, such as the Samurai, with solid front axles.) The reasons for this discrepancy are not known. Further analysis could reveal parametric problems or an inadequacy in the steering or suspension system models for vehicles with solid front axles.

The second set of time domain metrics, response time, T_r, and peak response time, T_p, are speed-of-response criteria. They provide an indication of vehicle stability and hence controllability. Increased response time (sluggish vehicle response), as maneuver severity increases, suggests decreased vehicle stability that may indicate a vehicle will be difficult to control in an accident avoidance maneuver (high severity) (15).

Response time and peak response time are defined relative to a "reference" time, as defined in (15), corresponding to the time where the input reaches 50 percent of its steady state level. This is used since the input is not a pure step or ramp and therefore does not have an easily measured starting or ending time. Response time is defined here as the time from the reference time to the time when the vehicle output reaches 90 percent of its steady state value. Peak response time, which can only be defined for responses that exhibit some overshoot, is defined as the time from the reference time to the time when the vehicle output reaches the maximum value of its first peak.

Table 2 provides yaw rate response time and peak response time values for the Hyundai Excel performing 50 mph J-turn maneuvers. Experimental and simulated values are given for three different lateral accelerations. Both response time metrics show that the vehicle is more responsive than both simulations predict. This trend was found for all four vehicles tested. This arises from the fact that both of these simulations do not accurately predict the frequency bandwidth of the actual vehicles. More will be said about this during the discussion of frequency domain metrics.

The final time domain metric, percent overshoot, O_p, is a relative stability criterion. Since road vehicles behave as high order systems, standard analysis techniques based on second order system behavior cannot be used to compute standard second order system response parameters, such as damping ratio, for the total vehicle system. However, percent overshoot can be used instead of damping ratio to provide a metric related to vehicle damping and thus stability. Percent overshoot for a response time history is defined as the amount of overshoot (maximum value of the response less the steady state value) divided by the steady state value multiplied by 100. For this metric, a higher value means less damping and therefore provides a good indication of a simulation's ability to predict vehicle system damping.

Table 2 shows that the yaw rate experimental percent overshoot for the Hyundai Excel increases as the maneuver severity increases. The two simulations do not follow this trend. The reasons for these discrepancies are not, at this time, known. Further research in this area may reveal the need for simulation model modifications to better simulate nonlinear dynamic effects.

FREQUENCY DOMAIN METRICS - A second group of metrics can be derived from the frequency domain data. Figure 15 shows a simulated yaw rate frequency response (magnitude and phase angle) to handwheel angle for a 50 mph sinusoidal sweep steer maneuver. Frequency domain metrics of importantance for this research program include peak frequency, bandwidth and peak amplitude ratio. These metrics will be computed for yaw rate, lateral acceleration and roll angle frequency response data from both 25 and 50 mph tests. Again, as with the time domain metrics, a road vehicle is a high order system and the metrics computed are not standard second order system parameters. They are merely a means to quantify the vehicle system and provide a method to directly compare the simulation predictions with the experimental data.

The first frequency domain metric, peak frequency, is defined as the frequency at which the peak occurs in the frequency response magnitude. It is easily recognized in the magnitude plot of Figure 15. However, for other vehicle outputs or other vehicles, specifically those responses that are overdamped, no peak, above the steady state magnitude, in the frequency response will occur.

A second frequency domain metric is bandwidth. The traditional definition, the frequency at which

TABLE 1 -- Yaw Rate to Handwheel Angle Steady State Gains Nominal +0.4 g. Lateral Acceleration J-Turn Maneuver

YAW RATE GAINS
(deg/sec)/deg

25 MPH

Vehicle	Experimental Mean and C.I.	IDSFC Simulation	VDANL Simulation
1987 Hyundai Excel	0.217 ± 0.002	0.203	0.211
1987 Ford Thunderbird	0.202 ± 0.004	0.153	0.162
1987 Ford E-150 Van	0.126 ± 0.002	0.121	0.111
1988 Suzuki Samurai	0.153 ± 0.002	**	0.167

50 MPH

Vehicle	Experimental mean and C.I.	IDSFC Simulation	VDANL Simulation
1987 Hyundai Excel	0.290 ± 0.004	0.271	0.284
1987 Ford Thunderbird	0.223 ± 0.011	0.155	0.156
1987 Ford E-150 Van	0.166 ± 0.004	0.164	0.132
1988 Suzuki Samurai	0.125 ± 0.003	**	0.170

** The IDSFC simulation does not model vehicles with solid front axles.

TABLE 2 -- Yaw Rate Response Time, Peak Response Time, and Percent Overshoot - 1987 Hyundai Excel - 50 mph J-Turn

Yaw Rate Time Domain Metrics
Hyundai Excel
50 mph J-Turn

		Response Time (sec)		
Lateral Acceleration	Handwheel Angle	Experimental Value	IDSFC Simulation	VDANL Simulation
+0.2 g.	12.7°	0.22	0.23	0.24
+0.4 g.	31.6°	0.18	0.20	0.22
+0.6 g.	43.3°	0.17	0.21	0.22

		Peak Response Time (sec)		
Lateral Acceleration	Handwheel Angle	Experimental Value	IDSFC Simulation	VDANL Simulation
+0.2 g.	12.7°	0.38	0.52	0.44
+0.4 g.	31.6°	0.37	0.51	0.44
+0.6 g.	43.3°	0.43	0.52	0.43

		Percent Overshoot (%)		
Lateral Acceleration	Handwheel Angle	Experimental Value	IDSFC Simulation	VDANL Simulation
+0.2 g.	12.7°	12.5	13.8	12.5
+0.4 g.	31.6°	20.0	17.4	12.1
+0.6 g.	43.3°	30.0	15.8	15.2

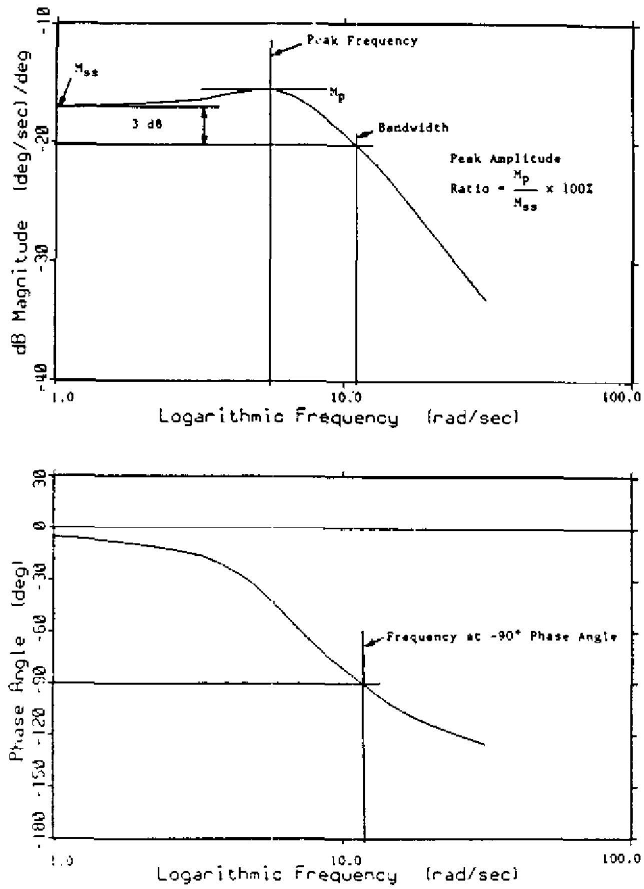

Fig. 15 - **Yaw rate frequency response from a simulated 50 mph sinusoidal sweep steering maneuver**

the magnitude drops 3 dB below its steady state magnitude, is being used for this research. The bandwidth metric is a system speed-of-response type measure where a wider bandwidth means that the vehicle response characteristics will be maintained to a higher input frequency. For the case of yaw rate frequency response to handwheel angle, this metric gives the frequency where the vehicle starts becoming less responsive. Bandwidth can be determined (measured) even for responses that are overdamped and have no peak, above the steady state magnitude, in their frequency response magnitude.

Table 3 shows experimental and simulated peak frequency and bandwidth values for the yaw rate response of all four vehicles tested. These frequency response results were generated from 50 mph sinusoidal sweep steering maneuvers (Results from *Figure 10 through Figure 13*). Both simulations predict low values for both peak frequency and bandwidth. The simulated responses begin to attenuate at much lower frequencies than the actual vehicles. This means that the simulations will predict sluggish vehicle response since most steering inputs will, to some extent, excite frequencies in the range between the simulated and experimental bandwidths. This result, that the vehicle is more responsive than the simulations predict, is in accord with the time domain response time metrics presented in Table 2.

The IDSFC simulation predictions for yaw rate peak frequency and bandwidth are worse than the VDANL predictions. This is because the IDSFC simulation has too much effective yaw damping. Research has indicated that tire dynamics play a very important role in determining the yaw damping of the vehicle. Since the VDANL simulation does include a first order tire dynamics lag model, its damped frequency response is generally better than the IDSFC simulation's.

The next frequency domain metric, peak amplitude ratio, is the ratio of the peak frequency magnitude to the steady state magnitude. This gives a measure, similar to percent overshoot from the time domain data, of vehicle damping, with a higher value implying less damping. The experimental and simulated peak amplitude ratios for the four vehicles are shown on Table 3. Based on the peak amplitude ratios, the IDSFC simulation predicts more damping than the experimental results for all three vehicles for which it can be used. The VDANL simulation, with the exception of the Hyundai Excel, predicts a less damped yaw rate response than was experimentally measured.

As is the case for peak frequency and bandwidth, the yaw rate peak amplitude ratio is effected by tire dynamics. Steering system dynamics and hysteresis, which are not currently modeled in either the IDSFC or VDANL simulation, also influence frequency response in the frequency range studied in this research.

Good simulation validity in the transient region is very important when using a simulation for crash avoidance research. For example, if a vehicle has a high yaw rate peak amplitude ratio, it may become very difficult to control if excited near its peak frequency. This situation can arise in a rapid lane change maneuver and lead to loss of control. If the simulation predictions are incorrect for the peak frequency or the peak amplitude ratio, predictions for complex maneuvers that excite frequencies near the peak frequency will be wrong.

Another frequency domain attribute that should be considered is the frequency response phase angle. A useful metric here is the input frequency at which the phase angle reaches some specified value. However, a general metric useful for all frequency responses is difficult to define because the overall phase shift for a given frequency response depends on the input and output quantities being studied. For example, if a given frequency response is known to exhibit second-order like characteristics (ie, a phase shift of -180°), then a useful phase angle metric would be the frequency at which the phase angle is -90°.

Table 3 shows, for all four vehicle at 50 mph, the frequencies at which the yaw rate frequency response phase angles reach a phase shift of -90°. Although the experimental phase angles likely extend below -180° at frequencies above those studied during this research (see Figures 10 - 13), the -90° phase shift quantity appears to be an *appropriate measure in this case. The -90° phase* shift frequencies for the VDANL simulation are in fairly good agreement with the measured results. The VDANL values are less than, or equal to, the

TABLE 3 -- Frequency Domain Metrics

Yaw Rate Frequency Domain Metrics
50 mph Frequency Response

Peak Frequency (Hz)

Vehicle	Experimental Value	IDSFC Simulation	VDANL Simulation
1987 Hyundai Excel	1.01	0.62	0.66
1987 Ford Thunderbird	1.31	0.54	1.13
1987 Ford E-150 Van	1.33	0.54	0.84
1988 Suzuki Samurai	1.45	**	1.08

Bandwidth (Hz)

Vehicle	Experimental Value	IDSFC Simulation	VDANL Simulation
1987 Hyundai Excel	2.40	1.46	1.53
1987 Ford Thunderbird	2.38	1.59	2.02
1987 Ford E-150 Van	2.38	1.24	1.69
1988 Suzuki Samurai	4.53	**	3.03

Peak Amplitude Ratio

Vehicle	Experimental Value	IDSFC Simulation	VDANL Simulation
1987 Hyundai Excel	1.20	1.14	1.06
1987 Ford Thunderbird	1.07	1.05	1.34
1987 Ford E-150 Van	1.10	1.04	1.18
1988 Suzuki Samurai	2.81	**	3.15

Frequency of -90° Phase Angle (Hz)

Vehicle	Experimental Value	IDSFC Simulation	VDANL Simulation
1987 Hyundai Excel	1.96	None	1.72
1987 Ford Thunderbird	1.93	None	1.69
1987 Ford E-150 Van	1.79	None	1.79
1988 Suzuki Samurai	2.05	**	1.64

** The IDSFC simulation does not model vehicles with solid front axles.

experimental values. This is expected since the VDANL simulation predicts lower peak frequencies and bandwidths than are experimentally measured. The IDSFC simulation phase angle never goes to -90° in the frequency range studied in this research. This is because the IDSFC simulation lacks tire force and moment lag dynamics.

For linear systems, both magnitude (plotted using a logarithmic frequency and dB magnitude scales) and phase angle approach straight-line asymptotes at high frequencies. The apparent magnitude and phase angle asymptotes of the curves shown on Figures 10 through 13 reveal some interesting characteristics of the simulations. Qualitatively, the VDANL simulation exhibits much better frequency response "shapes", when compared with the experimental curves, than does the IDSFC simulation. Both VDANL high frequency asymptotes agree with the experimental results better than the IDSFC values.

The VDANL simulation has a magnitude that is asymptotic to -40 dB/decade and a phase angle that is asymptotic to -180°. The IDSFC simulation has asymptotes of -20 dB/decade for the magnitude and -90° for the phase angle. This behavior can be shown, mathematically, to be a result of the first-order tire dynamics that are modeled in the VDANL simulation. The inability of the IDSFC simulation to reasonably predict the high frequency asymptotes reveals a definite model inadequacy, specifically a lack of tire dynamics.

Notice also on Figure 10 that the experimental high frequency asymptotes do not match the VDANL simulation's exactly. Several possible causes for these differences are currently being studied. One possibility is that actual tire lag dynamics exhibit second-order, not first-order, characteristics (17,18). Steering system dynamics also likely effect the high frequency asymptotes. Further research is needed to determine the actual modeling modifications necessary to provide accurate simulation predictions, for all four vehicles, throughout the handling frequency range as specified by the experimental frequency response curves.

OTHER QUANTITATIVE METRICS - An additional quantitative method being investigated is the computation of the root mean square (RMS) of the difference between the experimental and predicted means over the entire test and/or sections of a test. This RMS error computation is useful when checking simulation predictions for complex crash avoidance maneuvers such as a lane change. These closed loop maneuvers do not lend themselves to the previously discussed quantitative validation method and some insight may be gained by looking at the discrepancy between the test data and simulation for the entire test. It should be kept in mind that RMS error used in this way will not differentiate between steady state and transient simulation problems. A possible solution or supplement would be to compute the RMS error for several different tests, each with an emphasis on a different aspect of vehicle behavior.

A final proposed method is computing the percent of time the simulation predictions are within the confidence limits of the experimental data. This computation is similar to RMS error and has the same limitations. Given that experimental test data contains some random error, by running repeated tests and computing the confidence level of the channel mean value, experimental data can only be said to be in this range. Therefore, if the simulation prediction is within this range, it is considered valid to the range of the experimental error.

These final two methods will have limited use in the validation of vehicle stability and control simulations and will need to be used along with the metric comparison and qualitative methods. For the validation of other types of simulations however, the RMS error and percent time within the confidence interval calculations may prove to be very powerful validation tools.

CONCLUSIONS

This paper presents a methodology for the validation of vehicle dynamics simulations by comparing simulation predictions with experimental data. A general definition of simulation validation is given followed by a presentation of the steps involved in the processing of experimental and simulation data. The use of both qualitative and quantitative comparison methods are presented.

A method of using repeated experimental runs to increase the confidence of a simulation validation procedure has been presented. The method, called ensemble averaging, involves computing mean values, and confidence limits on the mean values, for experimental data in both the time and frequency domains. Ensemble averaging the results from repeated runs has a smoothing effect on the data by reducing the random error in the measured signal. Ensemble averaging also acts to safeguard against isolated measurement errors associated with data collection problems, transducer malfunction, etc. Errors of this type can go undetected when just a single experimental run is made. Computing confidence intervals provides a measure of experimental repeatability. The validation methodology presented also reveals that using several different vehicles as well as an assortment of maneuver types and maneuver severity levels in the validation process is advantageous.

The benefits of analyzing experimental and simulated vehicle behavior in the frequency domain are revealed by way of examples. The frequency domain results provide a great deal of information, some of which may not be revealed through time domain analysis. The importance of validating simulation behavior in the transient response regime is identified. Transient response characteristics are examined using comparisons in both the time and frequency domains.

Simulation predictions are qualitatively compared with experimental results in both the time and frequency domains by plotting simulation predictions, experimental mean values, and experimental confidence intervals on the same graphs. In addition to the qualitative method, quantitative validation methods are suggested and results presented. Comparing both time and frequency domain metrics such as steady state gain, response time, percent overshoot, peak frequency and peak amplitude ratio is discussed. Also presented as a means of quantitatively validating a simulation are the computation of the RMS error and percent time the simulation output is within the experimental confidence interval.

Several conclusions can result from this simulation validation process. A simulation may be deemed valid for predicting the particular physical system behavior for which it was designed to model. The validation process may be used to select the best simulation, from a group of more than one, for a specific research project. The strengths and weaknesses of the individual simulations can be identified and compared to choose the most appropriate simulation. If the simulation validation process is used to check the validity of a complicated simulation, most likely certain aspects of the simulation will result in good predictions while other aspects will not yield accurate predictions. In these cases, the simulation validation methodology presented provides a valuable tool for modifying and enhancing a simulation. Areas of simulation disagreement with experimental results can be recognized. Also, possible vehicle parameter specification errors and/or experimental data offset or calibration errors may become apparent during the validation process.

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