Topsy - The Heitz Chassis Lab Parameter Measurement System

Edward J. Heitzman and Edward F. Heitzman
Heitz Chassis Lab, Inc.
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HISTORY

Topsy is not an acronym: it is named after the little slave girl in *Uncle Tom's Cabin.* Like its namesake, Topsy "was never born, but just grew". Topsy started out as a facility for accurate measurement of center of gravity height; then its use was expanded to include moments of inertia, bump steer, lateral acceleration weight transfer, etc.

Over a period of some 20 years Topsy has grown in two ways: first in response to or anticipation of customer test requests; and second, by thinking in each test program how it could be done better. The growth process is naturally modular, with three results:

1. All measurements are simple and direct.
2. Individual components or assemblies are easily changed.
3. New capabilities can be added without extensive redesign.

The methodology for measurement of CG height, roll moment of inertia, and roll/yaw product have evolved from the methods used by Cornell Aeronautical Laboratory in measurements on a 1953 Buick, as described in unpublished memoranda written in 1956. Compliance methodologies were inspired by the "Nedly-Wilson" Vehicle Parameters Facility built at Chevrolet in 1971 [1]. More recently, development has been guided by SAE J1574 [2].

INTRODUCTION

At the most basic level, all inertial, kinematics, and compliance facilities are quite similar. In inertial testing the vehicle is fixed to a frame with its suspension locked in place. CG height is measured by making the frame into some sort of pendulum and obtaining CG height from the variation in pendulum restoring torques with tilt angle. Roll, pitch, and yaw inertias are measured by oscillating the frame against calibrated springs in roll, pitch, and yaw, and obtaining the inertias from the resulting resonant frequencies. For lateral compliances the vehicle is fixed to a frame, and shear forces and aligning torques are applied to the tire contact patches, while the resulting deflections are measured. For vertical compliances and kinematics the vehicle is fixed to the frame while the tire contact patches are exercised vertically, or the vehicle wheels are left on scales while the vehicle sprung mass is exercised in heave, pitch, and roll.

Every test facility does these things. The differences are in the details.

Regardless of the details, the accuracies, precisions, and methodologies should be consistent with SAE J1574, entitled "Measurement of Vehicle and Suspension Parameters: Test Equipment and Procedures".

The Chassis Parameter test Facilities at Chevrolet, NHTSA, Goodyear, Ford, and Motor Industries Research Association are all designed for standardized tests, with rapid turnaround and high productivity, in fixed permanent installations. Topsy is in all these details completely the opposite.

Topsy is a modular system, designed for customized measurement of all composite inertial, kinematic, and compliance parameters, but with more leisurely turnaround, to be used on an occasional basis, oriented toward researchers and toward insight into what is being measured and how. It provides the same data plotted in the same way as the "production facilities", but in addition it can provide specialized data that might be of interest only to a particular researcher.

To minimize the possibility of test errors all tests are run under servo control with continuous display of all data. Parameters are determined from the slopes of equations machine-fit to plotted data. Raw, partially-processed, and completely processed data are supplied to the customer in whatever format he requests. Data plotting and curve fitting can be performed by Heitz Chassis Lab or by the customer, according to economic considerations.

The test facility is organized around an "infrastructure" consisting of equipment used in many different kinds of tests. These include baseplates; vehicle locating fixtures; scales; hydraulic and pneumatic power sources; interchangeable valve assemblies and actuators; transducers; and data acquisition system.

This report will first describe the "infrastructure", and then how these components are organized with special-purpose devices for the different tests: whole-vehicle inertia; compliance and steering; kinematics; shocks and struts.

INFRASTRUCTURE

BASEPLATES - The various test devices utilize three baseplates: the laboratory floor; a vertically-mounted aluminum I-beam; and the Frame/elevator assembly.

The lab floor is concrete overlaid with industrial tile. Threaded steel inserts are installed in the concrete wherever necessary to locate fixtures. When not in use the holes are filled by flat-head stainless steel capscrews. When desired hole locations change the old holes are covered with new tile.

The 150 mm vertical I-beam is permanently attached between the floor, the cinderblock lab wall, and several ceiling joists of the adjacent electronics laboratory room. It serves as the baseplate for measuring springs, struts, and shock absorbers, and as the reaction member in sidepull testing.

All inertial, kinematics and compliance measurements are made on the Frame/Elevator assembly which is shown in lowered position in Figure 1. This assembly forms a testing baseplate which is assembled when needed. It can be raised and lowered to a convenient working height, and can be pitched for a "drive-on" capability.

FRAME DESIGN - The Frame consists of two transverse beams and two longitudinal siderail beams. The transverse beams are rectangular aluminum tubes, reinforced by aluminum plates welded on the upper and lower surfaces, and by welded-on plates where attachment is made to the siderails. The siderails are 380 mm wide welded assemblies, 4090 mm long, made up of three rectangular tubes capped by welded-on 3 mm aluminum plates and 13 mm thick endplates. Galvanized "Unistrut" steel channels with rolled-over edges are attached to top and bottom edges and along the bottom centerline of the longitudinal beams. These are bolted at 152 mm intervals to flush-mounted "Rivnut" threaded inserts. The Unistrut rails add considerably to bending stiffness, and with the many available clamps and fittings provide a convenient system for attaching equipment to the Frame.

The basic moment of inertia of each siderail is estimated to be 1500 cm^4 of aluminum. The Unistruts add 500 cm^4 of steel. The composite moment of inertia is therefore approximately 3000 cm^4 of aluminum. The effective moment of inertia of a current transverse beam is about 625 cm^4 of aluminum, which will soon be doubled by welding on another upper and lower cap. Bending deflections are generally low because the distance from vehicle wheel centers to support members is short. The mass of each side rail is 98 kg and that of each transverse beams is 15 kg without elevators.

The attachment between a siderail and transverse beam is through 12 bolts at each end, with holes tapped into the siderail ends. Spacing between siderail centerlines is adjustable from 1270 to 1702 mm to accommodate vehicles of various trackwidths. Wider trackwidths will require welding on more plate and drilling more holes in the transverse beams, or welding up longer ones. The side beam length of 4090 mm accommodates vehicles of up to 3300 mm wheelbase.

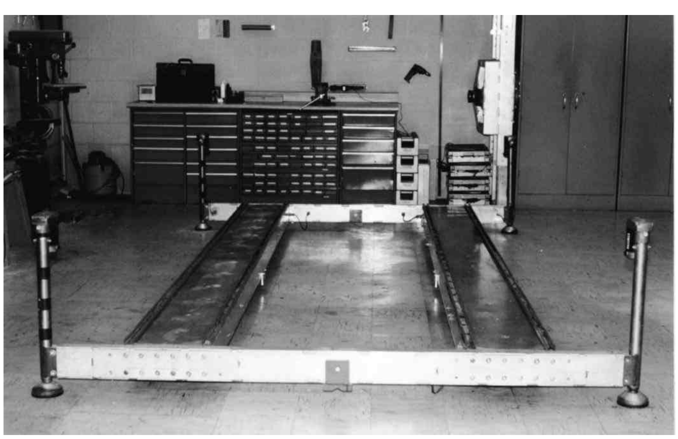

Figure 1: Frame/elevator assembly in down position

Ramps attach to the end of the siderails. With the Frame Elevator Assembly at maximum pitch angle, a vehicle can be driven onto the frame assembly.

The Frame has secondary use as a general- purpose work table to which components can be clamped for testing. When its space is needed for other projects, it is disassembled and stored on a shelf above the laboratory entrance doors. Assembly takes one man about one hour.

FRAME ELEVATORS - The Elevator Assembly consists of four motorized ball- screw linear actuators, attached vertically to the end rails at each corner of the Frame. Each actuator is capable of over 9000 N and has a stroke of 460 mm. The actuator ends are fitted to rubber or polyethylene feet through spherical bearings.

The actuator motors are connected through the inside of the Frame beams and a single cable to a hand-held control module, where they are controlled, singly or together, by toggle switches. The elevators provide a table height range of 215 to 675 mm, and a pitch range of ±6 degrees.

Each elevator actuator has a mass of 19 kg. They are stored in a closet when not in use.

VEHICLE LOCATING JACKS

SCISSORS JACK - The two scissors jacks, one of which is shown in Figure 2, were designed to lock the vehicle to the Frame at selected ride heights, for inertial and compliance measurements. When raised or lowered by turning the Acme-threaded screws with a wrench or air wrench they remain horizontal, so that front and rear ride heights can be adjusted without worry about introducing roll. They are laterally stiff, since parallelogramming requires bending the 19 mm diameter screw into a sine wave.

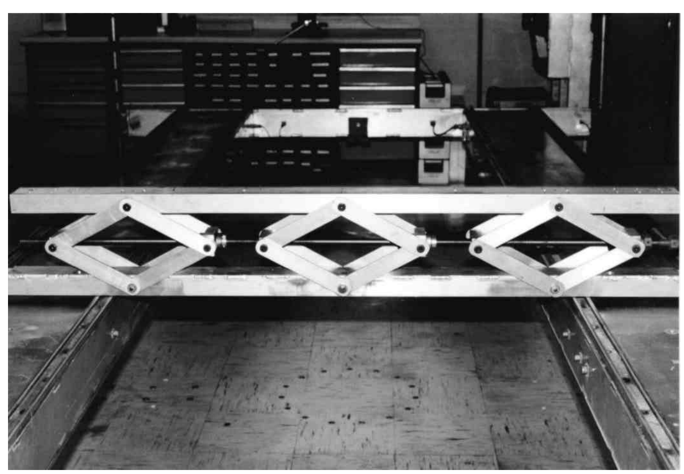

Figure 2: Scissors jack assembly

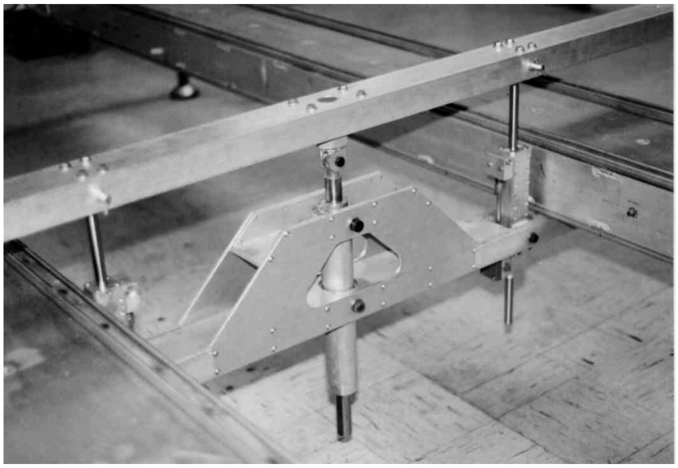

Figure 3: Ride/Roll crossmember mounted below Frame, with servo cylinders

Nevertheless, during early lateral compliance testing, in which displacement transducers were referenced to the Frame rather than the vehicle body, it became necessary to put adjustable threaded spacers under each threaded block to reduce the lateral compliance resulting from this sine wave bending. With the spacers installed, lateral deflections were reduced to 0.2 mm at 4100 N sideforce. The mass of each scissors jack is 19 kg. When not in use they are stored vertically in the "crossmember corner" of the laboratory.

RIDE/ROLL CROSSMEMBER - The "Ride/Roll" crossmembers, one of which is shown in Figure 3, were designed to exercise the vehicle suspension in kinematics testing, but can also be used in compliance tests. The center restraint for this crossmember is a 32 mm diameter steel shaft running in linear ball bearings in a cylindrical housing, which is attached to the lower crossmember "bridge truss" by shaft collar-trunnions. The cylindrical housing is adjusted up or down to optimize the loading on the vertical shaft. The inner shaft attaches by a spherical bearing to a clevis welded to a 38 mm diameter screw, which is threaded into a block in the upper crossmember. The location of the spherical bearing defines the point about which the vehicle rolls, provided that the shaft is vertically locked by tightened shaft collars. That point can be adjusted by screwing the clevis into or out of the crossmember. With the vertical shaft free, the vehicle finds its own roll center, which can be computed from the vertical shaft motion or by other means.

The ride/roll crossmembers can be assembled "upside down" or "rightside up", and they can be attached to either the bottom of the Frame as in Figure 3 or to the top side. One of the four configurations is chosen for a particular test according to test vehicle road clearance and ease of installation.

The vertical displacement and roll angle of the upper crossmember is controlled by any of several trunnion-mounted linear actuators. For compliance testing slow-moving electric motor ball screw actuators are used, or hydraulic cylinders are locked vertically by shaft collars. For kinematics and frequency response testing, a variety of hydraulic actuators are used, depending on force, stroke, and piston velocity requirements.

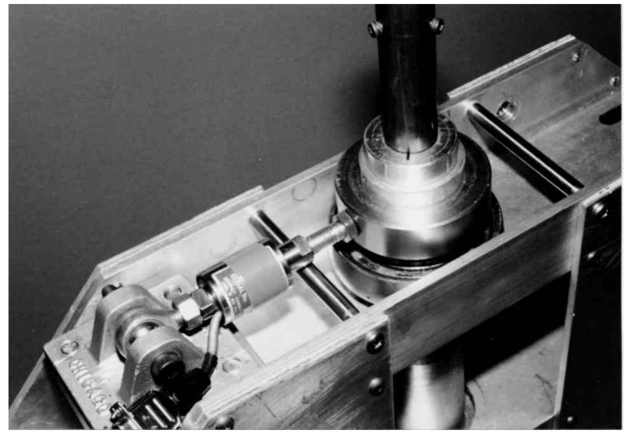

Figure 4: Side force measurement in roll dynamic testing

Removal of an actuator requires only pulling out a spring-loaded hitch pin from the upper spherical rod-end, and removing two trunnion bolts.

The housing for the vertical slider is normally located rigidly at top and bottom for lateral stiffness. However, for measurement of roll/yaw product in dynamic testing one end is located as shown in Figure 4. Here the housing is pivoted about its lower mount, but the upper end is free to move laterally except for the load cell restraint. Fore/aft loads are taken by the ball bearing, which tends to roll on either the front or rear surface of the channel with a clearance of about 0.2 mm. The vertical slider and the truss sides have been enlarged since this picture was taken.

The lower "truss" crossmember has a mass of 15.6 kg, and that of the upper crossmember is 11.6 kg. They store in "crossmember corner". The vertical bearing and shaft assembly has a mass of 5.3 kg, and stores in a closet.

ATTACHMENT TO VEHICLE - Where possible the Scissors or Ride/Roll crossmembers bolt to "rivnuts" installed in the vehicle frame or underbody. (After testing the rivnuts are plugged with setscrews.) For passenger cars, they are more often clamped to rocker pinch welds, using machinists vises modified for air-wrench installation. The vises bolt to or are clamped to the locating crossmembers.

ACTUATORS - An array of electric motor and hydraulic actuators are used interchangeably, with a "standard" trunnion mount. For essentially static positioning jobs, Saginaw ball-screw actuators are used, either with Saginaw motors or smaller Dayton motors with higher-ratio gearing. With one exception, motors are 90 volts, run off rectified 115 Vac and controlled by toggle switches or by operational amplifier-driven relays. For the pull-point locators in sidepull testing, the actuators use 12 volt motors driven by general-purpose servo amplifiers.

Hydraulic cylinders or servo cylinders are used whenever variable force or motion is needed. Four sizes of hydraulic cylinders are used, depending on force, stroke, and matching of piston velocity requirements to limited pump

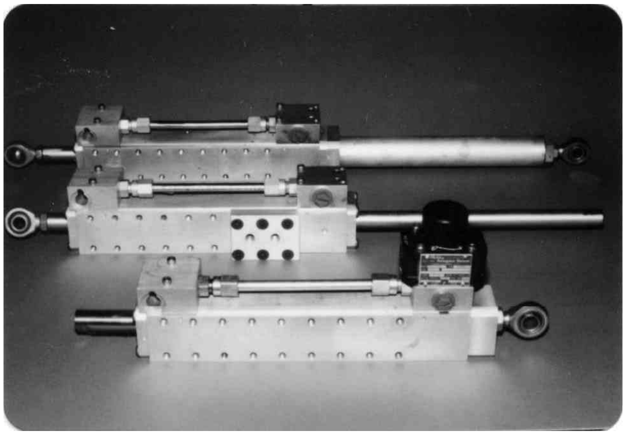

Figure 5: Hydraulic cylinder configurations

capacity. Piston areas (on the rod side) are approximately 1.3, 2.6, 6.5, and 12.9 cm^2 respectively.

The three smaller size cylinders use the same basic design, which may be configured as single-ended, or equal-area/double ended, by selection of end-caps and piston shafts. The photo in Figure 5 shows the same cylinder in three configurations: a single-ended, end mount in the foreground; with double-ended, trunnion mount and a double-ended, end mount in the rear.

Flow into and out of the cylinder is controlled by manifold blocks. The input end may be fitted with a simple ported block or with servo valve manifold blocks. The manifold blocks can be fitted with servo valves, or with red or blue "dummy blocks". Blue dummy blocks have the same flow direction as the simple porting blocks, and red dummies have porting of the opposite direction. Cylinders are easily changed from parallel to opposing directions by switching dummies.

The cylinder end opposite to the porting block has provision for a geared potentiometer, which is driven by a rack gear running in a groove milled into a corner of the cylinder. The potentiometer provides direct feedback without intervening compliance or backlash when the cylinder is used as a position servo.

The "big" cylinder is similar in design but is not fitted with a servovalve. The cylinder body is simply a section of thick-wall extruded tubing, which is drilled and tapped for end caps and fittings, then honed. Because it does not have to be bored, it is easy to make any length cylinder.

All cylinders use O-rings with less squeeze than that normally specified, for low friction.

The "bad news" about hydraulics is given by the approximate equation:

$$cm^3 / s = \frac{kW \times 10^6}{kPa}$$

At 10 000 kPa, a 4 kW pump supplies an average flow of about 400 cm^3/s. Since a single 6.5 cm^2 cylinder with a 5 cm peak-to-peak stroke at 1 Hz requires an average flow of 65 cm^3/s, the smallest diameter cylinder that will do the job is the one to be used in dynamic testing.

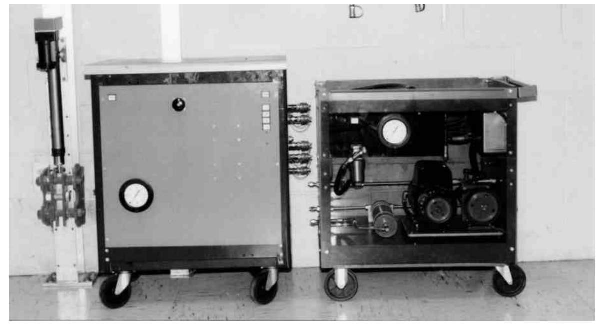

Figure 6: Hydraulic cabinets

SERVO VALVES - Flow control servovalves are used for position servos, and pressure control servovalves for force generation. The pressure control valves are mounted in assemblies containing a filter, valve, and differential pressure transducer. All servo valves are driven by standard operational amplifiers.

HYDRAULIC SUPPLIES - Two similar hydraulic supplies in roller cabinets are employed. Both are shown in the photo of Figure 6. One has a variable displacement pump driven directly by a 1.2 kW, 115 volt motor; the second one has an identical pump belt-driven by a 4.4 kW, 230 volt motor. The pump in the 4.4 kW cabinet can run at 10 000 or 20 000 kPa by changing the compensator spring and the belt pulleys. Both cabinets have the necessary filters, reservoir, accumulator, etc. The smaller one is fitted with two servovalves with outputs fed through solenoid valves for locking cylinders in place, for the sidepull system. The larger one is used in kinematics and compliance testing.

The hydraulic pump cabinets connect by flexible hoses to a floor-mounted distribution system, which for kinematics/compliance testing runs lengthwise under the Frame. The distribution system consists of parallel stainless steel lines with accumulators on each end, and servovalve manifolds or outlets where required. Relatively large accumulators are used to provide flow capacity well in excess of pump capacity for periods of several seconds. The accumulators are charged by a motor-driven 20 000 kPa air compressor.

WHEEL-LOAD SCALES AND BEARING PADS- Four strain-gage platform scales are used for vehicle weight and individual wheel loads. Each scale platform is 381 mm square and 100 mm high. An accuracy of ±0.02 percent is specified with the load located anywhere on the platform. Maximum load is 8900 N per scale. Accuracy certification is to NTEP Class III, one part in 6000.

The scales are not "drive-on": the vehicle must be raised and then lowered onto the scales.

For measurements of suspension kinematics and compliances, each wheel scale is fitted with ball-bearing low-friction pad assemblies. Each assembly consists of upper and lower hardened steel plates, separated by an array of hardened steel balls. Each upper plate is fitted with mounting ears for a wheelpad forcer assembly, and has a serrated steel friction surface.

POSITION TRANSDUCERS - Two types of position transducers are used. Spring-loaded rectilinear potentiometers are used for small, well-defined deflections such as lateral vehicle motion or deflections of the Frame in CG tests. For all other work we now use "home-made" string pots or string encoders. A string device (Figure 7) consists of a spring motor with its output cable wrapped in a single turn around a pulley mounted on a pot or encoder shaft. Spring motors come in several tension levels, and are inexpensive and reliable. Since the "chassis" is just a piece of flat stock it can be modified for a given application and replaced for the next. Precise encoder scaling is accomplished by selecting pulses per revolution and then machining a pulley to the correct diameter.

Encoders are used wherever possible because they have good resolution with infinite "stroke", and are zeroed anywhere in that stroke with a pushbutton switch. Potentiometers are used only when position data should not be lost when power is turned off.

ACCELEROMETERS/INCLINOMETERS - Force-balance servo accelerometers, either Systron-Donner Model 4310 or Sundstrand Model 303B, at 0.25, 0.5, or 1.0 g full scale, are used as inclinometers. These units have linearities of 0.05 percent full-scale, and resolutions of 0.0006 degrees. A Systron-Donner force-balance servo angular accelerometer with ±10 radians/second full scale, 0.1 percent linearity, and 0.005 percent resolution is used in angular inertia testing.

STEERING WHEEL - A "second steering wheel" instrumented to measure angle and torque is used in compliance and steering system measurements.

WHEELHUB TRANSDUCER PLATE - Wheelhub transducer plates shown in Figure 8 are designed for steering ratio and geometry, kinematics, and compliance measurements. Adapter plates are attached to lugnut extensions through four or five slots which guarantee accurate centering. The resulting spin axis extension is a threaded shaft containing a ball bearing hanger for a plumb bob or a tire deflection encoder, and a lockable attachment to the wheelhub transducer plate. This plate contains camber and caster inclinometers, a bubble level, a rearward-facing laser pointer for steer angle zeroing, a ±45 degree, 0.05 degree resolution “coarse” steer angle encoder, and attachment points for the fine steer angle encoders required for kinematics and compliance measurements.

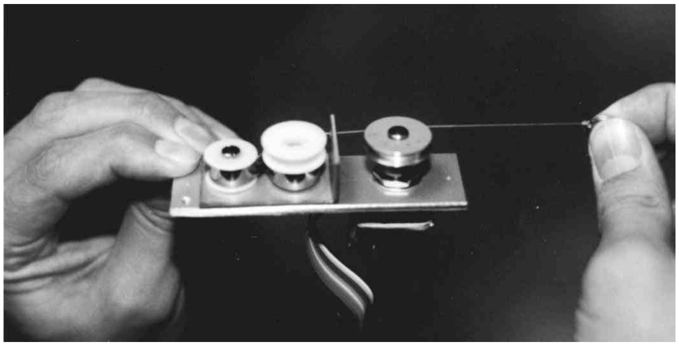

Figure 7: Small string encoder

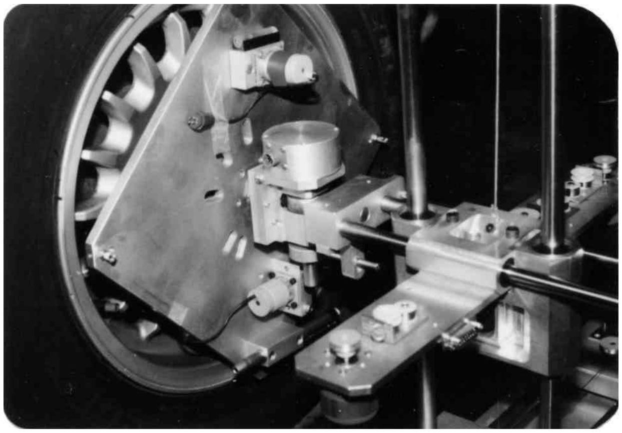

Figure 8: Wheelhub transducer plate

BALL-SLIDE FOLLOWER MECHANISM - The ball-slide mechanism shown in Figure 9 was designed to measure wheelcenter motion in kinematics and compliance testing, and vehicle bumper motion in sidepull testing. It consists of three orthogonal, dual-shaft, linear ball-bearing sliders, in X-Z-Y configuration, with motion measured by spring-motor encoders. Range of motion as determined by lengths of the slider shafts is arbitrary. At present, X-motion is 460 mm, Z-motion is 305 mm, and Y-motion is 230 mm, but the short Z-motion shafts shown in the figure are used for closet storage. Transducer range is arbitrary since the encoders can be zeroed anywhere. The encoders installed for X-Z-Y motion have 256 pulses per revolution and 20.7 mm diameter pulleys, which with the 12-bit data acquisition conditioners result in ±127 mm full-scale with 0.06 mm resolution at 4X, and ±508 mm full-scale with 0.25 mm resolution at 1X. Force levels are 4 N in the X-axis and 1.6 N in the Y-axis, determined by the spring motors. In the Z-axis the weight of the Y-axis portion and the pull-down Z spring motor are counterbalanced by a vertical spring motor.

For compliance and kinematics testing small steer angles are measured by two additional encoders, spaced 356 mm apart to match attachment points on the wheelhub transducer plate, as shown in Figure 8. These steer angle

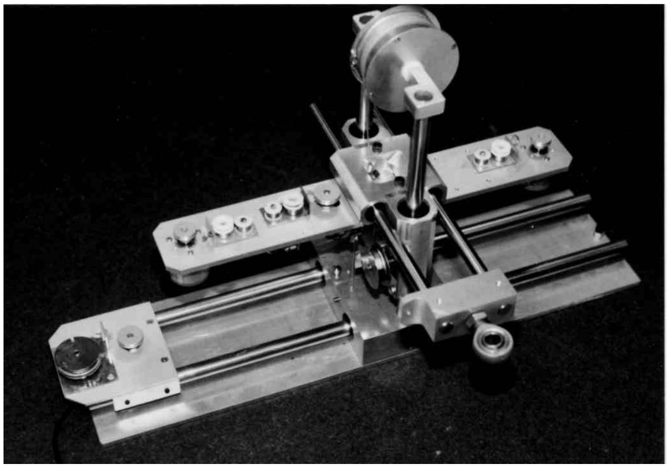

Figure 9: Ball-slide follower mechanism

Figure 10: Data acquisition console

encoders have 1,000 pulses per revolution and pulleys chosen for a 4X full-scale of ±4 degrees with 0.002 degrees resolution, and a 1X full scale of ±16 degrees with resolution of 0.008 degrees. As the slider mechanism follows the X and Z wheel motions, no corrections for such motions are needed.

DATA ACQUISITION SYSTEM - The proprietary data acquisition system shown in Figure 10 allows close monitoring of a test in progress, and useful review of completed test programs. All testing is videotaped, with raw transducer data recorded in serial binary form on one or more lines at the top of the picture. Voice, noises, and transducer outputs are recorded in synchronism continuously during the entire test on two-hour tapes, so that anomalies that might occur during or between test runs can later be investigated. Throughout the test, data is displayed as columns of numerics, twelve channels to a column, overlayed on the video picture, for visual monitoring. The displayed overlay does not have to be recorded, as it can be recreated and recorded at any time. All processed data is referenced to hours, minutes, and seconds of tape time, so that final plotted data can be referenced to the video. Figure 11 shows a typical compliance test screen with numeric overlay.

Transducer data is conditioned in general-purpose, strain gage, shaft encoder, or frequency input amplifiers; and in tenth-order 14.2 Hz Butterworth active filters. All signal conditioners have manual and autozero capability, and on-board calibration (E-Cal, R-Cal, Excitation Off, Fixed-Count, or crystal oscillator) appropriate to their function. The filtered data is sampled 60 times per second, digitized to 12 bit accuracy and resolution, and stored with 12 channels on one horizontal line of the video recorder. Each data line appears as a dot-dash line across the top of the visible video picture. Figure 12 shows a screen with 24 channels of binary data on two video lines.

The system is designed to use a maximum of nine "data lines" with 12 channels per line, for a total of 108

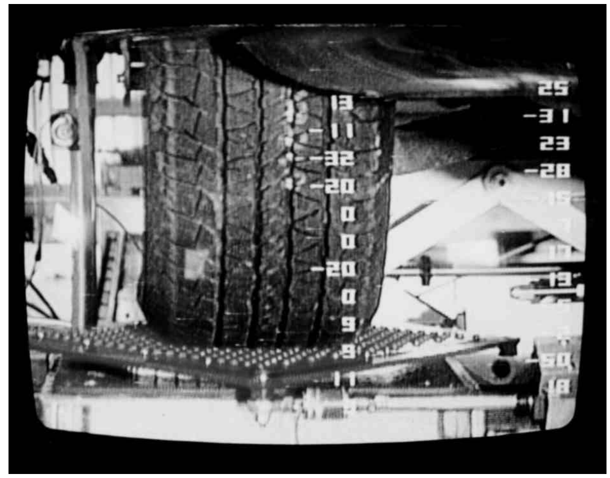

Figure 11: Screen with numeric overlay

10-Hz data channels sampled at 60 Hz. However, it is also possible to devote some data lines to "high-speed" use, with two data channels sampled at 480 Hz or one data channel sampled at 960 Hz.

Interface between the Pentium-PC computer and either the data acquisition system or the replayed tape uses a "RAM Buffer" module. Through this module, the computer sees the data being produced or reproduced as RAM which is updated with twelve 12-bit data samples, (or high speed equivalents) in 53 microsecond bursts occurring sixty times per second.

CALIBRATION OF INSTRUMENTATION - Wheel scales are calibrated by recording the placing of 24 NBS Precision F (±0.002 percent) test weights (each 50 pounds, approximately 222 N) on the scale, one at a time, then transferring them one at a time to the next scale, etc., plotting the results, and computing the best straight line through the data for each scale.

The CG moment arm load cell and roll inertia springs are calibrated in the same way: by putting the weights on the Frame against a Unistrut rail one at a time, then transferring them one at a time to the same location on the other side and plotting the result.

The small tension/compression load cells are calibrated by screwing several in series with a master load

Figure 12: Screen with only binary data

cell used only for that purpose, and loading the composite with a hydraulic cylinder and a triangle wave generator. Individually they are single-point checked with precision test weights. Sidepull load cells are single-point checked by hanging precision 2500 pound and 5000 pound test weights (approximately 11 000 N and 22 000 N), in visits by the New Jersey State Weights and Measures truck.

All load cells generally use identical 9 meter cables, both in calibration and in use. During calibration each load cell has its R-Cal checked against its own history for its output with a Micro-Measurements ±0.02 percent R-Cal resistor. Ordinary one percent resistors are then installed for checking during a test program. If for some reason a non-standard cable is used for a test the precision R-Cal can be used to adjust excitation level to compensate for cable wire resistance.

Amplifiers are checked before each use with an Electro-Services Corp "Transducer Simulator", in 0.5 millivolt/volt increments to 3.0 millivolts/volt.

Calibration of inclinometers are checked periodically on a sine table with machinist's gageblocks.

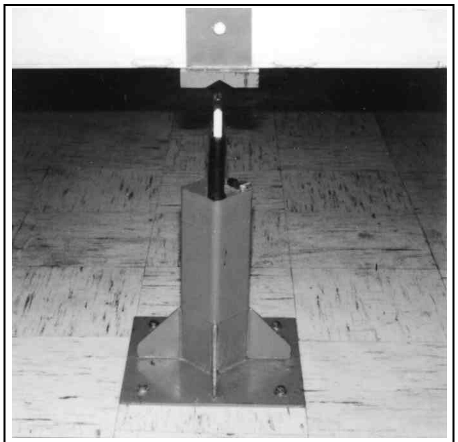

Figure 13: Knife edge assembly

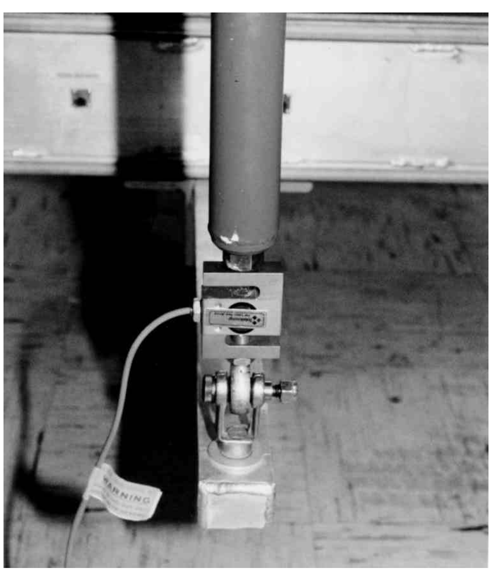

Figure 14: Torque arm with actuator load cell for CG height

Before each use string encoders and pots are checked over 450 mm travel by a Starret metal ruler, and rectilinear pots are checked by a vernier caliper.

EQUIPMENT FOR WHOLE-VEHICLE INERTIA PROPERTIES

For measurement of CG height, the Frame sits on strain-gaged knife-edges (Figure 13), with roll motion forced by a ball-screw linear actuator equipped with a 2225 N load cell (Figure 14). The actuator attaches between a transverse torque arm bolted to Unistrut rails on both sidebeams, and the lab ceiling. The actuator motor and position potentiometer connect through a ceiling plug to a wall-mounted speed control box and to the data acquisition system. For CG measurement the actuator forces constant-speed roll motion between reversing limit stops set at ±12.5 degrees.

An unloaded, rectangular tube, “longitudinal reference beam” is installed between the front and rear knife-edge pads. Transverse channel sections are clamped to the Frame lower-surface Unistrut rails below the front and rear wheelcenters to position vertically-mounted rectilinear potentiometers with their spring-loaded shafts impinging on the longitudinal reference beam, to measure Frame deflection. Vertical string potentiometers are installed between the reference beam and vehicle at front and rear axle locations to monitor ride height during a test.

For roll moment of inertia calibrated coil springs are bolted between the Frame ends and the floor. The Frame rocks back and forth at a frequency of approximately 1 Hz, determined by the spring rates and the inertia of the vehicle-plus-Frame. The knife-edge strain gages measure side forces which result from roll/yaw coupling. The oscillation period is measured by the angular accelerometer.

Pitch moment of inertia is measured in the same way, with the knife-edges relocated to the midpoint of the side beams.

For yaw moment of inertia the Frame sits on a "Yaw Cradle". The "cradle" (Figure 15) is a frame made up of welded rectangular aluminum tubing bolted to the lab floor through a 500 mm Rotek ball bearing. The Rotek bearing was chosen because it has bolt flanges on both inner and outer

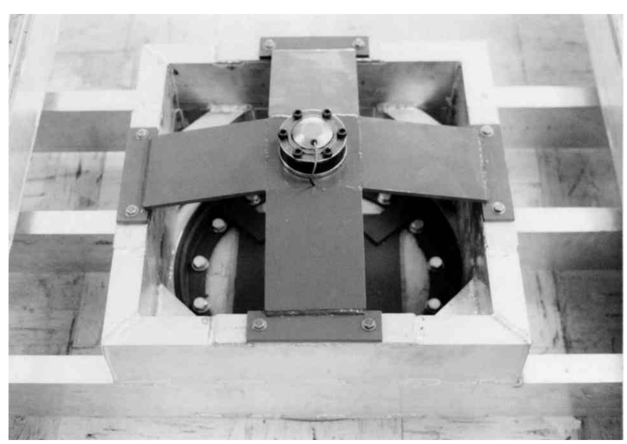

Figure 15: Yaw Cradle

races, and does not require extreme flatness in mounting surfaces.

The cradle inner and outer races are connected through a torsion bar for angular restraint. The torsion bar is a strain-gaged BMW axle shaft, complete with constant-velocity universal joints. Test frequency is about 0.33 Hz.

When not in use the Yaw Cradle stores vertically, chained to a storage cabinet to ensure against tipping.

Figure 16 shows the storage system for everything except the Frame: calibration weights, scales, turntables, knife edges, Yaw Cradle, and "crossmember corner" are external, and everything else fits inside the cabinets.

MEASUREMENT PROCEDURES FOR WHOLE-CAR INERTIA MEASUREMENTS

VEHICLE INSTALLATION - The vehicle is cleaned to remove accumulated dirt and its fuel tank is filled to overflowing; then it is weighed and the longitudinal and lateral location of its CG is computed. The longitudinal reference beam is installed, and the Frame-deflection potentiometers are attached below the calculated wheel center locations. No-load Frame deflection is recorded, and then the vehicle is driven on. The scissors jacks are attached to the vehicle, by bolting to Rivnuts installed in the vehicle underbody or frame or by clamping to rocker weld flanges. The vehicle is rolled to locate its longitudinal CG at the Frame midpoint, then rollers are placed under the scissors jacks. The vehicle is lifted slightly and centered laterally, then it is lowered and the rollers are removed. The scissors jacks are bolted to the Unistrut rails and the vehicle is set to the specified ride height.

Pots or encoders are installed to measure lateral deflection under load and any ride height changes. Supplementary, monitoring vertical stringpots are installed between the Frame and the rocker midpoints, the radiator support crossmember and the rearmost available point on the vehicle structure. All transducers are zeroed.

If moments of inertia are to be measured, scales are placed under the four corners of the Frame elevators, and barbell weights are attached to the Frame ends to adjust the longitudinal Frame CG location to its midpoint, compensating for equipment location. The vehicle is now located so that its CG and that of the Frame are directly over the Frame center.

CG HEIGHT - The Frame is lowered onto the knife edges, the vertical actuator is installed, and then the elevators are totally retracted.

The vertical actuator is set to cycle through its ±12.5 degrees range at its "fast rate" of 1 minute per cycle, while the data are observed. Any excessive deflections, or shifts as motion goes through center, require correction. The front and rear inclinometers must read the same, and they must agree with the potentiometer in the vertical actuator. The video recorder is turned on, and the actuator is then set to its 3 minutes per cycle "test rate" for at least six additional cycles.

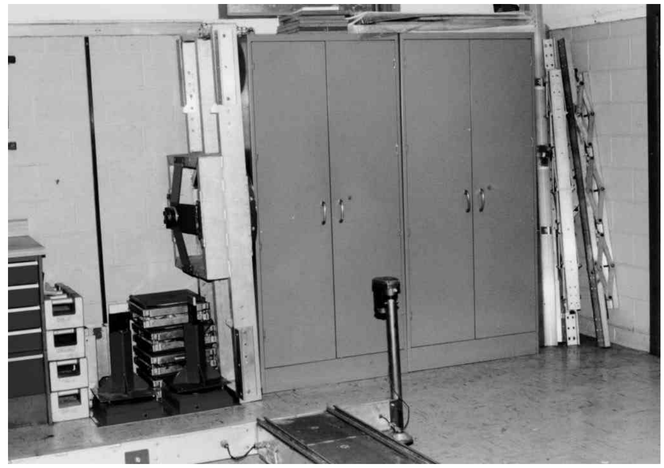

Figure 16: Equipment storage

If cycle-to-cycle transducer readings are identical in each cycle, the test run is over.

If desired, the ride height is then reset, and the procedure is repeated.

After all vehicle-on inertial testing is completed the location and heights of the scissors jacks is noted, and they are removed. The vehicle is removed, and the vehicle-off readings of the Frame deflection potentiometers are recorded.

The scissors jacks are re-installed on the Frame, the gain of the load cell signal conditioner is switched to a 10x higher sensitivity, all vibration-creating devices (overhead fan, any rotating machinery) are turned off, and the measuring process is repeated for computation of the Frame tare.

Data processing, as described in Appendix 1, consists of plotting the restoring force and vehicle lateral deflection versus the tangent of the tilt angle over 3 cycles, and computing the slopes for the vehicle-plus-Frame and for the Frame alone. The computation is summarized in the following equation:

$$H_v = \frac{R}{W_v}\left[\frac{dF_{V+F}}{d\tan\theta} - \frac{dF_F}{d\tan\theta}\right] - \frac{dY_s}{d\tan\theta}$$

where

H_v = height of the total vehicle CG above the knife edges

R = torque arm length

W_v = weight of the total vehicle

θ = roll angle

$dF/d\tan\theta$ = slope of the force/angle relationship for the vehicle-plus Frame and for Frame alone

$dY_s/d\tan\theta$ = slope of the lateral shift/tilt angle relationship of the vehicle CG

Typical data plots are shown in Figure 17. Forces are corrected before plotting as discussed in Appendix 1. A single slope is shown for $dF_{v+F}/d\tan\theta$ in Figure 17; however, an incompletely-filled fuel tank will cause a small, but noticeable

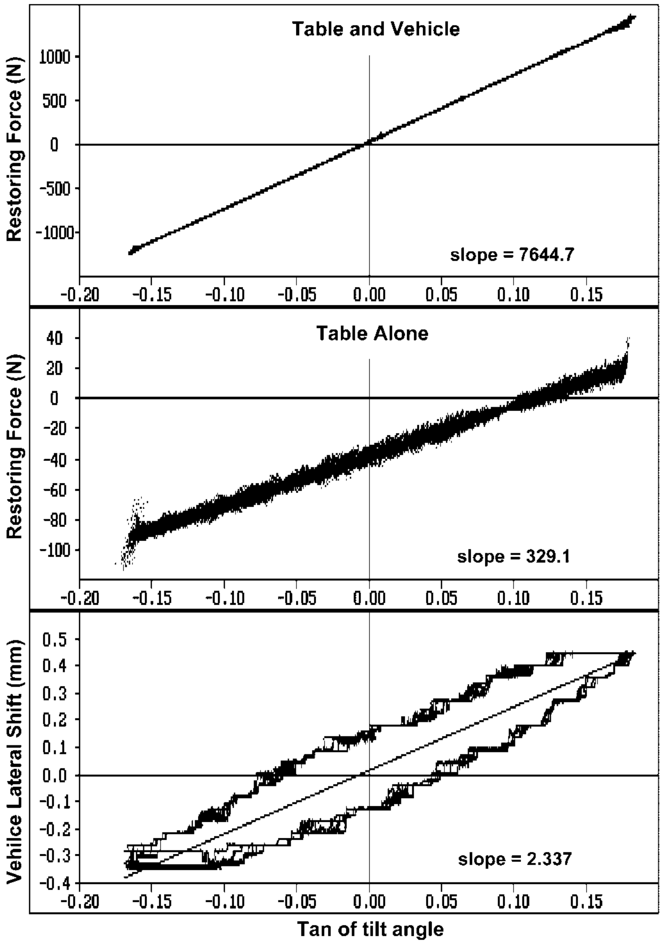

Figure 17: CG Height Test

local slope increase through the zero tilt angle point. For this reason we also compute individual slopes for positive and negative tilt angles, eliminating the center ±0.5 degree.

The height of the "road surface" above the knife edges, including the measured Frame deflections, is then subtracted to obtain the total-vehicle CG height above the road surface.

CG height is usually measured at curb weight, and at additional trim heights to encompass loading conditions. We prefer not to test at different loads, because of the complexity of locating and securing "equivalent dummies" and adjusting resulting ride heights. We believe that loaded CG height can better be computed from the spring deflections due to added load, the individual heights of any applied load, and the vehicle CG measured at different ride heights.

The CG heights of sprung and unsprung masses can be separated by measuring the total CG height at different trim heights, as described in Reference 3. This method must be utilized with care, since it assumes no resulting deflections of vehicle structure or engine/subframe mounts. For sensitivity the trim height differences should be as large as possible: however, too-large differences can create nonlinear displacements of heavy components which in some vehicles may create significant errors. Especially, unloading to the point where independent suspension rebound stops are engaged should be avoided. Since it is easy to re-measure at different trim heights, we feel it advisable to measure at three trim heights encompassing the loading range, plus one about 2.5 cm in rebound, and plot the result to evaluate the trends. Since it is also easy to add stringpots or encoders, it is advisable to monitor deflections of the engine cradle and the body structure, and make corrections if necessary.

ROLL MOMENT OF INERTIA AND ROLL/YAW PRODUCT

THEORY - In the test setup, the pitch axis is considered as being perpendicular to the plane of symmetry. The system is then one of two degrees of freedom for which the equations of motion are:

$$I_{xx}\ddot{\phi} - I_{xz}\ddot{\Phi} = L$$
$$-I_{xz}\ddot{\phi} + I_{zz}\ddot{\Phi} = N$$

where

I_{xx} = moment of inertia about the roll axis

I_{zz} = moment of inertia about the yaw axis

I_{xz} = product of inertia

$\ddot{\phi}$ = angular acceleration in roll, deg/sec^2

$\ddot{\Phi}$ = angular acceleration in yaw, deg/sec^2

L = Moment about roll axis

N = moment about yaw axis

The knife edges prevent any yaw displacement; therefore the equations of motion reduce to:

$$I_{xx}\ddot{\phi} = L$$
$$-I_{xz}\ddot{\phi} = N$$

ROLL MOMENT OF INERTIA - The Frame, with the vehicle installed and centered as described above, is lowered onto the knife edges, and calibrated coil springs are bolted to the corners of the Frame and to the laboratory floor. The angular accelerometer is attached to the vehicle. A string encoder is installed to measure roll amplitude. The knife edge strain gages are connected. The elevators are completely retracted. The recorder is turned on, and the Frame is oscillated by hand to about ±6 degrees amplitude and released. Settling time for the resulting oscillation is around 150 periods of about one second each. Transducer data is fed to the computer for immediate generation of stripcharts. These are checked to identify any glitches or inconsistencies. If everything looks good the vehicle test is over.

The test is repeated for the Frame and vehicle restraint equipment alone.

The moment of inertia of the total vehicle and Frame about the knife edges is found by equating the moment exerted by the springs to the angular acceleration. The moment of inertia of the sprung mass about a horizontal axis through its CG is then obtained by subtracting the moments of inertia about the knife edges of the Frame and the various unsprung masses, and then correcting this result to the sprung mass CG by the Parallel Axis Theorem:

$$Ikk_T = (4Ka^2 - W_T h_T)(P/2\pi)^2$$
$$Ixx = Ikk_T - Ikk_F - Ikk_{UM} - (W_S/g)h_S^2$$

where

Ixx = Moment of inertia of sprung mass about a horizontal axis through its CG

Ikk = Moments of inertia about knife edges of vehicle plus Frame, Frame alone, and Unsprung Masses

a = Lateral distance from knife edge to centerline of spring

K = Rate of each spring

h_T = height of Vehicle & Frame CG above knife edges

h_S = height of vehicle sprung mass above knife edges

W_T = total weight on knife edges

P = Period of oscillation

ROLL/YAW PRODUCT OF INERTIA

When the frame is oscillating about the horizontal knife edge roll axis, the lateral acceleration of the mass CG produces a side force on the knife edges, and the angular acceleration produces a yawing moment which is reacted by the knife edges. The product of inertia is found from the yaw moment by the following equation:

$$Ixz = (T/\phi_0)(P/2\pi)^2 - Ixz_{um} - (W_s/g)(x_{st}h_s)$$

where

Ixz = product of inertia about longitudinal and vertical axes through the cg of the sprung mass

T = moment reacted by the knife edges

ϕ_0 = roll displacement, in radians

P = oscillation period, in seconds

Ixz_{um} = estimated product of inertia of unsprung mass with respect to the knife edge axis and the vertical axis through the CG of the test configuration

W_s = weight of sprung mass

g = gravitational acceleration

x_{st} = longitudinal distance of sprung mass cg from cg of test configuration

h_s = height of cg of sprung mass over knife edge axis

Note that the Frame is symmetrical, and so does not contribute to roll/yaw moment.

PITCH MOMENT OF INERTIA - The knife edges are located under the midpoints of the side beams, and the angular accelerometer is installed to measure pitch motion. Otherwise the procedure is identical to roll.

YAW MOMENT OF INERTIA - The elevators are raised to full height, the longitudinal reference beam is removed, and the yaw cradle is slid underneath and bolted to the floor. The Frame is lowered onto the Cradle, centered by mating wedges on the Cradle and Frame, and the elevators are fully retracted. The Frame is bolted to the Cradle. The yaw accelerometer is attached to the vehicle, a linear accelerometer is attached to the end of the Frame, the cradle's torsion bar strain gage is connected, a yaw amplitude encoder is installed, and the recorder is turned on. The Frame is oscillated manually in yaw to about ±10 degrees and released. (Settling time is 10 - 12 cycle periods of about 3 seconds each). Transducer data is fed to the computer for immediate generation of stripcharts. These are checked for glitches or inconsistencies in torsion bar or accelerometer output signals, and for vehicle movement with respect to the Frame. Corrections are made if necessary. Five or six data runs are then made. After removal of the vehicle, the test is repeated for the Frame alone, to obtain the tare value.

The moment of inertia is computed from the amplitude and frequency of the oscillation and the torsional spring rate, or from the angular acceleration and the spring torque.

DRIVETRAIN INERTIA - Simple pendulums consisting of 23 kg cylindrical weights on 610 mm rods, attached to the wheelhubs are used to measure rotating inertia.

EQUIPMENT FOR KINEMATICS TESTING - The Ride/Roll vehicle location crossmembers are used, with hydraulic cylinders in double-ended/equal area configuration.

The front Ride/Roll crossmember is rigidly attached to the Frame. The rear crossmember is attached to the Frame through spherical bearings, which permit a 5-degree fore/aft rocking motion to accommodate "cosine foreshortening" when the vehicle is pitched.

The vehicle sits on the bearing-equipped scales, restrained only by the two ball-bearing vertical sliding shafts. Those shafts are unrestrained in ride, and may be unrestrained or locked in roll.

If unrestrained in roll the vehicle "finds its own" roll centers, which can be located by measuring the vertical displacement of the restraint shafts, or by wheelpad displacement or force equilibrium methods. If used, each vertical restraint defines a point on the roll axis, which can be located from 25 to 100 mm below the bottom of the rocker panels, and to the ground if spacers are added.

Wheel adapter plates are attached to lugnut extensions, then the transducer plates are added and leveled. The transducer plates carry camber and caster inclinometers, and rearward-facing lasers for setting wheels in the straight-ahead position. They mate to the ball-slide wheel follower mechanism.

The ball-slide wheel followers are attached to the vehicle, on beams clamped to the Ride/Roll upper crossmembers. Tire deflections are measured separately, by an encoder attached between a ball bearing on the transducer plate axis, and the wheelpad.

For parallel ride motion the cylinders are plumbed in parallel, with all dummy blocks blue, with the two front cylinders driven by one pressure control valve and the two rear cylinders by another. The front valve is driven by a triangle-wave generator, and the rear nulls an inclinometer to keep the vehicle level. For roll the dummy valve blocks on one side are changed to red, and both valves are driven by the triangle generator for equal front and rear moments.

KINEMATICS TEST PROCEDURE - The Ride/Roll Crossmembers are installed, scales fitted with ball-bearing pads are located under each wheel, and the various displacement transducers are attached. The instrumented "second steering wheel" is installed, and the steering wheel angle is set for straight-ahead road wheels and fixed. The brakes are locked ON.

The vehicle sprung mass is exercised by hydraulic servos through several cycles of jounce, pitch, and roll motions from compressed bump stops to wheel-off, at about one cycle per minute. Throughout each run measurement is made of wheel loads; steer angle; displacements of wheel centers, tires and shock absorbers; wheelpad rotation and lateral translation; and anything else that may be of interest. Test runs are normally made with anti-roll bars both connected and disconnected. All data is recorded for each run on the test videotape.

In data processing all data is plotted against time and inspected for smoothness and consistency. Three cycles of all desired data are plotted against the selected motion variable. Third-order least-square equations are fit to data plots except spring rates, where segmented first-order fits are usually required.

If roll centers are defined as the center of rolling motion in the absence of side forces, their heights can be computed from wheelpad displacements in ride or in roll [2], or from the vertical motion of unlocked lateral restraint shaft.

Figure 18 shows typical results for steer angle, camber, and wheelcenter Y-deflection. Figure 19 shows typical results for a roll test.

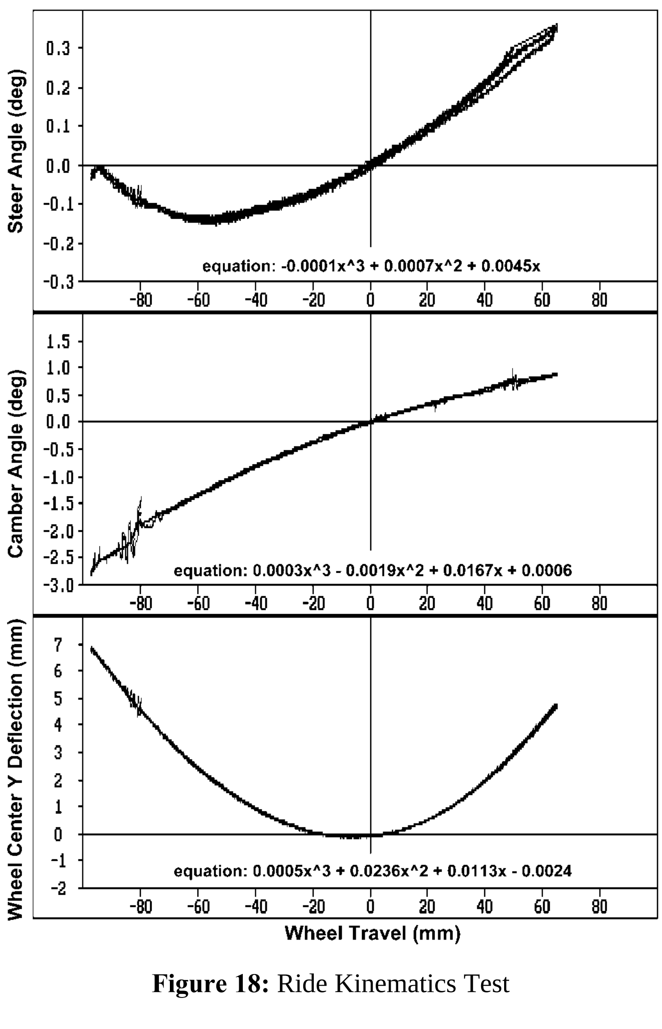

Figure 18: Ride Kinematics Test

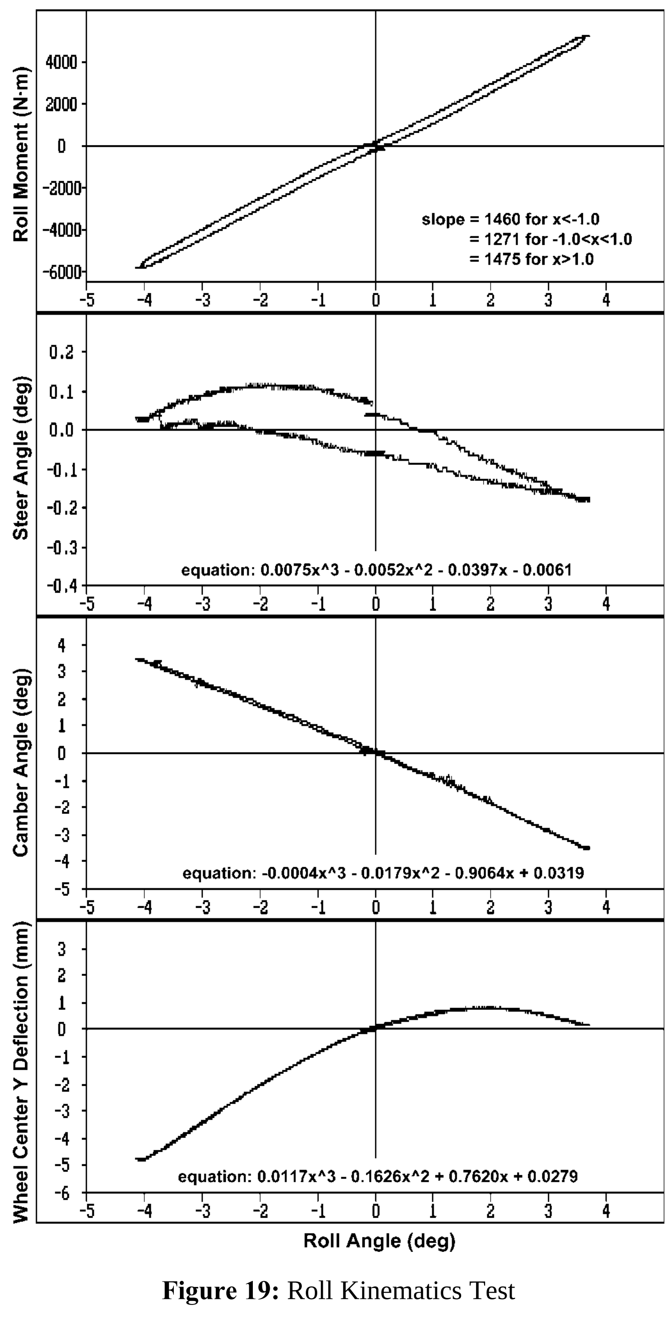

Figure 19: Roll Kinematics Test

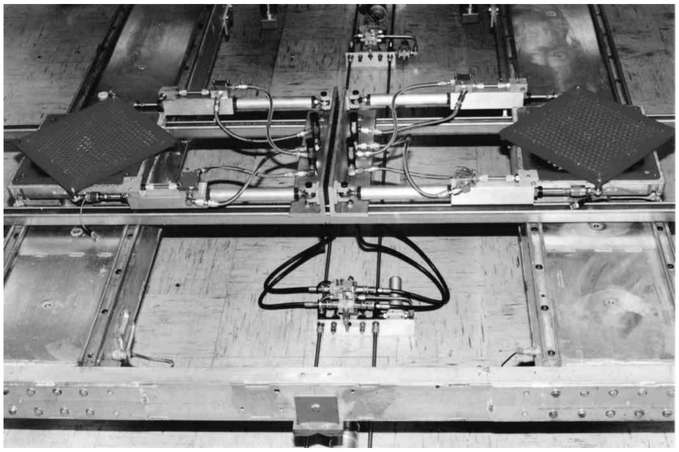

Figure 20: Wheelpad forcer assembly for compliance measurements

EQUIPMENT FOR COMPLIANCE MEASUREMENTS - Either the Ride/Roll or Scissors Jack crossmembers can be used. If the Ride/Roll crossmembers are used, the hydraulic cylinders are locked in position by rod-mounted shaft collars to prevent vertical motion.

The vehicle sits on the wheelscale-bearing plates. A set of four servo cylinders fitted with load cells are mounted on two transverse Unistrut crossmembers, and attached to mounting ears on the bearing pads, as shown in Figure 20. The four cylinders are driven by a single pressure-control servovalve, so that their force outputs are all identical. A small amount of 40 Hz dither is added to the servovalve signal to eliminate friction. Red and blue dummy valve blocks determine the direction of force output from each cylinder, to create "aiding" or "opposing" forces or moments. The servovalves are driven by a triangle wave generator.

LATERAL FORCE AND MOMENT COMPLIANCES - The vehicle is secured to the frame by either the Ride/Roll or the Scissors jack method at the desired trim height. Transducer plates and ball-slide wheel followers are used as in kinematics measurements. The wheel followers are attached to the vehicle. The force-measuring "second steering wheel" is attached to the vehicle handwheel and fixed by struts and suction cups to the vehicle windshield. The brakes are locked.

Tires are carefully centered on the wheelscale bearings, using a plumb bob or vertical laser for longitudinal and lateral location. A check run is made in "toe in/toe out" configuration to insure that sideforce produced by torque is negligible. The plumb bob or laser point is marked on the wheelpad surface, and the servo cylinder shaft extensions are noted, to restore position after wheelslip.

Test runs are made with aligning torques aiding and opposing, and sideforce aiding and opposing. In each configuration runs are made at incrementally increasing peak force levels, up to wheel-slip. Each run consists of at least four cycles at a rate of one cycle per minute.

When wheelslip occurs, the servos are used to power the tire back into place.

Steer angle and X,Y,Z wheelcenter displacements are measured from the ball slide follower. Camber and caster changes are measured by the accelerometer-inclinometers on the wheelhub transducer plates. Force and moment inputs are measured by load cells on each of the forcing servo cylinders. The cylinder differential pressure is measured as a check of measured forces.

Force and pad displacement inputs are plotted against time and inspected for quality and force/moment crosstalk. Three cycles of all desired data are plotted against input forces or moments, and third-order least-squares equations are fitted to the plots to obtain slopes. Typical plots, for an independent rear suspension measured in "sideforces opposing" configuration, are shown in Figure 21.

If roll centers are defined as points about which forces balance at zero roll angle, their heights are computed from the ratio of vertical load change to applied horizontal force in the sideforce-opposing test.

LONGITUDINAL FORCE AND MOMENT COMPLIANCES - The combined wheelpad- forcer servos are rotated 90 degrees to generate longitudinal forces. Longitudinal forces opposing at each pad produce the same toe in/out and steer moments as in the lateral case. All-forces-aiding measures longitudinal compliance, and one

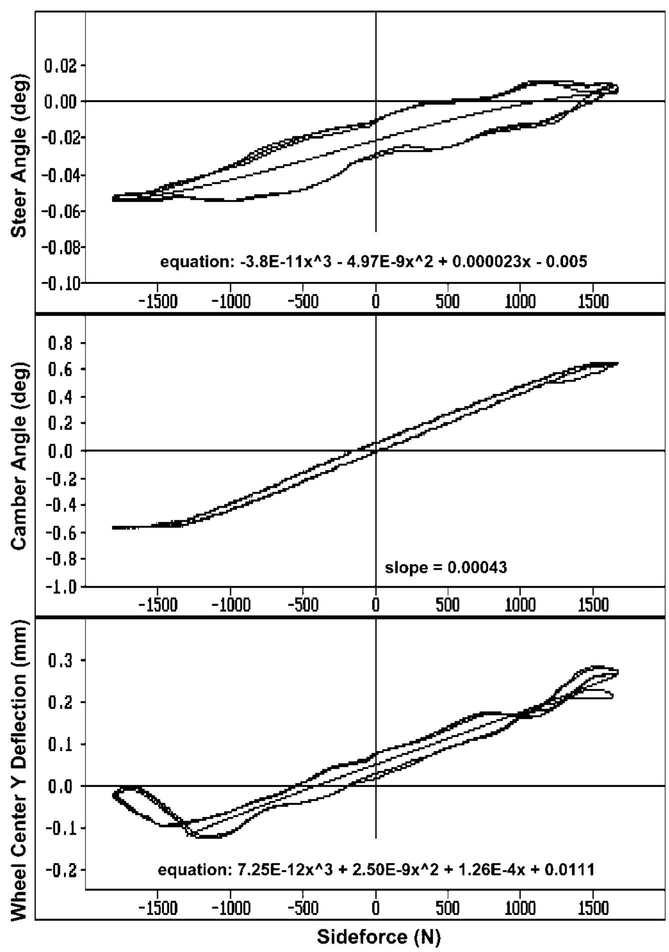

Figure 21: Typical compliance data

wheel aiding with the opposite opposing simulates unbalanced braking or thrust.

Data plotting is the same as in the lateral case.

STEERING SYSTEM MEASUREMENTS

COMPLIANCE - The compliance of the entire steering system including gearbox and column is measured in the sideforces and aligning torques aiding configuration. Sideforces and aligning torques opposing tend to balance at the steering gear, and so do not load the column. With power steering the only column torque is that required to twist a small torsion bar input to the valve: therefore with the engine running, aiding and opposing results are similar. Figure 22 demonstrates this characteristic. With the engine running in the "Aligning Moments Aiding" (i.e., steer) configuration the on-center steer compliance, before full boost occurs, is similar to that with "engine off, moments aiding"; while the off-center compliance is similar to the "engine off, moments opposing" configuration.

Figure 23 shows the torque reaction at the steering wheel resulting from total aligning moment (aiding) at the tire contact patch, with engine running and off.

STEERING GEOMETRY - Steering ratio is measured by the rotary encoder mounted between the ball slide follower mechanism and the wheelhub transducer plate. The encoder has a resolution of 0.05 degrees at 4X.

If ground-level steer axis offsets are required, instrumented turntables are used. These are standard units (Mac Tools, Inc.) to which rotary encoders and rectilinear potentiometers have been added. They were formerly fitted with ears for the hydraulic cylinders and used on top of the wheelscales in compliance testing, but they suffered from surface brinelling and were too high, and so they were replaced by the bearing pads. They directly replace the wheelscale/bearing pad units, and measure both steer angle and pad X and Y displacements. The encoder has a resolution of 0.05 degrees, and that of the cermet pots is infinite.

In testing, the instrumented handwheel is turned through its lock-to-lock range, while the outputs of steer angle, pad displacement, camber and caster inclinometers are recorded.

For computation of steering ratio both least-squares and trigonometric equations are fit to the data, to describe the basic linear or nonlinear ratio and the effects of column universal joints. Steer angles of "inside" and "outside" wheels are plotted against handwheel angle, along with Ackerman angles, and Ackerman errors are computed.

Static camber, caster, and steering axis inclinations are computed from the equations presented in Reference 4. Caster is computed by variation of the camber transducer output with steer, and steering axis inclination is computed from variation of the caster transducer output with steer. Steering axis offsets at ground level can be computed from the rotation and translation of the wheelpad or the turntable center, provided that the location of the tire is known with sufficient accuracy. Wheelcenter displacements and steering axis offsets at spin axis level can be computed from wheel follower measurements along with wheel rim offset measurement.

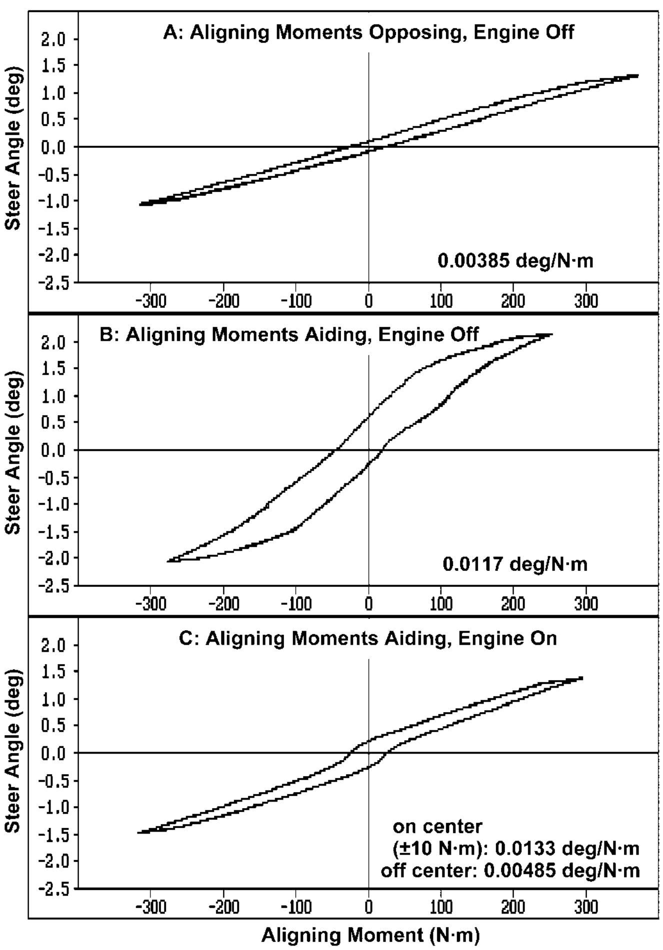

Figure 22: Steer Compliance

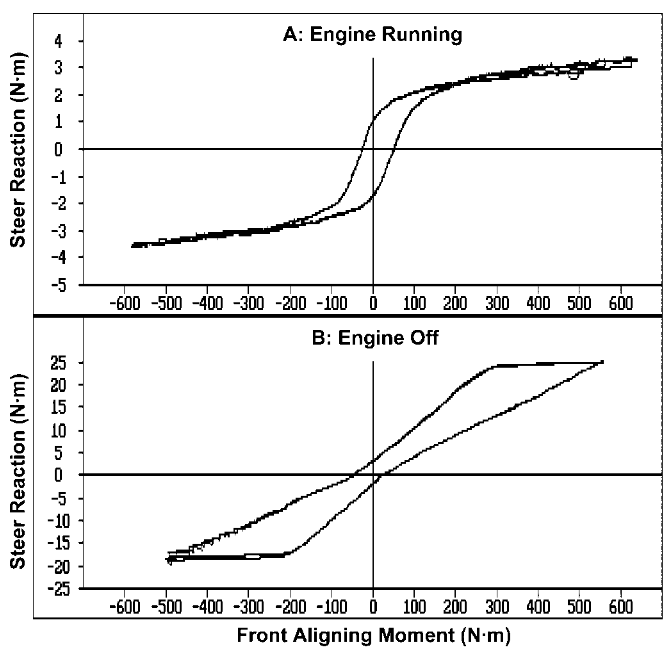

Figure 23: Torque reaction at steering wheel

COMBINED KINEMATICS AND COMPLIANCE TESTING - Topsy's mechanization of kinematics and compliance tests are essentially independent. For that reason "partial derivative" testing - varying body motion or wheel loading at constant tire patch forces or moments or *vice-versa* - is simple. For simultaneous variation of kinematics and compliance input variables it is necessary to burn an EPROM with the control program for each servo and clock their addresses in unison - the electronics hardware required has been developed in another project [5].

SHOCKS AND STRUTS - Shocks, springs, and struts are measured on the vertical baseplate by simple restraining fixtures containing a load cell and a servo cylinder. The cylinder is driven at selected constant rates by a triangle-wave generator. A typical spring-rate result is shown in Figure 24 for a gas-pressurized strut. Determination of one point of a damping curve is shown in Figure 25. At each constant velocity the force is a function of both piston velocity and compression/expansion of the pressurized gas. Damping values are taken at zero deflection.

FREQUENCY RESPONSE METHODS - Frequency response methods (in present software development) utilize the servo cylinders of 1.3 or 2.6 cm^2 area, fitted with load cells, in the ride/roll crossmembers to generate sinusoidal ride forces and roll moments. The cylinders produce continuous off-resonance 2.5 - 5 cm peak-to-peak motion to 3 times resonant frequency. The software computes sprung mass inertial parameters from measured forces, accelerations, and displacements.

Measurements on the unsprung masses are more difficult, because of tire variations. A tire-supported wheel hub can be oscillated vertically to 30 Hz, and the static spring rate of the combination of tire and suspension spring is measured in kinematics testing. However, non-rolling and rolling tires are significantly different [6], and even the non-rolling tire spring rate is known to double between 0 and 30 Hz [7], so the "easy way" appears to have little value. If the tire is eliminated, the vehicle weight must be carried by bias pressure in the hydraulic cylinders. One system under consideration uses two servo cylinders supporting a beam attached to the wheelhub, replacing the wheel and tire.

Steering system dynamics can be studied by applying sinusoidal steer moments to the wheelpads, up to at least 30 Hz. The handwheel can be fixed with its torque measured by the "second steering wheel"; or it can be free with steering wheel motion measured by a rate gyro or angular accelerometer. The relationship between wheel angles and handwheel can be measured, but the nonrolling tire precludes moment-input relationships.

Some lateral tire dynamics can be estimated simply from lateral tire spring rates as measured in compliance tests, as described in References 8 and 9.

SPECIFYING AND REVIEWING A TEST PROGRAM - Specification of a test program consists of selecting the data required and the form in which it is desired. Identification codes are given in compressed form in Appendix 2. A test specification consists circling the appropriate codes in the complete version contained in Reference 10.

The primary record for each test program is a series of two-hour videotapes. From this record all other data can be generated.

For all test runs raw data (vs time) is supplied to the customer on floppy disks, in his choice of formats. Some or all plotting and curve fitting can be done by the customer to save costs, or it can be done by Heitz Chassis Lab. All data is referenced to hours, minutes, and seconds on the test videotapes, so an engineer can "watch" the test run from which each data plot came.

A typical test program will generate much more data than is actually used. The excess consists of deliberate redundancies in which a parameter is measured in different ways for comparison, and of "checking data" - hydraulic cylinder pressures as well as forces; various vehicle or component deflections; etc. This information is generated for

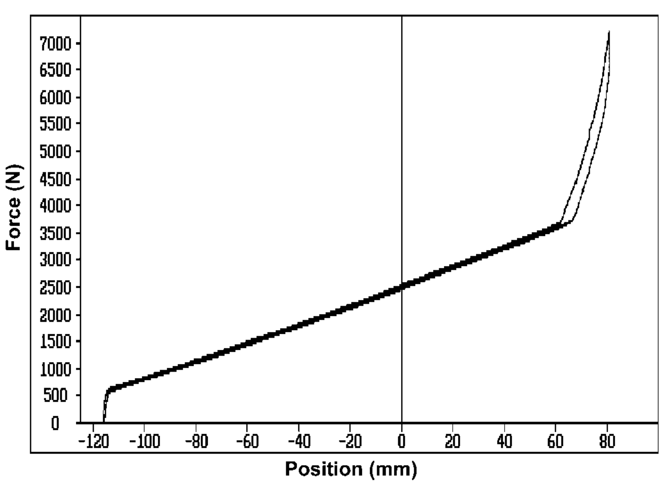

Figure 24: Strut spring rate

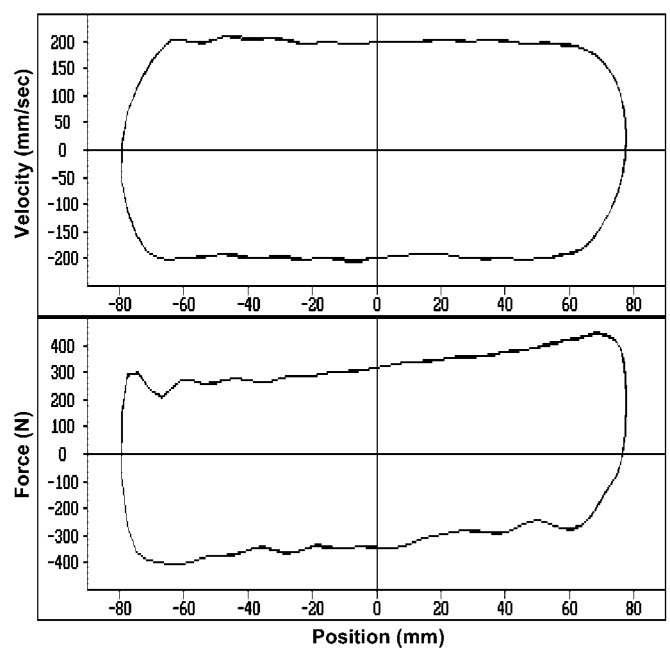

Figure 25: Strut damping

observation during a test, and is recorded under the adage "It is better to have it and not need it than to need it and not have it." This "excess data" remains available for review if questions arise, and is supplied on floppy disk at customer request.

The voice commentary, miscellaneous sounds of the test, and video camera view, all in synchronism with the instrumentation data, are additional forms of data which remain available for reviewing the testing procedures.

SUMMARY

This paper has described a compact laboratory for measurement of the inertial, kinematic, and compliance parameters required to model automobile control responses. The laboratory is oriented toward research rather than production. The approach is modular, based on an "infrastructure" of multipurpose components which are combined temporarily with a few special-purpose fixtures as required for each type of test. The modular approach permits addition of new capabilities without extensive redesign. The authors believe that in some instances the flexibility afforded by the modular approach may allow test capabilities not possible in "production" facilities. When not in use the facility is stored, so that its space is available for other projects.

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APPENDIX 1: DERIVATION OF EQUATIONS FOR CG CALCULATION

Figure A1 shows the table/vehicle assembly in untilted and tilted configurations. For the tilted case, taking moments about the knife edge, we obtain:

$$\begin{aligned} FR\cos\theta = &\; [W_v H_v \tan\theta \cos\theta] \\ &+ [W_v (Y + Y_s)\cos\theta] \\ &+ [W_t H_t \tan\theta\cos\theta] + [W_t Y_t \cos\theta] \end{aligned} \qquad \text{(A1)}$$

Dividing both sides by cosθ and solving for ($W_v H_v \tan\theta$),

$$\begin{aligned} W_v H_v \tan\theta = &\; FR - W_v Y - W_v Y_s \\ &- W_t H_t \tan\theta - W_t Y_t \end{aligned} \qquad \text{(A2)}$$

Differentiating, noting that Wv, Wt, Yt, Yv, R, Hv, Ht are constant, while θ, F, Ysh vary;

$$W_v H_v\, d\tan\theta = R\,dF - W_v\, dY_s - W_t H_t\, d\tan\theta \qquad \text{(A3)}$$

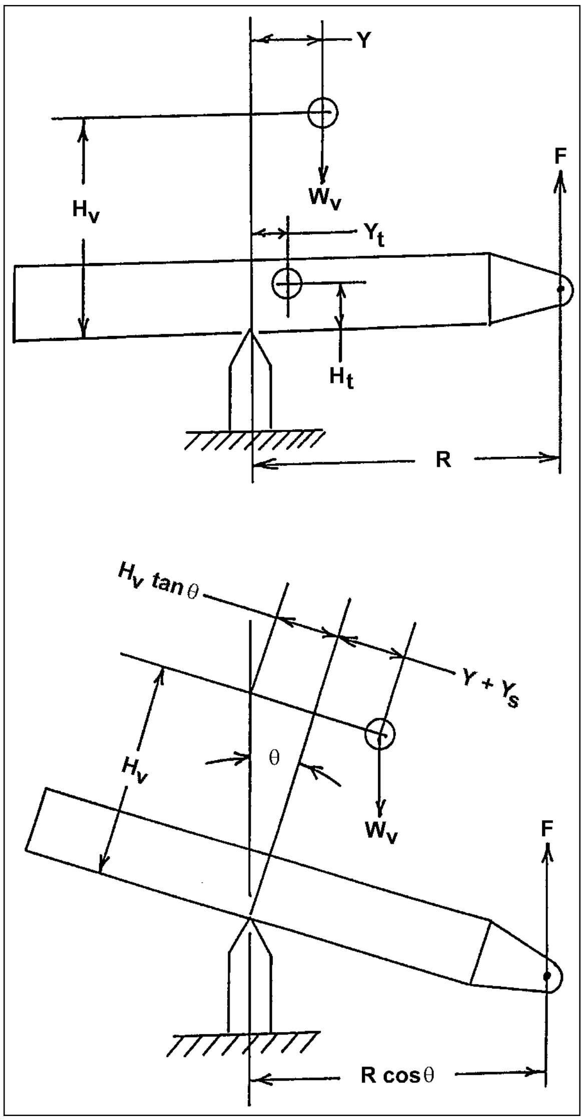

Figure A1: CG Geometry

Rearranging to solve for Hv,

$$H_v = \frac{R}{W_v}\frac{dF}{d\tan\theta} - \frac{W_v}{W_v}\frac{dY_s}{d\tan\theta} - \frac{W_t H_t}{W_v} \qquad \text{(A4)}$$

For the table alone, denoting the force by F';

$$F'R\cos\theta = W_t H_t \sin\theta + W_t Y_t \cos\theta \qquad \text{(A1a)}$$

$$F'R = W_t H_t \tan\theta + W_t Y_t \qquad \text{(A2a)}$$

$$W_t H_t \tan\theta = F'R - W_t Y_t \qquad \text{(A3a)}$$

Differentiating as above and rearranging;

$$H_t = \frac{R}{W_t}\frac{dF'}{d\tan\theta} \qquad \text{(A4a)}$$

Substituting this value for Ht into equation (A4);

$$H_v = \frac{R}{W_v}\left[\frac{dF}{d\tan\theta} - \frac{W_t}{W_t}\frac{dF'}{d\tan\theta}\right] - \frac{dY_s}{d\tan\theta} \qquad \text{(A5)}$$

H_v, the height of the CG above the knife edge is thus found from the slopes of the total assembly and table alone force/tilt angle relationship, plus that of the lateral vehicle displacement with respect to the table. The weight of the table itself drops out of the equation: however, the force/tilt angle relationship for the table alone must be measured with all ancillary equipment such as locating screwjacks and potentiometers in the same locations as for the vehicle test.

CORRECTION TO PRIMARY DATA - In the derivation of equation (A5) the effect of angularity in the vertical actuator is ignored. This angularity does cause a small error in the slope dF/d(tan θ) for positive and negative tilt angles, with the slope being lower in the "lifting" or "raising" direction. The resulting cosine error is 0.005 percent, but sine error is in the order of 0.1 percent. They are computed and corrected as follows.

Figure A2 shows a shortening of the torque arm, R(1-cos θ), which together with the actuator length (L - R sin θ) forms the angle ϕ. This angle amounts to 0.56 degrees in the raising direction and 0.45 degrees in the lowering direction.

The resulting cosine error, which is always in the same direction, is corrected by computing ϕ and multiplying the load cell output signal by (cosine ϕ). However, correction of the resulting <u>sine</u> error is a little more complex.

In Figure A2 it can be seen that in the raised position the actuator is in compression, producing a clockwise moment equal to (FR cos θ) about the knife edges; while the sine component produces a moment (F sin ϕ) x (R sin θ) which opposes that moment.

In the lowered position the actuator is in tension, producing a counterclockwise moment around the knife edges. However, in this case the sine component produces an <u>added</u>, counterclockwise moment.

The correction for the moment (FR $\sin\phi$ $\sin\theta$) is made as follows. For small enough angles sine and tangent can be assumed equal, so that:

$$F\sin\phi = F\tan\phi = \frac{FR(1-\cos\theta)}{L - R\sin\theta} \quad \text{(A6)}$$

and the moment,

$$FR\sin\phi\sin\theta = FR^2\sin\theta \, \frac{1-\cos\theta}{L - R\sin\theta} \quad \text{(A7)}$$

In equation (A1) the moment FR cos θ must be corrected by (A7). Dividing through by (R cos θ), we get a corrected value for force F.

$$F_{corr} = F\left[1 + \frac{R(1-\cos\theta)}{L - R\sin\theta}\tan\theta\right] \quad \text{(A8)}$$

With this correction, and with the gas tank filled to overflow to limit fuel slosh, the difference in slopes between positive and negative tilt angles is eliminated.

CG ERROR ANALYSIS - The measurement of CG height is seen from equation (A5) to consist of four separate measurements: the force/angle slopes with and without the vehicle, the lateral deflection/angle slope, and the "road" height above the knife edges. The maximum error in the resultant CG height is therefore found as the weighted sum of the four individual measurement errors. The weighted error contribution from each source is found as the percentage error in that measurement, multiplied by the contribution of the measurement to the net CG height. The percentage errors in the individual measurements is found by differentiation, as follows.

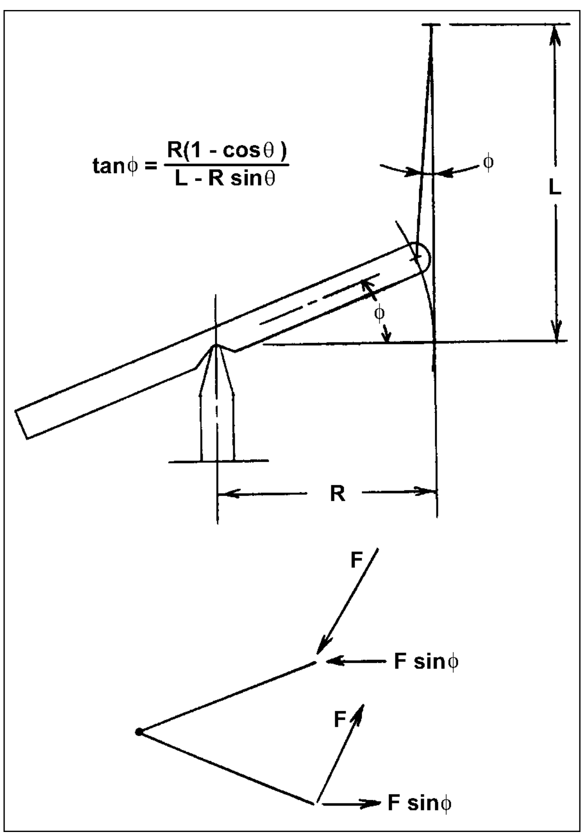

Figure A2: Sine Error

For the contribution to total error of table plus vehicle, from equation A5,

$$\frac{R}{W_v}\frac{dF}{d\tan\theta} \quad \text{(E1)}$$

At the end points, for "small angles", this equation becomes:

$$\frac{R}{W_v}\frac{F}{\theta} \quad \text{(E2)}$$

Differentiating,

$$dH_v = \frac{R}{W_v}\frac{dF}{d\theta} + \frac{R}{W_v}\frac{Fd\theta}{\theta^2} + \frac{dR}{W_v}\frac{F}{\theta} + \frac{RdW_v}{W_v^2}\frac{F}{\theta} \quad \text{(E3)}$$

Multiplying and dividing by $\frac{R}{W_v}\frac{F}{\theta} = \frac{R}{W_v}\frac{dF}{d\theta}$

$$(\delta H_v)_{v+t} = \frac{R}{W_v}\frac{dF}{d\theta}\left[\frac{\delta F}{F} + \frac{\delta\theta}{\theta} + \frac{\delta R}{R} + \frac{\delta W_v}{W_v}\right] \quad \text{(E4)}$$

Similarly, for the table alone,

$$(\delta H_v)_t = \frac{R}{W_v}\frac{dF_t}{d\theta}\left[\frac{\delta F_t}{F_t} + \frac{\delta\theta}{\theta} + \frac{\delta R}{R} + \frac{\delta W_v}{W_v}\right] \quad \text{(E5)}$$

For lateral deflection,

$$(\delta H_v)_y = \frac{dY_s}{d\theta}\left[\frac{\delta Y_s}{Y_s} + \frac{\delta\theta}{\theta}\right] \quad \text{(E6)}$$

And for the road - knife edge distance

$$(\delta H_v)_h = \delta H_v \quad \text{(E7)}$$

The total error is

$$\sigma H_v = (\sigma H_v)_{v+t} + (\sigma H_v)_t + (\sigma H_v)_y + (\sigma H_v)_h \quad \text{(E8)}$$

The individual percentage errors are found from the full-scale linearity-plus-hysteresis specifications of all transducers, along with the measured end-point parameter values (to obtain percent-of-reading), and measurement resolution.

The total error is dominated by the $(\sigma Hv)v+t$ term and by the "roadway height" over the knife edges, because their "weight" in determination of Hv is greatest. These errors are minimized by calibrating the combination of load cell force and moment arm together, and by measuring the height of the "road surface" over the knife edges directly under the front and rear axles, before each test. In the latter measurement the deflection curve of the simply-supported reference beam under its own weight (1.0 mm at center span) is included in the measurement.

The maximum errors calculated from equation (E8), with all errors stacked up in the same direction, are in the order of 0.6 percent. Rms errors are in the order of 0.3 percent. We believe these estimates to be realistic, because the individual calibrations encompass the testing range; the quantity (Ft x R) is actually a single calibration; hysteresis error is minimized by running three completely reversed test cycles; and resolution error is minimized by taking the best straight line through the resulting 60 000 data points.

In practice we believe the greatest error is most likely to be the setting of ride height, because front and rear ride heights are seldom specified with precision.

APPENDIX 2: SPECIFICATION OF KINEMATICS AND COMPLIANCE TESTS

STANDARD CONDITIONS

- All tests are run at, or referenced to, curb trim height.
- All kinematics tests are run with engine stopped.
- Steering system tests are run with engine running and with engine stopped. Only handwheel torque is plotted for the engine stopped condition.
- Front lateral force and aligning torque compliances with forces or torques adding are run with engine running; and with forces and torques opposing the engine is stopped.
- Front longitudinal force compliance tests are run with engine running.
- All rear compliance tests are run with engine stopped.

EQUATIONS FITTED TO PLOTTED DATA - Third-order least-squares polynomial equations are fitted to most data plots. Where dictated by the character of the data, three-element linear plots (through-center, and positive and negative extremes) are used instead. Examples of the latter are steer tests, where roadwheel steer angle vs aligning torque usually shows a distinct difference between on-center and off-center slopes; and wheel loads in ride and roll, where rates change as bump or rebound stops are engaged.

CHOICE OF DATA TO BE PLOTTED - Raw data (vs time) required for all plots will be supplied on floppy disks for all tests, and will be included in the test charge. All of the plots in the following listing can be generated from the floppy disk format. Since many of the data plots in the listing are useful only in some simulation models, and the quotation for Data Processing includes a separate charge for each plot, costs will be minimized by circling only those plots actually intended for use, or those for which fitted equations are desired.

ORGANIZATION OF DATA PLOTS - Data plots are organized into four groups:

Test type and configuration
- Non-standard test conditions if present
  - Test
    - Parameter plotted

Examples of non-standard test conditions are engine running or stopped, non-curb ride heights, etc. For example, a front compliance test at 30 mm jounce instead of curb height would be coded as FC-30J-SFA, etc.

IDENTIFICATION CODES FOR PLOTTED DATA

STEERING TESTS

FS: FRONT STEER, with engine running, except HTS

- RH: Roadwheel angles vs handwheel angle
  - SAL Left roadwheel angle vs handwheel angle
  - SAR Right roadwheel angle vs handwheel angle
  - SAC Left and right side compared (overlaid)
  - AVG Average of left and right sides
  - OSR Overall steering ratio vs handwheel angle
  - DIO Difference between inside and outside wheel vs handwheel angle, with theoretical Ackerman
  - AED Ackerman error in degrees vs handwheel angle
  - AEP Ackerman error in percent vs handwheel angle
  - HTR Handwheel torque vs handwheel angle with engine running
  - HTS Handwheel torque vs handwheel angle with engine stopped
- RW Parameters vs Roadwheel angles
  - DIO Difference between inside and outside wheel angles vs inside wheel angle, with theoretical Ackerman
  - AED Ackerman error in degrees vs inside wheel angle
  - AEP Ackerman error in percent vs inside wheel angle
  - CAL/CAR Camber angle vs wheel angle
  - CCL/CCR Caster change vs wheel angle
  - DXL/DXR, DYL/DYR, DZL/DZR Deflections vs wheel angle

COMPLIANCE TESTS

FC: FRONT COMPLIANCES

- SFA: SIDE FORCES ADDING (in parallel)
- SFO: SIDE FORCES OPPOSING
- LFS: LONGITUDINAL FORCES, SYMMETRICAL
- LFL: LONGITUDINAL FORCES, LEFT SIDE ONLY
- LFR: LONGITUDINAL FORCES, RIGHT SIDE ONLY
- ATA: ALIGNING TORQUES ADDING
- ATO: ALIGNING TORQUES OPPOSING
  - FT/MT Applied force/moment vs time
  - SAL/SAR Steer angle vs applied force
  - CAL/CAR Camber vs applied force
  - CCL/CCR Caster change vs applied force
  - DXL/DXR, DYL/DYR, DZL/DZR Deflections vs applied force
  - HRT Handwheel reaction torque vs applied force
- SFO: RCL, RCR Wheel load change vs applied force, for roll center height
- LFS: RCL, RCR Wheel load change vs applied force, for anti-dive angle

RC: REAR COMPLIANCES

- SFA: SIDE FORCES ADDING, (in parallel)
- SFO: SIDE FORCES OPPOSING
- ATA: ALIGNING TORQUES ADDING (parallel steer)
- ATO: ALIGNING TORQUES OPPOSING
- LFS: LONGITUDINAL FORCES, SYMMETRICAL
- LFL: LONGITUDINAL FORCES, LEFT SIDE ONLY
- LFR: LONGITUDINAL FORCES, RIGHT SIDE ONLY
  - FT/MT Applied force/moment vs time
  - SAL/SAR Steer angle vs applied force/moment
  - CAL/CAR Camber vs applied force/moment
  - CCL/CCR Caster change vs applied force/moment
  - DXL, DXR, DYX, DYR, DZX, DXR Deflections vs applied force/moment

RC - SFO: RCL,RCR Wheel load change vs applied force, for roll center

RC - LFS: RCL,RCR Wheel load change vs applied force for anti-dive angle

RIDE AND ROLL SPRING RATES AND KINEMATICS

RID: RIDE spring rates and kinematics

- FSR: FRONT SPRING RATES IN RIDE
- FSR: REAR SPRING RATES IN RIDE
  - DZT Wheel travel with respect to sprung mass vs time, left and right sides
  - WLL/WLR Wheel load vs wheel travel
  - TZT Tire travel (spin axis reference point with respect to ground) vs time, left and right sides
  - TLL/TLR Wheel load vs tire travel
- FKN: FRONT KINEMATICS in ride
  - SAL/SAR Steer angle vs wheel travel
  - CAL/CAR Camber angle vs wheel travel
  - CCL/CCR Caster change vs wheel travel
  - DXL, DXR, DYX, DYR, DZL, DZR Deflections vs wheel travel
  - PXL/PXR Deflection of wheelpad with respect to Frame, vs wheel + tire travel
  - PYL/PYR Deflection of wheelpad with respect to Frame, vs wheel + tire travel
  - STL/STR Shock absorber travel vs wheel travel

ROL: ROLL SPRING RATES AND KINEMATICS

- FRR: FRONT ROLL RATES
- RRR: REAR ROLL RATES
  - DZT Wheel travel with respect to sprung mass, left and right sides, vs time
  - SRA Suspension roll angle vs time, left and right sides
  - WLL/WLR Wheel loads vs suspension roll angle
  - RM Roll moment vs suspension roll angle
- CRR: COMPOSITE ROLL RATES
  - VRM Vehicle roll moment (total roll moment) vs suspension roll angle
  - RSD Percent of vehicle roll moment in front vs suspension roll angle

ROL - FK: FRONT KINEMATICS IN ROLL

ROL - RK: REAR KINEMATICS IN ROLL

- SAL/SAR Steer angle vs suspension roll angle
- CAL/CAR Camber angle vs suspension roll angle
- CCL/CAR Caster change vs suspension roll angle
- DXL/DXR, DYL/DYR Deflections vs suspension roll angle
- TZT Tire travel vs time, left and right sides
- DZT Total travel, suspension plus tire, vs time, left and right sides
- VRA Vehicle roll angle with respect to ground vs time, left and right sides
- PXL/PXR, PYL/PYR X and Y Deflections of wheelpad with respect to Frame, vs vehicle roll angle
- STL/STR Shock absorber travel vs roll angle