Highway Effects on Vehicle Performance

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FOREWORD

This report presents an overview of the efforts to develop a convenient procedure to simulate operation of motor vehicles on highways of an arbitrary configuration and to estimate fuel consumption and exhaust emissions resulting from reasonable operations of those vehicles.

Highway pavements, grades, curves, and wind and traffic flow rates affect the fuel consumption and air contaminant emission rates for a given section of highway or a network of highways. Vehicles were tested on a large-roll dynamometer and under various road and traffic flow conditions. Evaluations of other analytical and experimental results were also made. Based on experiments and evaluations, clear relationships were developed relating specific loads and speeds to pollutant emissions and fuel consumption rates. These data were used to develop a user-friendly personal computer program called the Vehicle/ Highway Performance Predictor Algorithm. This model is intended to receive any reasonable mix of data for a selection of various vehicles that may approximate the traffic mix for given locations for past, current, and reasonable future years. The procedure can be used by highway planners and designers, environmental engineers, and traffic engineers, particularly those involved in Intelligent Transportation Systems, to evaluate local microspace air quality evaluations or larger area air pollution emission rates to determine impacts and conformity to the State Implementation Plans for the areas. This model is intended to use modal emissions and fuel usage rates that are based on various speeds and loads of vehicles in operation.

This report reviews the principles involved in determining the external loads on vehicles from longitudinal and lateral accelerations, aerodynamic drag, rolling resistance, and various grades. Examples of loads measured in the field and related dynamometer tests for selected vehicles for fuel consumption and air contaminant emissions are provided.

Detailed data have not been archived. However, informal interim reports containing added experimental information are available from the Federal Highway Administration Offices of Natural Environment, Infrastructure Research and Development, Traffic Operations Research and Development, and four Federal Highway Administration Resource Centers.

T. Paul Teng
Director, Office of Infrastructure
Research and Development, P.E.

NOTICE

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16. Abstract
A user-friendly model for personal computers, " Vehicle/Highway Performance Predictor," was developed to estimate fuel consumption and exhaust emissions related to modes of vehicle operations on highways of various configurations and traffic controls for highway designers and planners and strategists optimizing Intelligent Transportation Systems. This model simulates operations of vehicles by evaluations of the vehicle external loads or propulsive demands, which are determined by longitudinal and lateral accelerations, positive and negative road grades, rolling resistance, and aerodynamic drag for various transmission gears. The computations of fuel consumption and air pollutant emissions are then related to the vehicle maps of fuel consumption and air contaminant emission rates as evaluated from large-roll dynamometer measurements for vehicle operations under various loads, speeds, and transmission gears or as may be estimated based on engine maps, speeds, loads, and vehicle drive-train characteristics.

The supportive experimental program for the model development showed the following:
1) Propulsive or external loads imposed by highway features such as curves and grades can be measured and simulated on a large-roll chassis dynamometer and are predictable, but the dynamometer simulation is not needed if the vehicle fuel consumption and air contaminant emission rates vs. total propulsive demand are used to create a vehicle data base.
2) Driveshaft torque measured on an instrumented vehicle showed that, for steady speeds on flat highway, the road load is a quadratic function of speed, a quadratic function of lateral acceleration, and a linear function of positive or negative grades or positive or negative longitudinal accelerations.
3) For one tested vehicle, the energy loss in drive axle and tires was described well as a loss of tractive force expressed as a quadratic function of only the total drive torque or tractive force.
4) A prototype method was developed to compare rolling resistance of pavement surfaces based on known vehicle speeds, road grades, and total external load indicated by driveshaft torque.

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SI* (MODERN METRIC) CONVERSION FACTORS

APPROXIMATE CONVERSIONS TO SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
in	inches	25.4	millimeters	mm
ft	feet	0.305	meters	m
yd	yards	0.914	meters	m
mi	miles	1.61	kilometers	km
		AREA		
in^2	square inches	645.2	square millimeters	mm^2
ft^2	square feet	0.093	square meters	m^2
yd^2	square yards	0.836	square meters	m^2
ac	acres	0.405	hectares	ha
mi^2	square miles	2.59	square kilometers	km^2
		VOLUME		
fl oz	fluid ounces	29.57	milliliters	mL
gal	gallons	3.785	liters	L
ft^3	cubic feet	0.028	cubic meters	m^3
yd^3	cubic yards	0.765	cubic meters	m^3
NOTE: Volumes greater than 1000 l shall be shown in m^3.				
		MASS		
oz	ounces	28.35	grams	g
lb	pounds	0.454	kilograms	kg
T	short tons (2000 lb)	0.907	megagrams (or "metric ton")	Mg (or "t")
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5(F-32)/9 or (F-32)/1.8	Celcius temperature	°C
		ILLUMINATION		
fc	foot-candles	10.76	lux	lx
fl	foot-Lamberts	3.426	candela/m^2	cd/m^2
		FORCE and PRESSURE or STRESS		
lbf	poundforce	4.45	newtons	N
lbf/in^2	poundforce per square inch	6.89	kilopascals	kPa

APPROXIMATE CONVERSIONS FROM SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
mm	millimeters	0.039	inches	in
m	meters	3.28	feet	ft
m	meters	1.09	yards	yd
km	kilometers	0.621	miles	mi
		AREA		
mm^2	square millimeters	0.0016	square inches	in^2
m^2	square meters	10.764	square feet	ft^2
m^2	square meters	1.195	square yards	yd^2
ha	hectares	2.47	acres	ac
km^2	square kilometers	0.386	square miles	mi^2
		VOLUME		
mL	milliliters	0.034	fluid ounces	fl oz
L	liters	0.264	gallons	gal
m^3	cubic meters	35.71	cubic feet	ft^3
m^3	cubic meters	1.307	cubic yards	yd^3
		MASS		
g	grams	0.035	ounces	oz
kg	kilograms	2.202	pounds	lb
Mg (or "t")	megagrams (or "metric ton")	1.103	short tons (2000 lb)	T
		TEMPERATURE (exact)		
°C	Celcius temperature	1.8C + 32	Fahrenheit temperature	°F
		ILLUMINATION		
lx	lux	0.0929	foot-candles	fc
cd/m^2	candela/m^2	0.2919	foot-Lamberts	fl
		FORCE and PRESSURE or STRESS		
N	newtons	0.225	poundforce	lbf
kPa	kilopascals	0.145	poundforce per square inch	lbf/in^2

* SI is the symbol for the International System of Units. Appropriate rounding should be made to comply with Section 4 of ASTM E380.

(Revised September 1993)

PREFACE

A user-friendly model for personal computers, "Vehicle/Highway Performance Predictor"(HPP), was developed for highway designers and planners and strategists to estimate fuel consumption and exhaust emissions related to modes of vehicle operations on highways of various configurations and traffic controls, e.g. the optimization of Intelligent Transportation Systems with considerations for fuel consumption and air pollution impacts. This model simulates operations of vehicles by evaluating the vehicle external loads or propulsive demands determined by longitudinal and lateral accelerations, positive and negative road grades, rolling resistance, and aerodynamic drag for various transmission gears. The resultant computations of fuel consumption and air pollutant emissions are then related to the vehicle maps of fuel consumption and air contaminant emission rates as evaluated from large-roll dynamometer measurements for vehicle operations under various loads, speeds, and transmission gears or as may be estimated based on engine maps, speeds, loads, and vehicle drive-train characteristics.

The supportive experimental program for the model development showed that:

1) Propulsive or external loads imposed by highway features such as curves and grades can be measured and simulated on a large-roll chassis dynamometer and are predictable, but the dynamometer simulation is not needed if the vehicle fuel consumption and air contaminant emission rates vs. total propulsive demand are used to create a vehicle data base.

2) Driveshaft torque measured on an instrumented vehicle showed that, for steady speeds on a flat highway, the road load is a quadratic function of speed, a quadratic function of lateral acceleration, and a linear function of positive or negative grades or positive or negative longitudinal accelerations.

3) For one tested vehicle, the energy loss in the drive axle and tires was described well as a loss of tractive force expressed as a quadratic function of only the total drive torque or tractive force.

4) A prototype method of comparing rolling resistance of pavement surfaces based on known vehicle speeds, road grades, road curves, and total external load indicated by driveshaft torque could be developed.

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1.0 PROJECT DEFINITION

1.1 PROJECT OBJECTIVES

This project was initiated at the John A. Volpe National Transportation Systems Center (VNTSC) by the Federal Highway Administration (FHWA) in 1978. The main objective was to develop an easily - used calculation procedure with which highway designers could estimate the effects of highway geometrical design features on vehicle performance. Secondary goals included supplying the FHWA with updated operating parameters on modern vehicles (i.e., 1980s vintage). These data included fuel economy, exhaust emissions, and other pertinent information for various road conditions and vehicle operating modes. The new information is intended to update references such as Claffey (1), which had been published in the early 1970s on vehicles manufactured in the late 1960s.

The need for this study arose largely from the drastic increases in automotive fuel prices in the mid-1970s. Prior to that time, fuel was plentiful and cheap; highways were designed principally with concern for safety, durability, and cost of construction and maintenance. As fuel costs became a significant component of operating expense, highway designers needed a practicable means of estimating, with reasonable accuracy, the impact of highway design features on fuel consumption. The designer then could compare the relative fuel costs and air pollutant emissions of alternative designs for a particular highway and relate the difference in fuel costs and air pollutant emissions to the differences in construction costs, i.e., develop a cost/benefit ratio.

However, meaningful estimates of fuel consumption and air pollutant emissions require information about specific characteristics of automotive vehicles and more knowledge of automotive engineering than most highway designers could be expected to have. The FHWA anticipated that the requisite knowledge of automotive technology could be incorporated into the calculation procedure and thus relieve the highway designer of this responsibility; the designer then could concentrate on his or her specialty, design and cost analysis of the highway.

1.2 HIGHWAY FACTORS AFFECTING FUEL ECONOMY

The factors affecting vehicle fuel economy that fall within the province of the highway designer include principally (a) highway geometry and structure and (b) vehicle operation (speed vs. distance) as influenced by highway geometry, traffic controls, and highway surroundings. For a given vehicle, accelerations (i.e., increases in longitudinal speed) can impose the largest demands in fuel flow rate; the second-largest influence on instantaneous fuel rate is road grade (longitudinal slope). Important secondary considerations applicable to each of these operational factors often are overlooked: for accelerations, the percentage of total operating time that is spent accelerating and the relative magnitude of acceleration rates, and for grades, both the degree of slope and (most important) whether the same change in elevation is to be accomplished on each of two or more different grades. These matters are addressed separately later in this report.

After grades, the second geometrical feature of highways that most affects fuel economy and air pollutant emissions is horizontal alignment or curvature. If the road must change direction, the only controls left to the designer are radius of curvature, superelevation (transverse "banking"), and design speed. Generally, it is desirable to avoid or minimize speed changes because these tend to decrease fuel economy and can cause safety problems. Aside from keeping the curve radius long enough to provide safe driving, how much does curve radius influence fuel economy and air pollutant emissions, are the relationships predictable, and can they be generalized over a range of vehicles? This project demonstrated that the effects of curve radius and speed can be measured, are mathematically continuous over a wide range of both parameters, and can be generalized for a range of automobile types.

Road surface roughness is another highway "feature" that can affect fuel economy and air pollutant emissions. The information gathered on this subject during this project indicated that most roads carrying any significant volume of traffic probably would be resurfaced for reasons of safety and user comfort before the roughness influence on constant-speed fuel economy approached about 4 to 5 percent on the specific patches of roughness. Note that, if a rough spot causes drivers to slow down (especially by braking) and then to accelerate, this speed change can increase fuel consumption; however, as will be shown, this effect can be calculated in terms of the speed change. This conclusion was reached also in a recent report (2).

The manner in which a particular vehicle is operated on a given highway can change materially the fuel economy attained by that vehicle. Speed, and changes in speed, exert a clear influence on fuel economy. The highway designer can affect the operating speed of the majority of traffic by changes in highway geometry (horizontal and vertical curvature, number and width of lanes, superelevation of curves), traffic controls (speed limit signs, traffic lights and stop signs), and the general surroundings of the road (intersections, entrance/exit ramps, shoulder width, clearing to provide visibility around curves and beyond grade changes, and roadside development).

Beyond these factors over which the highway designer has some control, variations in the behavior of different drivers broaden the range of fuel economy values that can be expected from a given vehicle. This was demonstrated, at constant speed on open highways, by a limited experimental effort under this project. Therefore, it was concluded that the purposes of this project did not demand the highest degree of fidelity in reproducing the fuel consumption characteristics of a particular vehicle that had been tested; those test parameters would be influenced by various amounts in the hands of different drivers. Rather, the highway performance calculations should reflect with reasonable accuracy the relative effects of different highway design features on the performance of the vehicles analyzed.

1.3 PROJECT REQUIREMENTS

The major requirements of this project included the following:

- Devise a computational algorithm to estimate realistically the fuel consumption of typical vehicles when operated over highways of arbitrary configuration. This was the principal objective of this project.

  In support of the principal objective, accomplish the following:

- Perform a preliminary assessment of the technical feasibility of the project's major objective; i. e., determine the likelihood that a computation method could be developed that would provide useful estimates of vehicle fuel economy by simulating realistic operation of vehicles on highways of arbitrary configuration.

- Demonstrate that the effects of principal highway geometrical features (grades and curves) on a vehicle's propulsion system can be measured and are independent of vehicle parameters except for known physical characteristics (such as weight, frontal area, drag coefficient, drive train details, etc.).

- Demonstrate that the above effects can be calculated for different vehicles, provided that the necessary physical characteristics of the vehicles are known.

- Demonstrate that a large-roll chassis dynamometer ("dyno") can apply loads to a vehicle under test to appropriately simulate desired highway features; and, show the extent to which such simulation is required to meet the objectives of this project.

- Develop methods for measuring the requisite operational parameters of vehicles to permit preparation of a vehicle data base.

- Procure and/or develop instrumentation necessary to support the measurement of needed vehicle parameters.

- Test a small number of modern (1980s-era) vehicles and assemble a prototype data base that describes the fuel consumption and exhaust emissions of these vehicles as functions of driveshaft torque[1] and such other parameters as would be found necessary. (Exhaust emissions were deleted from data requirements for some of the later tests.)

- During vehicle testing, obtain measured operating data for modern vehicles to update existing references giving similar data on older vehicles. Parameters would include; e.g., fuel economy vs. constant speeds, on grades, on curves, while idling, and during accelerations and decelerations—or, under load conditions simulating these. Such data would be obtained to the extent compatible with other project objectives.

[1] Driveshaft torque or torque is related to force or load on vehicle from road loads, accelerations, grades and curves and has units of Newton meters. For further discussion see Appendix A - A.1.1.7 and Appendix C- C.2.1.

2.0 APPROACH

A fundamental requirement for this project was to measure the propulsive demands imposed on a vehicle while that vehicle was in operation on open roads and/or other outdoor test facilities. This necessitated measurement of vehicle loads from within the vehicle rather than by an external means, and recording of those data by a system on board the vehicle. It was intended that the instrumentation system would be transferred from vehicle to vehicle; thus, it was necessary to choose a measurement location that afforded external access to the measured part.

Perhaps the optimum test locus to sense only propulsive loads on the vehicle was in the drive train between transmission and drive wheels. This would measure both torque and rotational speed at the same location; torque would indicate tractive force, rotational speed would measure drive wheel and road speeds, while the product of torque and rotational speed at the same point would yield propulsive power. Such a test locus would exclude sensing of engine accessory loads and of losses in the transmission. For rear-wheel-drive (RWD) vehicles, only a single measurement system on the drive or propeller shaft would be required. For front-wheel-drive (FWD) vehicles tested with two separate half-shafts driving the two front wheels, instrumentation had to provide dual measurements of both torques and shaft speeds for each half-shaft individually.

A pilot study was conducted to demonstrate the measurability of the effects of grade and horizontal curvature, and the suitability of the large-roll chassis dynamometer (dyno) to simulate such loads. To determine measurability, an existing instrumented test automobile (1975 Dodge Dart, an RWD vehicle) was further equipped with instrumentation and a digital data logger. The auto was operated on private roads with grades to 8 percent and curves with radii from 240 ft to 2000 ft (73 m to 714 m), at speeds up to 60 mi/h (97 km/h). The dyno was used to measure fuel consumption and exhaust emissions over the full normal operating range of the vehicle; the resultant data yielded a prototype data base for this car that demonstrated the simple format required to describe the observed performance.

To develop data on current (1980-1981) production automobiles, new, high-performance digital data acquisition systems and two 1980 model-year autos were procured. After exploratory tests at airports relatively near to VNTSC, the Bangor (ME) International Airport was chosen as the primary test site because of its adequate facilities and the willingness of its management to accommodate the necessary tests. Driveshaft torque was measured at a number of constant speeds on the runway and, at low speeds, on the heavy-duty apron of the airport. Torque on curves was determined by running multiple continuous laps of circles with three different radii, at five speeds on each circle to produce lateral accelerations ranging uniformly up to about 0.5 G. The possible difference in curve effect upon an FWD auto (as compared with the RWD cars) was explored after procurement of a third vehicle, a 1981 model with this type of drive train. During some of these tests, the vehicles were equipped with fuel flowmeters to monitor fuel consumption. On one car, two complete sets of tires, both bias-ply and radial-ply, were tested; the other two cars used only radial-ply tires. Some of the test cars were operated on certain public highways in the Bangor area to evaluate potential highway and traffic effects.

Each of the three vehicles was tested, also, on the dyno over the full operational range of speeds and both positive and negative torques, while fuel consumption was measured either by exhaust-gas analysis (for one car) or fuel flowmeter. The exhaust-gas test procedure also yielded exhaust emissions data over the operational range for that car.

In addition to the principal tests described above, limited experiments were conducted to explore related aspects of vehicle operation on highways and to support the planned development of a performance prediction algorithm. A study of 10 different drivers operating one of the instrumented vehicles on a highway with a constant posted speed limit, but otherwise free to drive as they normally would, assessed the influence of driver variability on fuel economy. One vehicle was operated on the dyno in a series of replicate accelerations and decelerations while exhaust emissions were sampled to determine average fuel consumption over these maneuvers. A very limited field test of the effect of different road surface roughnesses on driveshaft torque was performed.

A computer program (Vehicle/Highway Performance Predictor -HPP) was written that would simulate driving a vehicle over a highway of arbitrary configuration specified by the user; the speed "profile," or variation in speed as a function of distance, also was arbitrary and input by the user. The program calculated the quasi-steady-state net driveshaft torque on each consecutive increment of highway, usually only a few meters in length; for each increment, the net torque was used along with the performance data base for that vehicle to calculate the amount of fuel consumed (and exhaust emissions produced, if data were available). These increments were summed over the entire length of highway. Accelerations (positive or negative) were approximated as a series of short-distance travels, each at a constant but incrementally-different speed and with the torque component required for acceleration superimposed on the constant-speed torque demand.

Performance data bases to support the above computation were available only for the three vehicles completely mapped under this project. However, a data base (vehicle data base for module) for only one vehicle was used in the HPP software. This "fleet" could be augmented by adding vehicles that have had appropriate testing done elsewhere or by synthesizing data bases for additional vehicles by the use of models such as VNTSC's VEHSIM vehicle simulation computer program. VEHSIM requires specification of a vehicle in terms of its operational components and physical characteristics: engine type and displacement, transmission type and performance characteristics, subsequent drive train and components, vehicle weight, frontal area and drag coefficient, etc. Drive cycles could be specified to generate speed and torque conditions that could have been used in dynamometer testing of such vehicles, and fuel consumption rates then calculated. From synthesized data, performance maps could be derived in the same manner as when actual cars are tested.

3.0 MAJOR ACCOMPLISHMENTS AND EXPERIMENTAL RESULTS

The following sections discuss briefly the most significant accomplishments and experimental findings of this project. If the reader desires more information on a particular subject, references are made to later sections of this report.

3.1 COMPUTER PROGRAM FOR PREDICTION OF VEHICLE PERFORMANCE ON HIGHWAYS (HPP)

This project culminated in the development of a computer program that predicts the fuel consumption and air pollutant emission of vehicles when operated on highways of arbitrary length and geometric design. The algorithm considers the effects of realistic driving patterns involving accelerations and decelerations and of operating on roads that have positive and negative grades, and can evaluate the added propulsive demands of horizontal curves. If vehicle data include information on vehicle exhaust emissions in appropriate formats, the program can also estimate the exhaust emissions resulting from the above kinds of vehicle operation. The algorithm operates efficiently and economically; it simulates vehicle operation in piecewise fashion over short increments of road (typically about 50 m/increment), and processes about 2 to 3 mi (3.22 to 4.83 km) of road travel by one vehicle/second of computer operation. The method was implemented on personal computers. The computer program incorporates the necessary level of automotive engineering insights, and the highway designer need input only highway geometry and vehicle speed vs. distance. Alternative configurations of any particular highway can be entered sequentially and vehicles "driven" over each with little interaction by the user. The differences (in fuel consumption and emissions," if data are available) between the optional configurations can provide objective, realistic comparisons of the relative operating costs to the driving public of the alternative highway designs or traffic flows or both as desired.

The computation method is described in detail in Appendix A.

3.2 EFFECT OF ROAD GRADE

In the first vehicle tests under this project, total driveshaft torque was measured in a series of replicate runs on a 4-mi (6.4-km) section of private/ road at speeds of 15 to 60 mi/h (24 to 96 km/h) and on grades up to +8 percent. A linear correlation between driveshaft torque and road grade was found that showed a standard deviation of 2.2 percent. When vehicle-specific factors were used to calculate the increment of torque for a given grade change, the experimental data agreed with calculated values within 1.2 percent. This quality of agreement between experimental data and calculated values corroborated general experience in automotive engineering. It was considered to have demonstrated sufficiently that the effect of road grade on driveshaft torque could be calculated for typical operating conditions. Thus, it would not be necessary to measure this characteristic of additional vehicles for which performance data bases were to be prepared.

Attempts to identify an effect of horizontal curvature on torque in this set of test data were unsuccessful. It was concluded that a more sensitive test of this phenomenon, in which the desired effect was isolated as much as possible from perturbing influences (such as grade), was required.

For additional information, see Appendix D, Section D.2.

3.3 EFFECT OF HORIZONTAL CURVATURE

At the outset of this project, the FHWA's early requirements were to demonstrate (a) whether the effects of highway geometrical features on vehicle propulsive demand could be measured, (b) whether such effects could be calculated reliably for vehicles without having to test each vehicle on the road, (c) whether vehicle operation on various highway geometries could be simulated adequately on a large-roll research-grade chassis dynamometer, and (d) whether such simulation was necessary. While these questions were answered quickly for the effects of road grades, the influence of horizontal curvature was more difficult to quantify. It was necessary to develop special test methods and equipment with greater sensitivity and to eliminate sudden transients in road geometrical configuration before adequately quantitative data could be obtained.

A suitable method of quantifying the effect of horizontal road curvature (without superelevation) on driveshaft torque was developed. Three concentric circles with radii of 100, 200, and 300 ft (30.5, 61, and 91.5 m) were marked out with road-marking tape on a portland cement concrete-paved plane area that was inclined at about 0.8 percent (for drainage). A common starting point for all circles, at the high point of the surface, was marked with a radial strip of tape. The test vehicle was driven around each circle in continuous laps at nominally constant speed while data were recorded for six laps. This procedure was repeated for a total of five speeds on each of the three circles, for each of three cars. The five speeds on each circle were chosen to produce lateral accelerations of approximately 0.1, 0.2, 0.3, 0.4, and 0.5 G. (Lateral acceleration = speed squared/radius of curvature.)

Analysis of the data showed that the dependent variable was the increment of driveshaft torque, i.e., the difference between torque on a curve at a given speed and torque on a straight, level road at the same constant speed; this parameter was termed the "curve-torque increment." Similarly, the independent variable was found to be lateral acceleration, which conveniently incorporated both the curve radius and speed on the curve into one parameter. The curve-torque increment showed extremely good correlation with a quadratic function of lateral acceleration, was independent of speed per se, and—for radial-ply tires—appeared to be independent of curve radius. Bias-ply tires on one car also showed clear dependence on curve radius. The self-consistency of each set of data for a given car and tire set was much better than normally is encountered in measurements on operating vehicles.

The curve-torque increment was divided by the straight-road torque at the same speed to "normalize" the quantity i.e., to reduce or eliminate the explicit restriction of the relationship to an individual vehicle or type of vehicle. This normalization introduced a modest dependence on curve radius and increased the dispersion of the data points. However, when all normalized data for each car individually were used to calculate a single regression curve for that car, and the three regression curves were compared, a reasonable degree of agreement between the curves was apparent. Another regression of data points calculated from each of the three curves produced a single composite regression curve that represented an approximate description of any of the three cars.

This composite regression curve of normalized curve-torque increment vs. lateral acceleration is proposed for general use in estimating the effect of horizontal curvature on any vehicle that has not been curve-tested but for which the straight-road, tractive force, or driveshaft torque relationship to constant speed is known.

Subsequent to much of the curve testing and analysis, and independent of that work, a theoretical analysis of "cornering drag" vs. curvature and speed was published (3). This study, developed admittedly with almost no experimental data to support it, found precisely the same parameters to be active as are described above. The relationships also were in agreement. The theory explained the divergence of normalized curve-torque increment data, i.e., it showed there is a dependence on curve radius; further, the theory demonstrates there also is a dependence on the cornering compliance of the tires. Therefore, there is theoretical opposition to the validity of the single "universal" normalized curve-torque increment relation postulated above to be applied to all passenger cars if no other data exist. However, for the purposes of this project, the availability of at least one empirically based approximate description of the effect of horizontal curvature is preferable to having no relationship simply because the cornering compliance of unknown makes of tires is unavailable.

Thus, this project validated experimentally a simple analytic expression relating added propulsive requirement on curves to a function of both speed and curve radius, viz., to lateral acceleration. The relationship was shown to be smooth, stable, and repeatable.

The software needed for the input of modules that defines horizontal curvature vs. distance along the road is a direct parallel of that for grade vs. distance. Hence, the computation routine for calculations of curve-torque increment is known. The software for grades was copied and modified slightly for horizontal curvature.

For additional information, see Appendix A, Section A.3.2; Appendix C-Sections C.1.2, C.1.3, and C.2.2; and Appendix D, Section D.2.3.

3.4 DETERMINATION OF ROAD LOAD FROM CONSTANT-SPEED TORQUES

The term "road load" as used in this report and project has a specific meaning; it refers to the total propulsive requirement, in terms of either driveshaft torque or tractive force of the drive tires, as a function of vehicle speed on a level, straight road at any constant speed and in the absence of any ambient wind (i.e., wind speed relative to the ground). Road load, as used here, excludes any power demands on the engine by engine auxiliaries, air conditioning, etc. Road load, an important characteristic or property of any vehicle, typically is described by a mathematical expression always containing a term independent of speed and another term that is dependent on the second power of speed; the expression may or may not contain another term dependent on the first power of speed, based on the mathematical technique by which the regression is performed.

Determination of road load was essential in this project. First, the road load relationship was an important component of the vehicle data bases that would be constructed to support the analytical algorithm for estimating vehicle fuel consumption on highways. Second, road load was required in analysis of the curve-torque phenomenon to provide the base quantity to subtract from total on-curve torque.

For uninstrumented vehicles, road load most often is determined by the coastdown technique (4, 5). However, roads and airport runways in New England lack the requisite straight length (typically, 2 mi or 3 km) with grade that is essentially constant (+0.1 percent) and of small magnitude (+1 percent or less). Coastdown tests under this project did not produce satisfactory data.

Since the test vehicles already were equipped to measure and record driveshaft torque, an alternative technique was to run the vehicles in both directions at several different, nearly constant speeds over a modest length of straight road that was adequately constant in grade. The central 6000-ft (1.8-km) portion of the Bangor runway, with a slope of 0.6 percent, proved quite satisfactory. Test speeds usually were approximately 25, 40, 55 and 70 mi/h (40, 64, 88, and 113 km/h). Regressions of mean torque against a quadratic function of mean speed were performed simply with a programmable calculator; the correlations were consistently excellent. The resultant equation provides data on both the rolling friction (speed-independent and speed-linear terms) and aerodynamic drag (speed squared term) characteristics of the vehicle.

During these tests, a limited number of tests was conducted at speeds of less than 25 mi/h (40 km/h) on the circle-test pad over distances of less than 1000 ft (300 m). The results of these shorter-length tests indicated that torque-instrumented vehicles such as these also could be used for another type of measurement: comparison of the rolling friction characteristic of different pavements. This could be done as described in Section 3.5 below.

3.5 PAVEMENT FRICTION COMPARISON WITH TORQUE-INSTRUMENTED VEHICLE

The sections of pavement on which friction is to be measured should have as little slope as feasible and should have contiguous approach segments that permit approach from both directions at the speeds at which measurements are to be made. The measurement sections preferably should be straight, or at least have curvatures low enough to generate lateral accelerations of less than about 0.05 G at the highest intended test speed. Test length need not be more than few hundred feet, but more length should promote better data. The approaches need not be as straight as the test section, but should not have substantial transients that would hinder stable operation on the test section. Replicate series of torque measurements, taking data points every few feet of travel, should be made on the test section in pairs at each of several speeds spanning as wide a range as possible, with consecutive runs made in opposite directions at each speed (see Appendix B, Section B.4.2). Similar measurements should be made on each type of pavement surface to be evaluated.

Regressions of the data will produce coefficients of both rolling friction and aerodynamic drag. Since the tests all were performed with the same vehicle, the aerodynamic coefficient should be essentially the same in all tests (except for normal experimental variation). The only reason for variation in the rolling friction coefficients should be the differences in the pavements driven on.

It must be noted that the absolute values of rolling resistance on each kind of pavement will be specific to the particular vehicle, tires used including tire pressure, moisture condition on road, and length of time the vehicle has been in operation. (Until engine and tires are warmed up, rolling resistance is higher. Water, snow, or ice on pavement will keep tires much cooler) .

A variety of vehicles should produce somewhat diverse rolling resistance values on a given pavement. However, pending evaluation of this application of the technique, it is reasonable to expect that the relative effects of several types of pavement on each of a variety of vehicles should produce approximately comparable relative changes in rolling resistance for each of the vehicles. Therefore, friction measurements with only a single test vehicle should give a reasonably valid assessment of the relative behavior of the different pavements for traffic in general.

3.6 INFLUENCE OF DRIVER VARIANCE ON HIGHWAY FUEL ECONOMY

An experiment was conducted with 10 drivers selected at random from the VNTSC employee staff and consisted of 5 women and 5 men of various ages, plus a "control" driver who was the experiment conductor. The 10 drivers were not told that their driving habits were the subject of the experiment; they were given to understand they were evaluating some prototype road-test equipment. The test road was a rural section of a limited-access, divided highway used for this test only during low-traffic times; it had a uniform posted speed limit of 55 mi/h (88 km/h). The test vehicle was one of the instrumented cars with automatic transmission; the data logger recorded speed, torque, and fuel flow continuously throughout the test drives at 1 sample per second. Each driver made six runs, three as he or she "would normally drive" and three while keeping speed as constant as possible at 55 mi/h (88 km/h); two runs were made on each of three different days.

The experiment was conducted at highway speeds because a literature search showed no previous work had been performed at this level, while a considerable amount of testing had been done at urban conditions. The literature indicated a general consensus that, at highway speeds, fuel economy drops as average speed increases, is dependent mainly on the speed driven, and is largely independent of the driver (6).

The results of this experiment refuted this usual assumption. The fuel economy observed was found to be driver-dependent beyond the effect attributable to speed chosen by that driver. The maximum range of fuel economy values was about 4.5 times that which should have resulted from the spread of speeds involved. For all drivers at all speeds from 52 to 57 mi/h (84 to 92 km/h), there was no statistical correlation between fuel economy and speed; however, there was a very strong statistical indication that fuel economy was associated with the driver.

3.7 EFFECT OF TRACTIVE FORCE LEVEL ON ENERGY DISSIPATION IN TIRES AND AXLE

Whenever chassis dynamometer testing is performed on a vehicle equipped to measure driveshaft torque, the torque or tractive force (positive or negative) observed at the dynamometer roll surface always is smaller in magnitude than the value corresponding to torque measured in the driveshaft. This "torque loss" results from friction in the bearings of the drive axle and from energy dissipation both within the drive tires and between the tires and the dynamometer rolls. The sum of the last two quantities usually is termed the "rolling friction" of the tires, and generally is assumed to comprise nearly the total amount of torque loss. Rolling friction is measured routinely by tire manufacturers and researchers at zero transmitted torque; their test equipment generally is not equipped to apply or measure a significant amount of drive or braking torque. Consequently, very few data have been published on the effect of drive torque level on energy dissipation, i.e., on rolling friction under significant drive torque output.

During performance mapping of one of the test vehicles on the large-roll chassis dynamometer, the torque loss was analyzed for nearly every test point. When the torque loss was plotted against total drive torque as the independent variable (or, alternatively, loss in tractive force against total tractive force), the relationship could be described quite well by a quadratic function of total torque (or force). The relationship appeared to be essentially independent of speed over the range from 15 to 117 km/h (9 to 73 mi/h). Thus, this was the simplest manner in which to express the phenomenon; had it been expressed in terms of power, a speed-dependency would have been introduced that would have complicated the evaluation. The effect of speed was, in fact, introduced implicitly by the higher drive torque or tractive force required to maintain higher speeds.

The magnitude of the torque loss, in relation to total torque, tended to be about 6 to 7 percent of total torque over most of the normal operating range of the vehicle. At very high torques corresponding to low gear and heavy throttle, the loss increased to more than 10 percent. As drive torque approached zero, the torque loss reached a minimum but still significant value—equivalent to the normal "rolling friction" measurement; the ratio of loss to total torque, of course, became meaningless as the denominator approached zero. At negative drive torques, the loss began to increase again.

For additional information, see Appendix C, Section C.1.5.

3.8 EXPERIMENTAL VEHICLE DATA BASES

Data bases of experimental fuel economy (measured on a chassis dynamometer) were produced for three cars; one of the three included associated data on exhaust emissions. Test conditions spanned the normal operational ranges of speed and torque. Fuel economy data for the first vehicle (plus an earlier one used in the pilot study, for which the test conditions did not include large negative torques) could be represented quite well, over essentially the entire highway range of speed and torque, by a simple, second-order equation in torque alone. This was true, also, for the last two vehicles tested, at medium to high torque levels in all gears; but, from low positive to negative torques, a separate equation of the same form for each individual transmission gear furnished better descriptions of the data. It may be significant that the earlier two cars both had automatic transmissions with torque converters, while the latter two had manual transmissions.

Some analysis of the above data was performed using multiple independent variables (such as speed, torque, and transmission gear, sometimes plus products of these quantities). However, provision of sufficient data to support adequately such detailed analyses would have required testing at two to three times as many torque and speed levels. The objectives of this project could be met satisfactorily with the smaller number of test points used, and with the simple equations in torque only, sometimes using a separate function for each gear. This smaller amount of data reduced the costs of vehicle testing and of data analysis and utilization.

The exhaust emissions data for the one car also were represented by simple analytic functions. While the scatter in these data was much greater than was observed for fuel economy results, this is consistent with auto industry experience.

3.9 SYNTHESIS OF FUEL ECONOMY (AND AIR CONTAMINANT EMISSION) DATA FOR VEHICLES NOT TESTED

The experimental data base described in Section 3.8 contained information on only three cars; this did not provide much variety for use with the HPP. Only the data base for one vehicle tested was used for a Vehicle Data Base Module (VDBM)(Pontiac Le Mans). VDBM modules could have been made for other vehicles tested. This was not done. Methods have been devised by which fuel economy and air contaminant emission data might be synthesized for other vehicles that had not been actually tested. These procedures include VEHSIM described in Section 2.0. Other vehicle data can be used if the information required is placed into a VDBM as prescribed in the instructions for the HPP that are provided. If any data required for a VDBM, namely fuel consumption rate or carbon monoxide, hydrocarbon, or nitrogen oxide air contaminant emission rates are not included, the missing data is identified as a zero and no answers will be available for that vehicle for that factor. Precautions should be exercised if a vehicle with no input for part of a VDBM is used as part of a "fleet" in the HPP so the proper interpretation of results will be made.

3.10 FRICTION MEASUREMENTS BEFORE AND AFTER RESURFACING A HIGHWAY

A minimal effort was made under this project to evaluate the use of a torque-instrumented vehicle for road surface-friction measurements. At the time when one test vehicle was undergoing field tests on Bangor airport, a section of Interstate I-95 running through Bangor was being prepared for resurfacing; the condition of the surface had become seriously degraded. The Maine State Department of Transportation (MeDOT) suggested that the instrumented car be used to measure running friction on that portion of road before and after resurfacing, to see how much difference would be observed.

A single record run was made on a 5-mi (8-km) length of the high-speed (left-hand) lane of the southbound portion of I-95; data were recorded 10 times per second. Approximately 1 year later, the same test vehicle was returned to Bangor and three replicate runs were made over the same portion of road. MeDOT provided highway construction drawings showing the elevations along the test route. Limited observations and analyses of results showed that observations of total load could indicate variations of road roughness after accounting for inertia, gravity, curvature, and aerodynamic loads. Others have seen this implicitly by observing fuel usage as the road surface varied or measurement of loads as a function of tire-road interface conditions. Despite the potential for using dedicated vehicles measuring driveshaft torque to evaluate road roughness, the procedure was not adopted because direct means of simultaneously measuring linear acceleration and therefore gravity loads (accurate measurements of grade or slope) were not successful. Since this work was completed, Sierra Research for California Air Resources Board (CARB) and University of California at Riverside for National Cooperative Highway Research Program (NCHRP) project 25-11 were able to make continuous measurements of grade simultaneously with driveshaft torque measurements. (Linear accelerometer and use of Global Position Monitoring (GPS), respectively.)

3.11 FUEL CONSUMPTION DURING ACCELS AND DECELS ON CHASSIS DYNAMOMETER

Fuel consumption of one test vehicle was measured during transient accelerations (accels) and decelerations (decels) on a large-roll chassis dynamometer. The intent was to obtain empirical data on such consumption against which to test estimated values for the same maneuvers generated by the vehicle/highway performance prediction computer program. Operating problems with the dynamometer precluded further tests of this type, and resource limitations prevented running of computer simulations to compare against the empirical data.

The experiment was conducted with the 1980 Chevette. To enhance repeatability of the test, the cycle was driven completely in top (fourth) gear of the manual transmission, and acceleration rates consequently were limited to modest values. During acceleration, speed was increased from 30 to 50 mi/h at 1.0 mi/h/s, and from 50 to 60 mi/h at 0.67 mi/h/s; in the deceleration, speed was reduced from 60 to 30 mi/h at 1.33 mi/h/s. (In SI units, these were: accel from 48 to 80 km/h at 1.6 (km/h)/s, and from 80 to 97 km/h at 1 (km/h)/s; decel from 97 to 48 km/h at 2.1 (km/h)/s). Exhaust gases were collected between 35 and 55 mi/h (56 and 88 km/h) during accels and decels and accumulated in two separate sample bags. Throttle position had to be changed continuously during both halves of the cycle to maintain the accel/decel rates. Three series of runs were made, and consisted of 20, 22, and 24 cycles in order to accumulate enough exhaust gas for analysis. Note that fuel consumption was not measured by a fuel flowmeter because these devices tend to lag behind fluctuations in power and produce erroneous results for sharp, short-duration transients.

The composite fuel economy values observed for the consecutive series were: accelerations, 18.8, 19.7, and 20.9 mi/gal (8.0, 8.4, and 8.9 km/L); decelerations, 79.8, 91.2, and 87.2 mi/gal (33.9, 38.9, and 37.1 km/L). The acceleration results suggest that the driver's technique was continuing to improve with increasing experience in driving this new test cycle.

4.0 CONCLUSIONS

4.1 EFFECTS OF HIGHWAY GEOMETRIC FEATURES ON VEHICLE PERFORMANCE

This project has documented by empirical data that the effects of major geometrical features of highways on propulsive demand on vehicles are significant, can be measured, and can be predicted, i.e., calculated without measurement. The effect of grades is linearly related to the slope of the road. The increase in demand on curves was found to be a smooth and repeatable function of lateral acceleration; and lateral acceleration provides a convenient single parameter that expresses the interrelated influences of radius of curvature and speed on the curve. These empirical effects of horizontal curvature were consistent with a theoretical analysis of the phenomena that were developed independently of this project.

Experimental data have been used to construct data bases of operational parameters of modern (1975-1981) automobiles; these data bases relate such parameters to driveshaft torque (and, in some cases, to transmission gear) by simple analytic functions of no higher than second order. Thus, it has been shown that the fuel consumption of an automobile can be calculated with adequate accuracy for such real-road maneuvers as accelerations, decelerations, and travel on positive and negative grades and on horizontal curves. Further, if test data on exhaust emissions are available, the effects of such maneuvers on these parameters can be calculated in a similar manner. While emissions computations provide a lesser degree of accuracy than for fuel consumption, they still afford an objective comparison of the effects of different geometries that is based on sound automotive engineering principles. Such computations ought to be better approximations of real conditions than estimates based on long runs at various constant average speeds over totally flat straight terrain.

The combinations of maneuvers experienced by vehicles operating on highways of arbitrary configurations can be simulated adequately by decomposing combined maneuvers into a series of quasi-steady-state operations on short lengths (usually only a few meters in length) of road. Fuel consumption (and emissions) can be calculated for each increment of road and summed over the entire length of highway. Alternative geometrical configurations can be evaluated quickly and objectively to compare their relative effects on fuel consumption. Road segments where the intended combination of speed and grade cannot be met by one or more types of vehicles in the design fleet are quickly identified for appropriate design action.

4.2 VEHICLE/HIGHWAY PERFORMANCE PREDICTOR (HPP) SIMULATION

A computer program has been developed that can calculate fuel consumption and exhaust emissions of automobiles when operated with realistic accelerations and decelerations on highways that have grades and curves. This software incorporates the essentials of automotive engineering principles that address vehicle response to combinations of applied loads; the intent is to provide a valid approximation of actual fuel consumption on the specific highway. The quality of the approximation is further improved when the algorithm is used to compare fuel consumption on two or more alternative configurations of the same road. This is the intended principal application of this method. The improvement in approximation is obtained because any errors in absolute magnitude of fuel consumption on one highway layout likely will be largely canceled out by similar errors on the other designs; thus, the relative magnitudes of the calculated fuel consumption should tend to reflect quite well the relative effects of the different configurations.

The computation method is convenient to use, runs very rapidly (typically, it analyzes 2 to 3 km road/s of computation time), and has been developed for use on personal computers. The computer program described above, the HPP, was first made operational on the VNTSC mainframe computer with three slightly different versions; later, an interactive version for personal computers was developed that allows additions of other vehicle data bases. The computational procedure, the vehicle highway performance predictor algorithm (HPP), could contribute significantly to analyses of the merits of proposed or planned upgrading of existing highways, to design of new or relocated highways where fuel consumption rates and exhaust emissions are important, and to develop and evaluate traffic management.

APPENDIX A
THE VEHICLE/HIGHWAY PERFORMANCE PREDICTOR (HPP) ALGORITHM

A.1 DESCRIPTION OF OPERATION, PROTOTYPE VERSION

This appendix describes the operations of the prototype Vehicle/Highway Performance Predictor (HPP) algorithm. First, the unit operations performed on each increment of road length are outlined. The results from each increment are summed over the entire length of highway. The procedure for tracking position on the highway is presented. Reasons for computerization of the HPP are given. Finally, a brief overview of the entire algorithm is sketched, and a logic diagram demonstrates how all the component operations relate to each other.

A.1.1. UNIT OPERATIONS ON ROAD ELEMENT FOR ONE VEHICLE

A.1.1.1 Road Elements

The HPP subdivides the total road length into elements of distance, each of which must have a constant grade (positive, negative, or zero), and a limited change in speed over its full length. Elements generally average about 50 m (0.050 km) in length; however, an element may be of arbitrary, much greater length if speed, curvature, and grade are constant. If curvature, speed, or grade change within the nominal 50-m length, the element is subdivided further into two or more smaller elements that meet the constancy constraints.

To illustrate this process briefly, consider the example shown in figure 1. The input speed profile calls for a constant acceleration of 2.083 (km/h)/s from a starting speed of 20 km/h to a final speed of 40 km/h. For this example, it is assumed that the maximum speed change per road element is 5 km/h. Consequently, this acceleration will be distributed over four road elements.

The length of each element is calculated from the classical constant-acceleration relationship:

$$V_2^2 = V_1^2 + 2\ ax$$

This equation is transposed to solve for distance:

$$x = (V_2^2 - V_1^2)/(2a)$$

The elapsed time per element is calculated by another constant acceleration equation:

$$V_2 = V_1 + at$$

which is transposed to solve for time:

$$t = (V_2 - V_1)/a$$

For the example cited, the results are listed in table1 and plotted in figure 1. It is clear that element length varies, that speed is non-linear vs. distance, and that the linear-average speed for each of these short elements is very nearly equal to the actual instantaneous speed at mid-element. (The significance of this last point will be apparent in the Road Load section below.)

Table 1. Example of speed and elapsed time vs. distance at constant acceleration over four road elements.

Start = 20 km/h

End = 40 km/h

Acceleration = 2.083 (km/h)/s= 1.295 mi/h/s = 7500 (km/h)/h

$x=(V_2^2 - V_1^2)/(2a)$ $\quad$ $t = (V_2 - V_1)/a$

Element No.	Distance, m		Speed, km/h		Elapsed Time, sec		Element Linear-Average Speed, km/h
	Element	Sum at End	Start	End	Element	Sum at End	
1	15.0	15.0	20	25	2.4	2.4	22.5
2	18.33	33.33	25	30	2.4	4.8	27.5
3	21.67	55.00	30	35	2.4	7.2	32.5
4	25.0	80.00	35	40	2.4	9.6	37.5

For each element of road, the following operations are performed if applicable; any process not pertinent to a given element is bypassed.

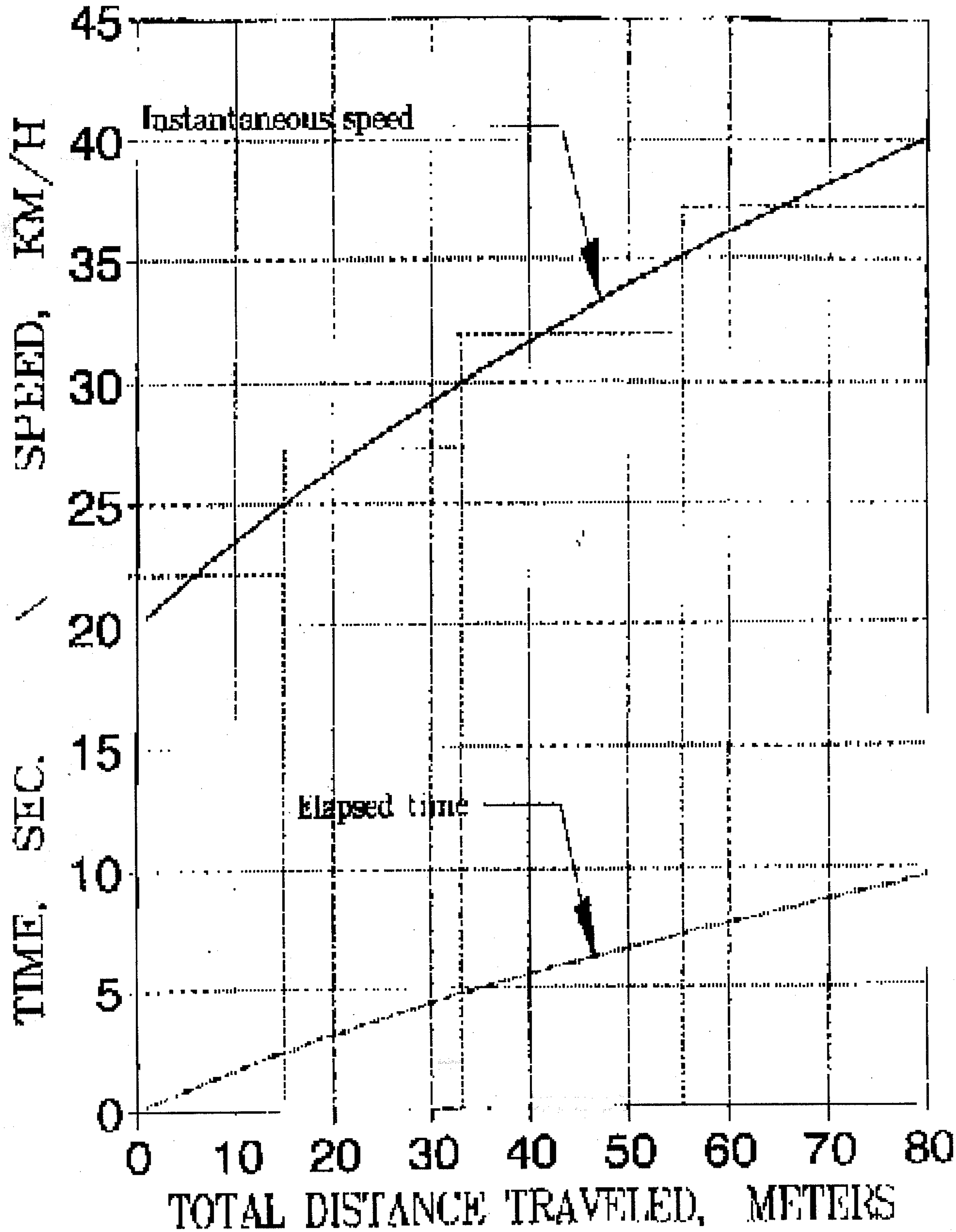


Figure 1. Example of speed and elapsed time vs. distance.

A.1.1.2 Speed

Vehicle speed at the start of each element is equated to the speed leaving the preceding element (for the first element of the road, the user is required to enter the starting speed). Speed leaving the element is determined from the speed module specifications (see Section A.2.2 below) and the limitation in speed change over one element. Average speed on the element is the linear average of entering and leaving speeds; this average speed is used only to calculate the basic road-load tractive force for the element (see following section), and is not used to compute cumulative distance or time.

A.1.1.3 Road Load

As described earlier, the term "road load" has been restricted throughout this project to refer exclusively to total propulsive demand (tractive force at the wheels, or driveshaft torque) at a given constant speed on a level, straight road in the absence of any wind (air motion relative to the road surface). It defines the basic propulsive requirement of the vehicle before any perturbing influences are considered. It does not consider the peripheral power demands imposed on the engine by vehicle accessories (except that the effect of peripheral loads active when the vehicle was being performance-mapped are included implicitly in the fuel economy data).

For a given vehicle, the vehicle data base contains an equation that defines the road load tractive force as a function only of vehicle speed. On any element of road, the change in speed is sufficiently small that speed can be considered to be constant (at element average speed) for computation of road load.

Thus, road load is calculated by entering element average speed into the road load equation. This yields the first component of total tractive force for the particular road element.

A.1.1.4 Acceleration

The acceleration on a given element is derived from the input speed vs. distance specification. The HPP obtains from the pertinent VDBM (described later in Section A.2.4) the vehicle effective inertia weight; this includes an increment of inertia weight corresponding to the rotational inertia of the tires and wheels.

The force (positive or negative) required to produce the specified acceleration then is calculated. This yields the acceleration component of total tractive force for the element.

A.1.1.5 Grade

For the HPP, highway grade vs. distance is entered in terms of grade in percent (m of rise per 100 m horizontal distance) and distance in km. The tractive force increment is calculated from the vehicle weight (w) (obtained from the vehicle data module) and the grade:

$$F_G = W\,(\text{Grade, \%})/100$$

To be mathematically rigorous, the relationship would use the sine of the road elevation angle. However, for grades up to 4 percent (greater than normally encountered on most modern highways), the error from using grade is less than 0.08 percent of the correct value of this force component. And, for grades up to 14 percent, the error still is less than 1 percent. Consequently, this convenience imposes no significant error.

A.1.1.6 Curvature

For the HPP, the highway radius of curvature in meters is entered. Based on vehicle speed, the lateral acceleration can then be determined. The load due to curvature is then found to be proportional to the road load for a straight road determined for given vehicle (s) and speeds according to the equation shown in figure 10; namely, Δ T (curve) = T(straight) 1.349 $A_{Lat} + (3.37\ A_{Lat})^2$. The force or load due to curvature can be derived based on discussion in Sections A.1.1.7, Driveshaft Torque and C .2.1, Derived Tractive Force for Three Vehicles.

A.1.1.7 Driveshaft Torque

In accordance with original guidelines for this project, vehicle operating parameters such as fuel economy and exhaust emissions were related to driveshaft torque as the independent parameter. Correlations with this single independent variable for the first two vehicles tested were surprisingly good and afforded great simplicity and computational economy. Subsequently, road speed was added as a second variable in recognition of the fact that these power-dependent parameters should, in general, be dependent on more than just torque. The degree of correlation improved somewhat, for some vehicles, with this addition. Both of these independent variables also were the actual parameters that were measured in test operations.

The four tractive force components (road load, acceleration, grade, and curves) are summed to obtain the net tractive force on the road segment. The HPP then obtains from the vehicle data module the effective rolling radius of the drive wheels and the final drive ratio of the drive axle, if applicable. (In the case of an FWD vehicle, driveshaft torque usually is accessible only in the half-shafts between differential and wheels, and hence already is outboard of the differential. For these cases, the drive ratio is set equal to 1.0.)

The driveshaft torque corresponding to the net tractive force is calculated from the equation:

$$T_s = (\Delta F_i)*R_{wheel}/(\text{Drive ratio})$$

This calculation neglects two sources of energy dissipation: friction in the final drive gears (for RWD vehicles) and loss(es) in the drive axle(s) that are functions of the total tractive force transferred by the drive tires. The final drive efficiency generally is assumed to range around 97 percent at moderate to high torques; at low torques, efficiency falls off sharply, but the absolute magnitude of power loss at low torques tends to be small.

The second loss, the force-transfer loss in drive tires and axle, is a parameter about which very little data have been published. When rolling friction of tires is measured, either by manufacturers or by others, the standard practice is to run the tire on an unpowered axle in free rotation. Under this project, the force-transfer loss was measured for only one pair of tires on one vehicle (see Section C.4). It would be risky to generalize the observed relationship to other vehicles without further investigation. Therefore, this correction to driveshaft torque is neglected for most of the vehicles in the present data base.

However, the 1980 Pontiac was the vehicle for which this parameter was measured. Here, the driveshaft torque calculated above is used in a quadratic equation to compute a correction increment; this increment is added to the above shaft torque to obtain the final net shaft torque.

A.1.1.8 Fuel Consumption

For many highway-related calculations, fuel consumption is a more convenient parameter to use than the more familiar fuel economy. However, when experimental data are being regressed to establish a mathematical relationship, fuel economy proves to be considerably more tractable. Accordingly, an expression for fuel economy is developed and then used as a reciprocal:

Fuel consumption = 1/(fuel economy)

To determine fuel consumption for the road element being analyzed, the HPP delivers the net driveshaft torque and speed to the VDBM and requests the corresponding fuel consumption rate and exhaust emissions rates. The VDBM is configured to perform the calculations internally, in the manner of a subroutine, and to return the requested outputs. Thus, if a given parameter for separate vehicles can be best represented by equations of different form (e.g., a power equation for one and a polynomial for another), these differences in data format are transparent to the HPP. Further, if the data module for a given vehicle has no information on exhaust emissions, the module simply returns zero for these parameters.

The HPP receives the requested fuel economy (and emissions rates, if available) in terms of units/km of distance traveled. Each of these rates is multiplied by the length of the road element in question to produce an increment of fuel used (or emissions generated).

A.1.1.9 Performance Limits

Although performance parameters are included in the VDBM, the current version of HPP does not include checking to determine if performance limits have been exceeded.

The above description completes the unit operations on any given road element. It is evident that no great mathematical complexity is involved. Also, an intimate knowledge of automotive engineering relationships by the highway designer is unnecessary; as much of that technical specialty as is required has been built into the procedures of the HPP and especially into the VDBM. The net output of the procedure, for a single element of road, is either one or four numbers: always, liters of fuel used; and, if emissions data are available, grams of hydrocarbons (H.C.), grams of carbon monoxide (CO), and grams of oxides of nitrogen (NOx) emitted.

A.1.2 SUMMATIONS OVER ENTIRE ROAD LENGTH

The unit operations described above in Section A.1.1 determine a realistic estimate of the amount of fuel consumed (and emissions produced, if data are available) on a piece of road only a few meters in length, for a single iteration. The process can be repeated as many times as necessary to "travel" the entire highway. The increments of fuel and emissions are summed in individual accumulators to compile a running total of each parameter along the highway.

A.1.3 HIGHWAY GEOMETRY PROCESSOR (HGP)

The iterative process described above, when performed on a highway length measured in kilometers (or miles), requires a "navigator" to keep track of position along the highway, of the starting and ending speeds for each element, length and road grade for the element, or duration of stationary idle. The HGP allows the speed vs.

distance specification to be changed independently of the grade vs. distance specification. When such a substitution is made, the lengths of many road elements will have to be modified, and their positions along the highway will shift.

All of these operations are quite simple, but the amount of "bookkeeping" very quickly becomes oppressive and subject to human error.

A.1.4 COMPUTERIZATION OF ALGORITHM

A.1.4.1 Rationale

As noted above, the procedure for calculating the effects of highway geometry and vehicle operation on vehicle fuel economy and emissions is quite simple. However, it requires a very large number of iterations and much recordkeeping. When either the speed or grade profile is changed, the entire process must be repeated. When different vehicles are operated over the same speed and grade profiles, all calculations must be repeated again.

The resultant information is far more specific to a particular highway than other calculation procedures we are aware of. One important advantage is that the effects of road grades and curves are considered; many earlier programs operate only on a flat world. Also, the influence of speed transients is included; many other programs operate with piecewise-discontinuous, constant average speeds but do not consider the substantial effects involved in achieving the different speeds. At best with the earlier models, the designer would be required to calculate average numbers of speed changes and stop/starts per mile, lockup tables of correction factors, and hand-calculate a number of corrections.

Information on relative fuel consumption of alternative highway geometries can be valuable to a highway designer, but only if achieving it does not encroach seriously on highway design time. The designer's primary responsibility still is the design and cost estimation of highway construction.

The entire computation procedure is ideally suited for operation on a digital computer. Once the highway geometries and speed profiles are entered into data files, essentially no interaction by the user is required. The vehicle data would be stored in data modules. New vehicle data modules could be added at any future time without requiring any changes in computation software.

Consequently, it was decided to develop a prototype computer program to evaluate the procedure and to identify any problem areas that might not have been foreseen.

A.1.4.2 Implementation

The HPP program was first written in FORTRAN IV language for a minicomputer. Subsequently the HPP was transferred to a mainframe computer. Then, the HPP program was rewritten in Visual Basic language and adapted to personal computers. The software and instructions for installation of the Highway Performance Software have been placed on a CD ROM.

A.1.5 OVERVIEW OF COMPLETE HPP PROGRAM

This section presents the Vehicle Highway Performance Predictor (HPP) logic diagram and discusses the major components thereof and their interrelationships.

Figure 2 is a diagram of the logical organization of the HPP. The diagram and the discussion below show the functionally correct relationships of the major components. Note, however, that the diagram does not necessarily represent the organization of actual computation procedures. In some cases, the programmer chose to subdivide the tasks in a different way for operational convenience and economy.

The core of the HPP operations is the Computation Executive, labeled in figure 2 as V/HPP. This Executive is in charge of the sequential activation of the several peripheral modules on a given elemental length of highway, of the iteration of this sequence many times over the entire length of a highway, and of output of data as specified. Each major function is identified on one facet of the border of the Executive and is connected with the associated peripheral component. The general sequence of operations is illustrated by numbers located near each function.

(1) Vehicle speed, speed change, and curvature are derived from the two principal user input modules. The specifications of grade, speed, and curvature vs. distance are combined and then subdivided into individual road elements in the HGP (described below).

Information flow between input modules and Executive basically is one-way, into the Executive on demand.

(2) The HGP is the "navigator" for the HPP. It keeps track of where on the highway the analysis presently is located. For each road element, the HGP tells the Executive the beginning and ending speeds on the element (or, alternatively, that the vehicle is stationary at idle and for how long), the grade of the road element, the horizontal curvature, and the length of the element. Information flow between HGP and Executive (V/HPP) is bidirectional.

The VDBM is consulted by the Executive at various times as specific pieces of information are required. The first such contact is made to determine road load tractive force (as stipulated earlier in this report, "road load" is restricted in this report to mean load at constant speed on a level, straight road).

(3) Road load force is calculated as a function only of vehicle constant speed. For a given road element, the Executive calculates the linear average of starting and ending speeds to determine the average speed on the element. The Executive sends this average speed to the VDBM and requests the corresponding road load. The road load force is calculated within the VDBM and is returned to the Executive. In this way, the specific form of the road load equation is transparent to the Executive; different forms of the equation can be used by the VDBM for various vehicles without affecting the software of the Executive.

(4) The Vehicle Dynamics section of the HPP is addressed next by the Executive. The values of acceleration and road load force in effect on the element are sent to the Dynamics unit. The Executive also must obtain from the VDBM the effective inertia weight of the vehicle(s), and send this to the Dynamics unit. The Dynamics section calculates the tractive force required to produce the specified acceleration.

The Executive then sends the current road grade to the Dynamics unit, which calculates the grade and curvature (lateral acceleration) force components. The Executive obtains from the VDBM the drive wheel radius and the drive axle differential ratio and sends these data to Dynamics. All tractive force components are summed algebraically, and the driveshaft torque is computed.

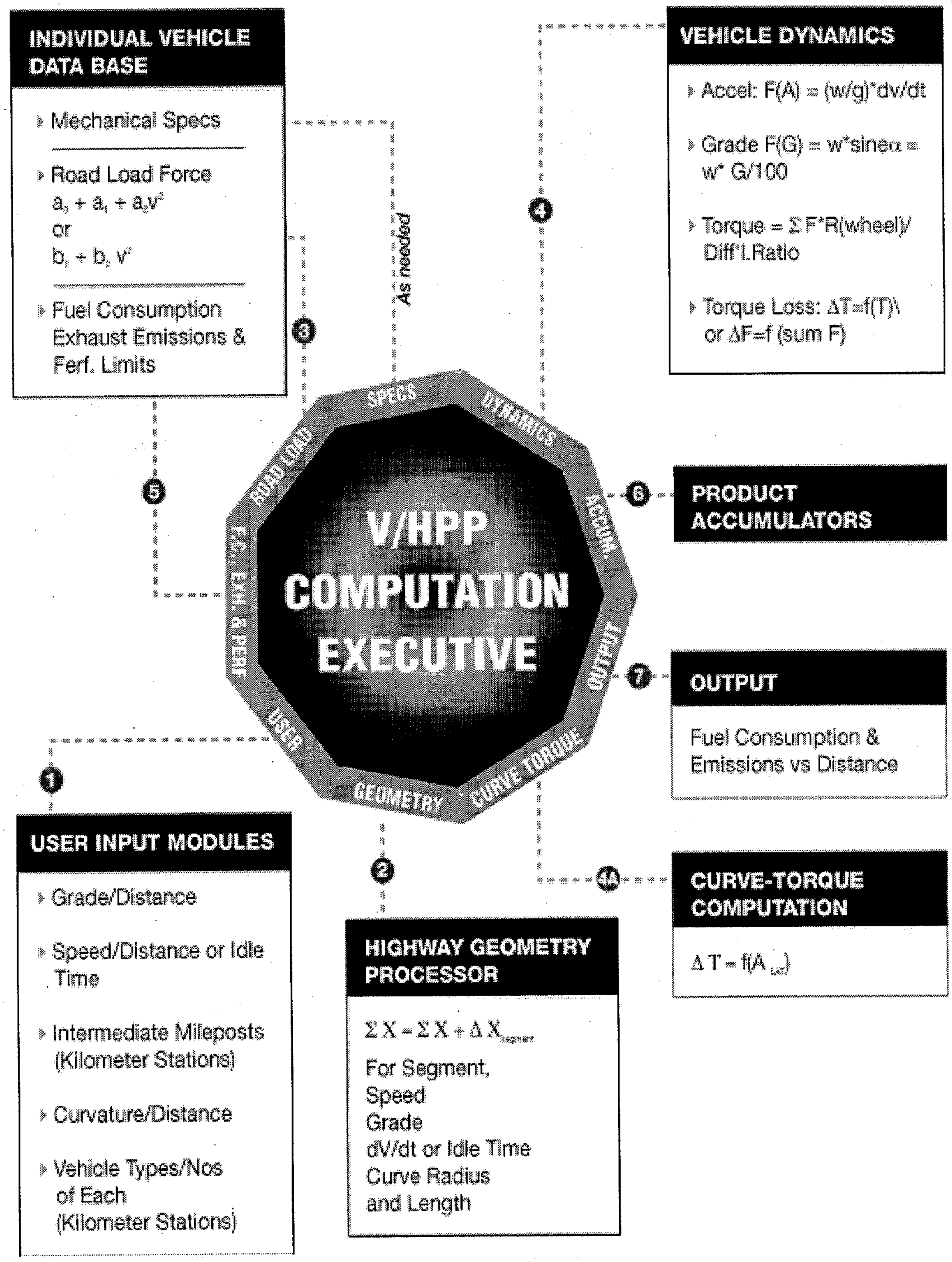


Figure 2. Vehicle/highway performance predictor software–logic diagram.

If the VDBM contains an expression for tractive force-transfer loss, this is used by the Dynamics section to calculate and add to shaft torque an increment for this effect.

The Dynamics unit finally returns to the Executive the final net driveshaft torque for the road element.

(5) The Executive sends this net torque to the VDBM and requests the rates of fuel consumption and exhaust emissions. Ancillary information such as vehicle speed also is transmitted to the VDBM. Within the VDBM, logic associated with the equations for the requested operating parameters may decide which gear in the transmission is engaged and, therefore, which of several equations should be used. The form of equation for a given parameter may vary for different vehicles; but, again, this is transparent to the Executive because the calculations are performed within the VDBM. The appropriate flow rates are returned to the Executive.

The Executive multiplies each flow rate (expressed in liters/km or grams/km) by the element length and sends the resultant quantities to the Product Accumulators.

(6) The Product Accumulators sum individually the fuel consumed and each of the three exhaust emissions of interest (H.C., CO, and NO_x). Whenever the Executive is required to output data on these parameters, the current totals are transferred from the accumulators to the Executive without clearing the accumulators.

(The VDBM may not always have all of the desired data; namely, fuel economy or air contaminant emissions of carbon monoxide, hydrocarbons, or nitrogen oxides. For that vehicle there would be then noted zero outputs for any input lacking. When operating the HPP program one should be cautious about whether the outputs are correct or not whether only one vehicle is involved or a "fleet" of vehicles is involved, which would mean re-runs of the model and additions for each vehicle.) At the end of operations for each road element, the Executive returns to the HGP for information on the next element, and the above process iterates to the end of the road.

(7) Data output is, of course, the ultimate objective of the exercise. The Executive prints the names of the grade and speed modules, the identification of the vehicle, cumulative totals of fuel and emissions at each intermediate milepost (see following paragraph) specified along the road and at the end of the road. If the user has specified a "long-form" output, each segment of all grade and/or speed modules is listed.

The mileposts mentioned immediately above are input by the user at the beginning of operations. They may be at any arbitrary positions within the length of the highway, and may number from zero up to a maximum of 20 for a given highway. This number does not include the final output at the end of the road, which always is printed. The Executive reads the specified mileposts and delivers them to the HGP. Mileposts may be located within segments of either or both grade and speed modules, not necessarily at boundaries. The HPP monitors the distance traveled and adjusts the length of an element if necessary to coincide precisely with a milepost. At each milepost, the HGP directs the Executive to print an output.

The above sequences are iterated along the rest of the highway in normal fashion except for omission of the aborted functions, and mileposts are ignored. All subsequent occurrences of performance overload, if any, are reported in the same manner as the first. Consequently, the designer can see, as a result of the first analysis attempted, all locations where grade, curvature, speed and/or acceleration must be modified and by approximately how much, in order to operate that vehicle successfully over that road.

The above discussion summarizes the operation of the prototype version of the HPP. For clarity, it does not include the mechanics of preparing grade, speed, curvature, and milepost modules. These are outlined briefly in Section A.2 below. The present HPP requires a user to set up and run a separate case for each vehicle type to be operated on a given highway, and outputs performance data for that vehicle only. The present version of HPP only has one of the test vehicle's results (1980 Pontiac Le Mans) of fuel consumption and air contaminant emission vehicle load performance mapping and vehicle characteristics. Comparable data can be added for other vehicles tested, as indicated in the instructions provided on the CD ROM for the HPP program.

A designer may want to analyze a highway for a representative fleet. The fleet might consist of 669 Vehicle A, 632 Vehicle B, 215 Vehicle C, 170 Vehicle D, and 114 Vehicle E (a very simplified fleet has been chosen for brevity in this example). The designer then will run five individual cases for each alternative highway geometry: one each for Vehicles A through E. The output values for each vehicle then are multiplied by the number of that vehicle type in the fleet and the results summed to obtain the totals for each road configuration. Similar calculations may be performed for any selected mileposts, or for all mileposts if desired; increments between mileposts can be calculated by difference. These intermediate milepost results can, for example, isolate the effects of a particular grade.

These calculations represent the principal effort of the designer for these highway analyses, after the initial creation of grade, curvature, and speed modules. The results give the designer a realistic estimate of the effects of the alternative highway geometries on fuel consumption. The total quantities of fuel can be multiplied by appropriate fuel costs and the number of times the fleet would travel those roads per day, per year, or over the life of the highway, to obtain comparative fuel costs or savings. Comparable calculations for air contaminant emissions are possible.

A.2 INPUT DATA REQUIRED PRESENTLY

Data required presently by the HPP consist of two types: (a) mandatory user inputs, such as highway geometry, vehicle speed vs. distance, and station data for reporting; and (b) HPP-based inputs resident in the software, such as vehicle mechanical specifications and performance data, which may require no action from the user. Presently only results from one vehicle tested are resident in the software (1980 Pontiac Le Mans). Results of tests made on other vehicles may be added if testing and data are adequate.

These inputs are described below:

A.2.1 HIGHWAY GRADE VS. DISTANCE (USER INPUT)

Data that describe highway grade vs. distance are organized into "Grade Modules" that consist of 1 to 40 segments per module. Each segment is entered in km, and grade in percent (meters of rise/100 meters horizontal, or feet of rise/100 feet horizontal). An example of an unusually long segment would be the Massachusetts Turnpike (I-90) through the Berkshire Mountains in the western part of the state. The road maintains a nominally constant grade of 3.5 percent for 7.7 mi (12.4 km). If one accepts the nominal grade for the entire length, this would be entered in a grade module as a single segment, "12.4,3.5"—regardless of whether speed changed anywhere along the segment.

Any number of modules can be assembled in any sequence to define a highway. Once a module is created and stored (with a name) in computer memory, it can be recalled and reused at any future time. It also can be modified and stored to replace the original module; or modified, renamed, and stored to create a new module. For example, assume a highway 10 mi (16.1 km) long and described in five modules was being analyzed for reconstruction, and a 1.7-mi (2.7-km) length in the third module was going to have the grade changed. The elevations and horizontal distances at both ends of the 1.7-mi (2.7-km) section will remain the same after reconstruction. The user will create the initial five grade modules and store them. The third module then will be recalled, the appropriate segments changed, the module given a new name, and the new module stored. For one analysis, the first five modules will be used; for the second analysis, the modified third module will be substituted for the original third module, and the run repeated.

A.2.2 HIGHWAY CURVATURE VS. DISTANCE (USER INPUT)

The effect of cornering drag is to reduce fuel economy by about 10 percent for net lateral accelerations of 0.10-0.15 G, and the loss increases sharply with higher lateral accelerations. For modern highways, lateral accelerations above 0.1 G probably are rare in normal operation and, when found, comprise a very small percentage of road length. In such cases, the consequence of cornering drag probably can be ignored. However, older secondary/tertiary roads that exhibit high-lateral-acceleration curves (at whatever is normal speed for the road) over a significant portion of their length could be causing fuel economy penalties worth considering. If reconstruction and straightening of such roads is being contemplated, the fuel economy benefits of reducing road curvature may show a worthwhile long-term saving to the motoring public and thereby promote public support.

Calculation of this effect was added to the Vehicle Dynamics section of the algorithm. The HGP has been modified to track horizontal curvature and report this to the Executive for each road element. The Executive sends curvature and speed information to the Dynamics section along with other information that would include road load tractive force. The Dynamics section would calculate lateral acceleration and the resultant cornering-drag increment of tractive force, and include this in the summation of forces leading to net driveshaft torque.

Inclusion of horizontal curvature in the HPP required preparation of Curvature Modules to specify curvature vs. distance. The process resembled preparation of grade or speed modules. The data consists of length of curve and radius of curvature in meters. Where the segment is straight, a value of 0 is entered for the radius

A.2.3 SPEED VS. DISTANCE/IDLE VS. TIME (USER INPUT)

Data describing vehicle speed vs. distance along the road are organized into Speed Modules similar in form to Grade Modules. Within a speed module, each of 1 to 40 segments specifies either (a) an arbitrary distance and the speed at the end of the segment, or (b) the vehicle is stationary with engine idling and the duration of idle period in seconds. The HPP software recognizes which operation is described by each type of segment; they can be intermixed at will and, during modification, one type may be changed to the other.

Similarly to grade modules, speed modules may be assembled in any sequence to define a speed vs. distance profile over a highway of arbitrary length. The distances represented by any speed module, segment thereof, or string of modules need not correspond in any way to distances defined by grade modules used in a given run, with only two exceptions: the first segments of the first grade and speed modules start at zero distance, i.e., the beginning of the road; and, the total distance represented by all speed modules must equal or exceed by any arbitrary amount the total distance of the grade modules. Any speed-module distance in excess of total grade-module distance is ignored.

When assembling a string of speed modules, a user must be cognizant of the end speed of the last segment of each module and of the first segment of the following module. Remember that the starting speed of any segment of any (speed) module implicitly is the end speed of the preceding segment. Inattention to this matter could produce unrealistic and unattainable performance demands.

A.2.4 REPORT STATIONS FOR INTERMEDIATE OUTPUTS (USER INPUT)

HPP automatically prints the total fuel consumption (and total exhaust emissions, if data are available) at the end of each highway run. However, the user may want, in addition, to isolate and determine the effects of one or more particular portions of highway within the run. To facilitate this, the user is required to enter the filename for a file containing desired stations along the roadway for reporting; at each station, cumulative fuel consumption (and emissions) to that point will be printed. Stations can be set at the beginning and end of increments of interest; the difference in totals after and before the section comprises the subtotal for that section. The user may specify from zero to 20 mileposts at arbitrary distances from the beginning of the road, and need not include the end.

A 2.5 VEHICLE DATA (FOR HPP)

Data on the pertinent physical characteristics and performance of a representative vehicle (1980 Pontiac) are included in software for the HPP It was intended that data for each vehicle tested in this study would be incorporated into VDBMs and that these modules could be stored in the HPP package and called up individually by name. However, this was not done. A small effort is required to extract the data and equations for each vehicle tested and create named modules. Then one or more modules could be called up by specifying the file name or names.

The experimental data base described in Section 3.8 contained information on only three cars; this did not provide much variety of vehicle type for use with the HPP. Methods have been devised by which fuel economy and air contaminant emissions data might be synthesized for other vehicles that had not been actually tested in this study. VEHSIM, a model developed by the Volpe National Transportation Systems Center, could be used for such syntheses. (See Section 2.0).

For additional vehicles tested, particularly those having notably different performance, if their data are in a format suitable for use by the HPP, these data can be added to the present vehicle data base. No changes in HPP software will be required. To create a new VDBM, a user need only follow the format of any existing VDBM and assign an appropriate file name. The HPP User's Guide contains a format specification sheet for VDBM file names. (See Sections 2.0 and A.1.5-6.)

It should be noted that, within a new VDBM, equations of virtually any form may be used to describe the data, and logic constraints can be used to select specific equations for different operating ranges. All of these functions must be contained within the VDBM. The requirement for compatibility is that the VDBM operate with no more data than is sent by the Executive (although some such data may be left unused if not needed), and that the VDBM return to the Executive the requested information in the correct units.

A.3 OPTIONS FOR FUTURE EXPANSION

The FHWA or some other prospective user will have to determine whether technical community interest and applications of this prototype HPP software warrant further effort to increase its capabilities. Useful expansion, as seen at this time, could improve user convenience and add more vehicles to the vehicle data base.

Features recommended in future versions include the following:

- Replace or append the methods for inputting speeds, grades, and curves to enable input in the format normally found on plans (e.g., vertical information input by P.I. station, elevation, and vertical curve length).

- Revise the analysis of performance limits to analyze the data in the vehicle file format. The current program will only display a warning if a value is outside of the minimum and maximum range of any of the table values.

- Improve program performance to enable smaller increments of distance to be analyzed. The current program recalculates many values for the sake of readability for other programmers. The cumulative data can be calculated at the beginning of the analysis section and saved, thereby speeding up the plotting and listing sections.

Features not recommended for addition include road surface roughness and a traffic model. Roughness has been found to have an almost negligible direct effect up to the condition that justifies resurfacing for safety and driver comfort. A traffic model would provide an objective estimate of traffic speed, and would relieve the designer of this responsibility, but it would require a major and expensive effort, would tend to duplicate the capabilities of some existing traffic models, and its validity would be uncertain. If such a feature is desired, it is recommended that an existing model be used in its present form to generate a speed-distance profile. That profile could serve as the basis for generating speed modules in the HPP.

The hand calculations required when a user of the present HPP wants to consider a fleet composed of a mixture of different vehicles are described at the end of Section A.1.5 above. The computations are elementary in concept but require some time and effort; also, some potential for human error exists in correctly associating corresponding data items. Further, the results are not printed concisely and directly on the HPP data printout.

APPENDIX B
PROTOTYPE VEHICLE/HIGHWAY PERFORMANCE PREDICTOR ALGORITHM

HPP algorithms were developed for a mainframe computer at Volpe National Transportation Systems Center with FORTRAN. At a later date, a Visual Basic user-friendly interactive version of the vehicle highway performance predictor algorithm, HPP, for use on personal computers was developed. The software and installation instructions as well as the source code have been placed on a CD ROM. Instructions for adding data, such as for other vehicles, are also included on the CD ROM.

For added development of the HPP, it is suggested that the programmer acquire Microsoft Visual Basic 6.0, Professional or Enterprise Edition.

The following is pertinent text on the CD ROM:

HPP - Highway Performance Predictor Program
Alpha Release
Version 1.001

July 26, 2000
Installation Information:

This information supercedes or is in addition to information included in the following files:

Installation:
Double clicking on subtypes will begin the installation program to install the HPP program.

Program Limitations:

In order for the plot screens to print you must have your computer monitor resolution set to 1024 X 768.

The program currently does not adequately check to see if the combination of speed, geometry and acceleration are beyond the performance capability of the vehicle. Extremely unrealistic values for speed or grade will cause an error message but it is possible to get successful runs where the needed acceleration exceeds the capabilities of the vehicle. For this version of the program it will be up to the user to make sure that the demands on the vehicle are realistic.

When selecting different views of the data, the new information is appended to any text already in the viewing area. This means if you don't use the clear data option you won't see the new data without scrolling down. This was done to enable the user to have multiple reports saved in one output file.

There is no print option for the data view. The user needs to save the file to a new file name and use another program (e.g. Notepad or Wordpad) to print the data.

There is a limit to how much text can be shown in the view screen. If it exceeds approximately 64 k the output will be truncated. This should not be a problem except for cases where there are a very large number of different vehicles. In this case it may be necessary for the user to run individual vehicle runs to get the vehicle data.

Included files:

In the root directory of the CD the following files are necessary for installation:

subtypes
Hpp.cab
Setup.lst

The file Hpp.dep is not necessary. It was provided to show all of the files that must be installed in order to run the H.P. program. These files should reside in the system directory of your computer.

For Win95/98 machines this will most likely be:
C:\Windows\System\

for WinNT machines this will most likely be:
C:\WINNT\system32\

These files must all be registered. This is done automatically through the installation program. If you encounter problems getting HPP to run, checking whether the system files are registered may help solve the problem.

Data Files

The directory hpp_dat should be copied in its entirety to the same directory where the file HPP.exe resides. It is recommended that you browse these files with a simple text editor/viewer to become familiar with the format. The file Data.hpp was provided for illustrating the format of a saved data set of files. It is recommended that you create your first data set file through the program rather that trying to edit the path and file name information of this file.

Please be aware that the order that the files are selected is the order in which the distance segments are analyzed.

Also please note that the Lemans80.dat file is the only file that represents actual measurements; the other files are copies of this file for testing the program.

Other Documentation Files

The following files are included in the documents directory:

Original Installation file explaining computational methods, file structures, and construction of vehicle data files:

Word Perfect 8 format --
Highway Performance Predictor Software Installation and Users Guide.wpd
This includes directions for adding other vehicle data bases.

MS Word 97 format --
Highway Performance Predictor Software Installation and Users Guide.doc

Research Report --
Highway Effects on Vehicle Performance.wpd or
HEFFVPER.070.wpd

Notes on Documentation

1. The HPP program has been updated to a modern object oriented program. Consequently the report documentation describing the program operations is approximate and details and amplifications or changes are provided on the CD ROM.

2. The HPP program expects input entirely in metric format. Caution is advised regarding English units, for example: Mileposts, gallon, etc.

3. Based on recommendations from users, There may be enhancements to the program. These may include:

Program Functionality:

- Issues such as exceeding the vehicle performance capability may be addressed in a more comprehensive manner.

- Ease of use issues, such as saving default file paths, may be addressed if there is a demand.

File Formats:

- The file formats may be changed to better incorporate inputting information. For instance:

  - Input vertical curve information by station rather than individual grades by distance.

  - Input horizontal curve information by station rather than distance.

  - The vehicle file format may change to enable rapid file construction from other research data.

Source Code

The Visual Basic 6.0 (Professional Edition) source code is included in the source directory. If you wish to run this program from the VB IDE you should copy all of these files to a directory and load HPP.vbp. No third party tools were used in this program. It is also recommended that you copy the data files directory to the same directory where HPP.vbp resides.

APPENDIX C. EMPIRICAL TEST DATA

This appendix presents summaries of test data on the three vehicles that were fully tested (1980 Pontiac, 1980 Chevette, and 1981 Citation). Development of information from the tests is illustrated, first, by following one vehicle (Citation) through the procedure—except that one parameter, tractive force loss, which was determined only for the Pontiac, also is included in the one-vehicle group. Subsequently, comparable data (excluding tractive force loss) for the three vehicles are presented together.

C.1 EMPIRICAL DATA: DEVELOPMENT FOR ONE VEHICLE

C.1.1 ROAD LOAD VS. SPEED FROM RUNWAY TESTS—CITATION

Figure 3 illustrates one of the methods used to determine road load in terms of driveshaft torque from vehicle tests on a runway with a constant slight grade. Experimental data points are plotted for runs in the uphill and downhill directions at speeds from about 45 km/h to 120 km/h (28 to 75 mi/h); a total of six round trips (12 tests) was run. Replications were made at the two highest speeds to obtain a limited assessment of repeatability; the lower-speed tests were not replicated, partly because these speeds tied up the runway for excessive times. Further, the number of data points collected per test ranged from 340 to 580; it was felt that this number of points provided a sufficient body of data to permit multiple analyses of subsets of the entire file.

Additional data were desired, at speeds much lower than used on the runway, to guide regression analyses at speeds approaching zero. These were intended to eliminate (if possible) the tendency of earlier regressions, for other vehicles, to show torque passing through a minimum at speeds in the 20 km/h region and then increasing as speed decreased toward zero. Such behavior was considered a mathematical artifact that resulted from the absence of data at low speeds to guide the regression in that area. From the curve-torque tests (described below), actual experimental data were obtained at lateral accelerations from 0.10 down to 0.07 G (29 to 14 km/h), and extrapolated from regression equations for lateral accelerations from .04 to .01 G (22 to 6 km/h). These data are plotted in the lower right corner of figure 3.

The runway data were regressed first for uphill and downhill directions separately to evaluate the consistency of the individual data sets. The resultant equations are shown in figure 3 for each direction of travel, together with the coefficients of determination (R^2). Next, the same raw runway data for both directions were combined in a single regression together with the very-low-speed data. This produced the composite equation and graph for level road, shown in figure 3 between the uphill and downhill graphs. The combination of uphill and downhill runs should cancel the effect of the constant 0.58 percent grade; the circle test data on a 0.79 percent slope were averaged around complete circles and also should eliminate the grade effect; hence, this composite equation should be a valid description of level-road performance.

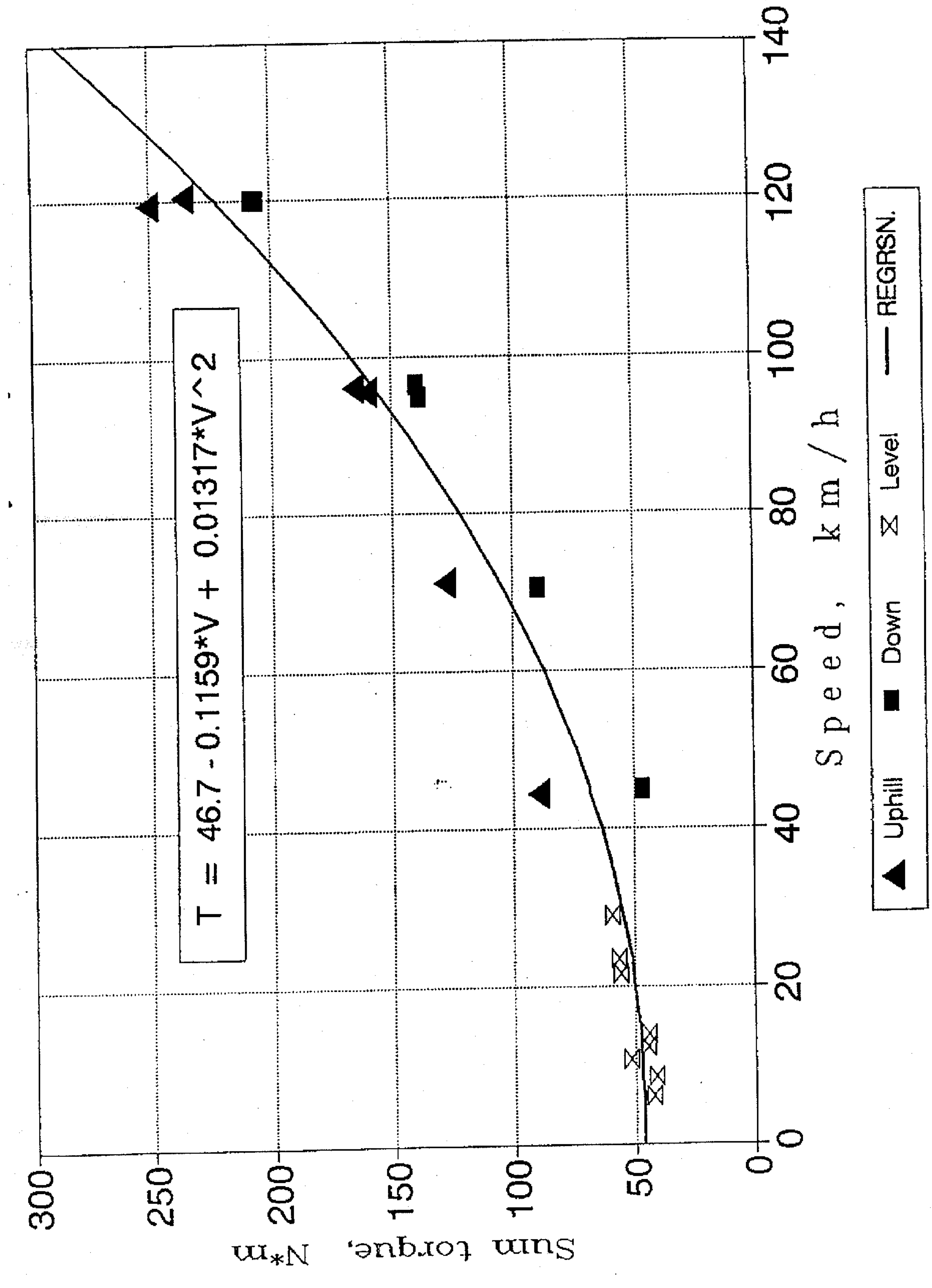


Figure 3. Citation road load torque vs. speed.

Table 2. Citation road load vs. speed, from runway tests.

(See figure 3)

REGRESSION EQUATION:

ROAD LOAD TORQUE, T_{RL}, (N.m) = $a_0 + a_1 V + a_2 V^2$ (V in km/h)

DATA:

GRADE	a_0	a_1	a_2	R^2
UPHILL (exptl.)	114.6	-1.4105	0.02054	0.982
DOWNHILL (exptl.)	14.13	0.2110	0.01159	0.999
LEVEL (ALL DATA, COMPOSITE)	46.7	-0.1159	0.01317	0.960

C.1.2 TORQUE INCREMENT ON CURVES—CITATION

The increase in driveshaft torque on curves at any given speed, additive to road load torque on a straight, level road at the same speed, was derived from experimental data as shown in figures 4 and 5. Most of the development is illustrated in figure 4. The data points associated with the upper three curves in figure 4 show the individual mean values of total shaft torque (for two shafts) on the three circle radii [100, 200, and 300 ft (30.5, 61.0, and 91.5 m)] plotted against lateral acceleration. The lateral accelerations for the points were calculated from mean speed and nominal curve radius. While total torque was not the parameter of primary interest, these values were regressed against lateral acceleration to investigate the consistency of the data. Graphs of the regression equations for the three circles are drawn through the data points, and the coefficients of determination (r^2) are listed for the corresponding curves. (In this figure, lower-case r^2 was used, instead of the upper-case R^2 shown elsewhere in this report, to avoid confusion with curve radius R in this figure.) The r^2 values of over 0.99 for all three circle radii indicate a high degree of consistency of the raw experimental data; this quality of results was observed for all three cars tested.

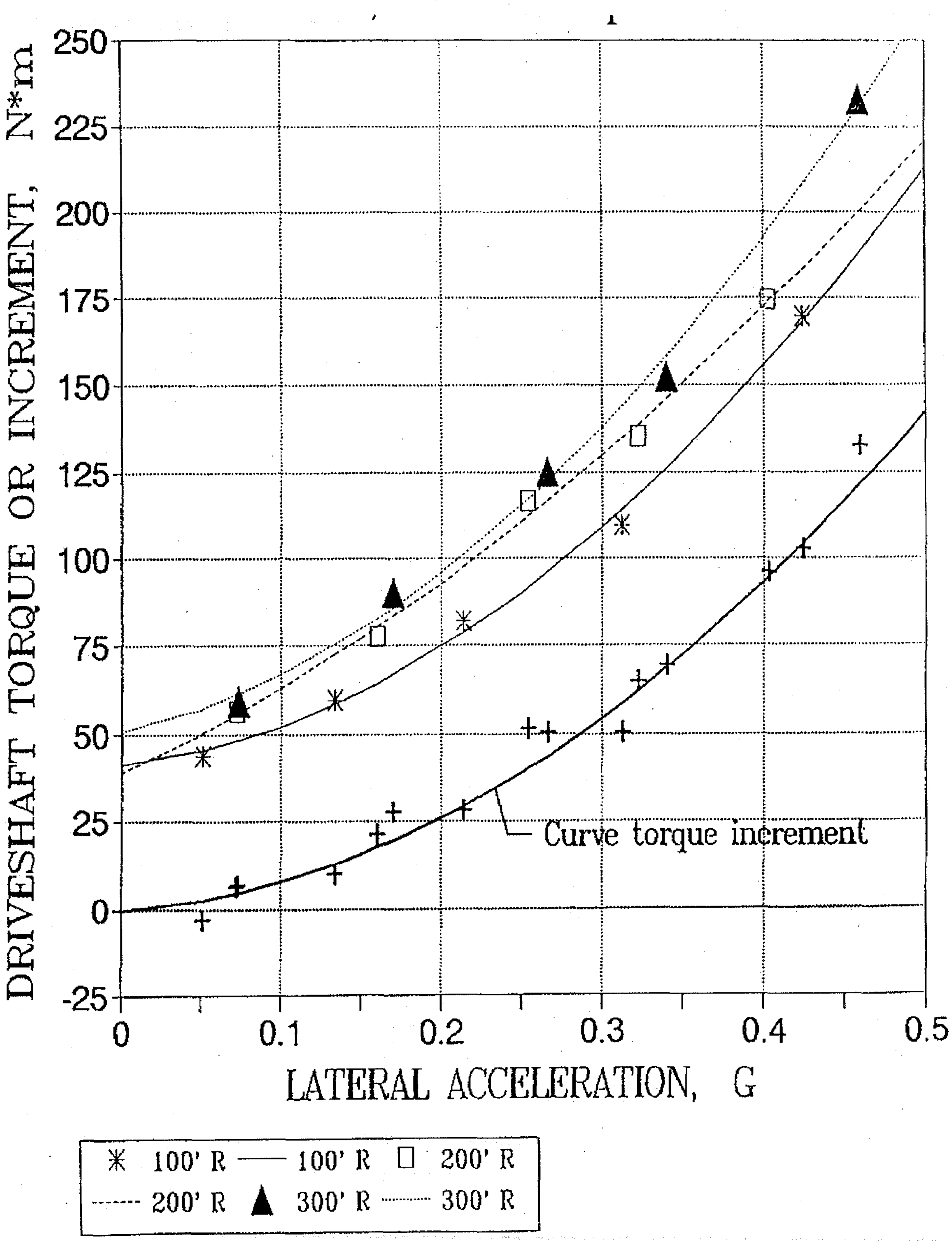


Figure 4. Total torque and torque increment on plane curves.

Figure 4 addendum

r^2 = coefficient of determination

radius 100 ft	– 0.996	(30.5 m)
radius 200 ft	0.995	(61.0 m)
radius 300 ft	0.996	(91.5 m)

Torque Increment, $\Delta T_{c,}$ on curves

$\Delta T_{c,}$ N.m =

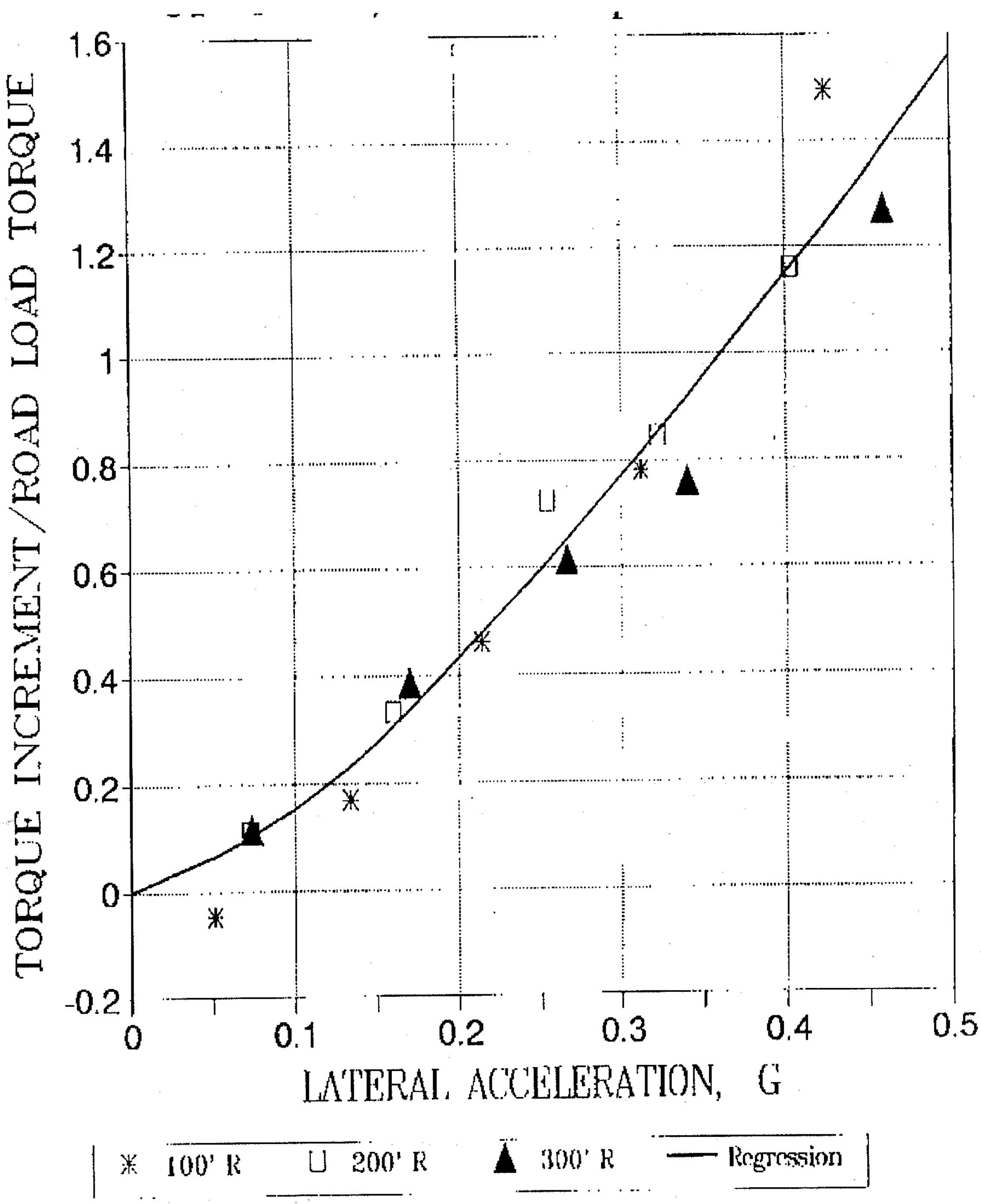
$$28.1\, A_{Lat} + 512\, (A_{Lat})^2 : r^2 = 0.986$$

Derivation of the curve-torque increment, Δ Tc, is shown graphically at the upper right of figure 4 for the highest-speed run on the 300-ft radius. At the mean speed of 73.0 km/h, the road load equation (Section C.1.1 and figure 3) yields a road load torque, T_{RL}, of 108.4 N.m. When this is subtracted from the total curve torque of 233.0 N.m, the result is $\Delta Tc = 124.6$ N.m. The values of ΔTc for all the other test points are plotted in the lower part of figure 4. The regression equation for these incremental torques is given in the figure 4 addendum. There is an r^2 of 0.986. Clearly, these three independent sets of data from different circle radii appear to collapse into one self-consistent population. This is in agreement with the analysis of Segel and Lu (3), which shows that the incremental curve torque, expressed in units of torque, is independent of curve radius. Note that these data for torque-increment on curves in figure 4 still are vehicle-specific; these are increments of torque for a 1981 Citation on various curves. For the purposes of this project, it was desired to find a relationship that would be applicable to a variety of different makes of automobiles, at least. This objective is explored in the next section.

C.1.3 NORMALIZED TORQUE INCREMENT ON CURVES—CITATION

The possibility of eliminating the vehicle-specific nature of curve-torque increment data was explored by "normalizing" the torque increment. To normalize each curve torque increment, ΔTc, it was divided by the road-load torque, T_{RL}, for the same vehicle at the same speed, to yield the dimensionless ratio, $\Delta T_c/T_{RL}$. The results for the Citation are plotted in figure 5 against lateral acceleration as the independent variable. As in figure 4, each point is plotted with a symbol showing the radius of its circle.

It is evident in figure 5 that the data no longer define a single curve; rather, the data tend to follow separate curves corresponding to the three circle radii. This is consistent with the prediction of Segel and Lu's analysis (3) that this normalized curve-torque increment is dependent upon curve radius; such behavior is in contrast to the previous torque-units relationship that was independent of curve radius. If Segel and Lu's analysis were followed rigorously, the advantage of the normalized incremental torque would be lost, to a large degree; the predicted relationship is dependent upon both curve radius and the cornering compliance of the tires, in addition to the desired lateral acceleration parameter. The cornering compliance coefficient is a function of tire design, tread depth, and inflation pressure; for highway analyses, the user generally will not know this coefficient. However, if one desires to merge these data into a single approximate relationship, the scatter is not extreme; regression of all the data yields the curves plotted in figure 5 with a coefficient of determination, R^2, of over 0.89 (0.893). More importantly, the scatter of the experimental points away from the regression line becomes insignificant below



0.305 m = 1 ft

Figure 5. Normalized curve-torque increment vs. lateral acceleration.

Figure 5 Addendum

$A_{Lat} \rangle$ 0.147 G:

$$\Delta T_c / T_{RL} = -0.1412 + 2.495 A_{Lat} + 1.812 (A_{lat})^2;$$

$r^2 = 0.893$ (for all test data)

$0 \le A_{lat} \le 0.147$ G:

$$\Delta T_c / T_{RL} = 0.957 A_{Lat} + 5.695 (A_{Lat})^2$$

lateral accelerations of about 0.2 G. Normal highway design generally avoids such high accelerations, especially in terms of net lateral acceleration after compensation by superelevation (these empirical data were taken on plane curves). Further, the relationship shown in figure 5, a composite of data from radii of 100, 200 and 300 ft (30.5, 61, 91.5 m), will yield a conservative estimate of curve effect on most curves (i. e., the calculated torque increment will be somewhat larger than actual). This occurs because the incremental normalized torque at any lateral acceleration becomes smaller as curve radius increases, and most highway curves will have radii greater than the 200-ft (61-m) radius of the above composite regression.

In view of the above, for the purposes of this project, it was considered a reasonable and practical approximation to use the regression function shown in figure 5 as a single-variable description of normalized curve-torque effect. One additional adjustment was required; the regression function indicates negative torque increments, i.e., a curve torque less than straight-road torque at lateral accelerations below about 0.05 G. This clearly is inconsistent with the physics of the situation (unless some peculiar phenomena occur at very small steering angles), and is not apparent in figure 4.

The anomaly probably was the result, primarily, of small experimental errors; the torque increment, ΔTc, comprises the small difference of two relatively large and nearly equal quantities, especially at low lateral accelerations (cf, figure 4). Consequently, the lower part of the regression curve in figure 5 was modified by trial and error to follow a path through zero increment at zero lateral acceleration (i.e., at 0,0) while blending smoothly with the regression curve at some point. For this vehicle, the transition point was at 0.147 G lateral acceleration. The derived function for low accelerations is not shown in figure 5.

Alternatively, if one wishes to investigate rigorously the forces acting on a specific vehicle on curves, the data for different, known curve radii may be regressed separately for each radius. Data for different radii then can be used to determine the cornering compliance coefficient for the specific tires on the vehicle.

C.1.4 FUEL ECONOMY VS. TORQUE, SPEED, AND GEAR—CITATION

The results of all Citation fuel economy mapping tests are summarized in figure 6. It must be recognized that this kind of figure is like a road map; it presents and relates a large amount of information very concisely, and requires some time to be absorbed. In figure 6, fuel economy at various speeds and torques is regressed against only torque; where the data indicated the necessity, the regression was performed separately for individual gears. Regression equations and coefficients of determination (R^2) are given in table 3 for the three fuel economy curves plotted. The figure shows that fuel economy tends to be a highly non-linear function of torque in the range from moderately positive to large negative values.

No measurements of fuel economy were made in first gear, partly because the torque amplifier gain had been set relatively high to give adequate resolution at road load levels and would cause saturations at high first-gear torques. Further, divergence of fuel economy in first gear from the curves for other gears would have been significant mainly at low positive to high negative torques, which comprise a negligible fraction of normal vehicle operation. Testing in this gear was omitted; if vehicle/highway analysis is required in this area (e.g., for traffic signalization studies), second-gear data can be used with insignificant error overall.

Table 3. Citation fuel economy vs. driveshaft torque and gear.

(See figure 6)

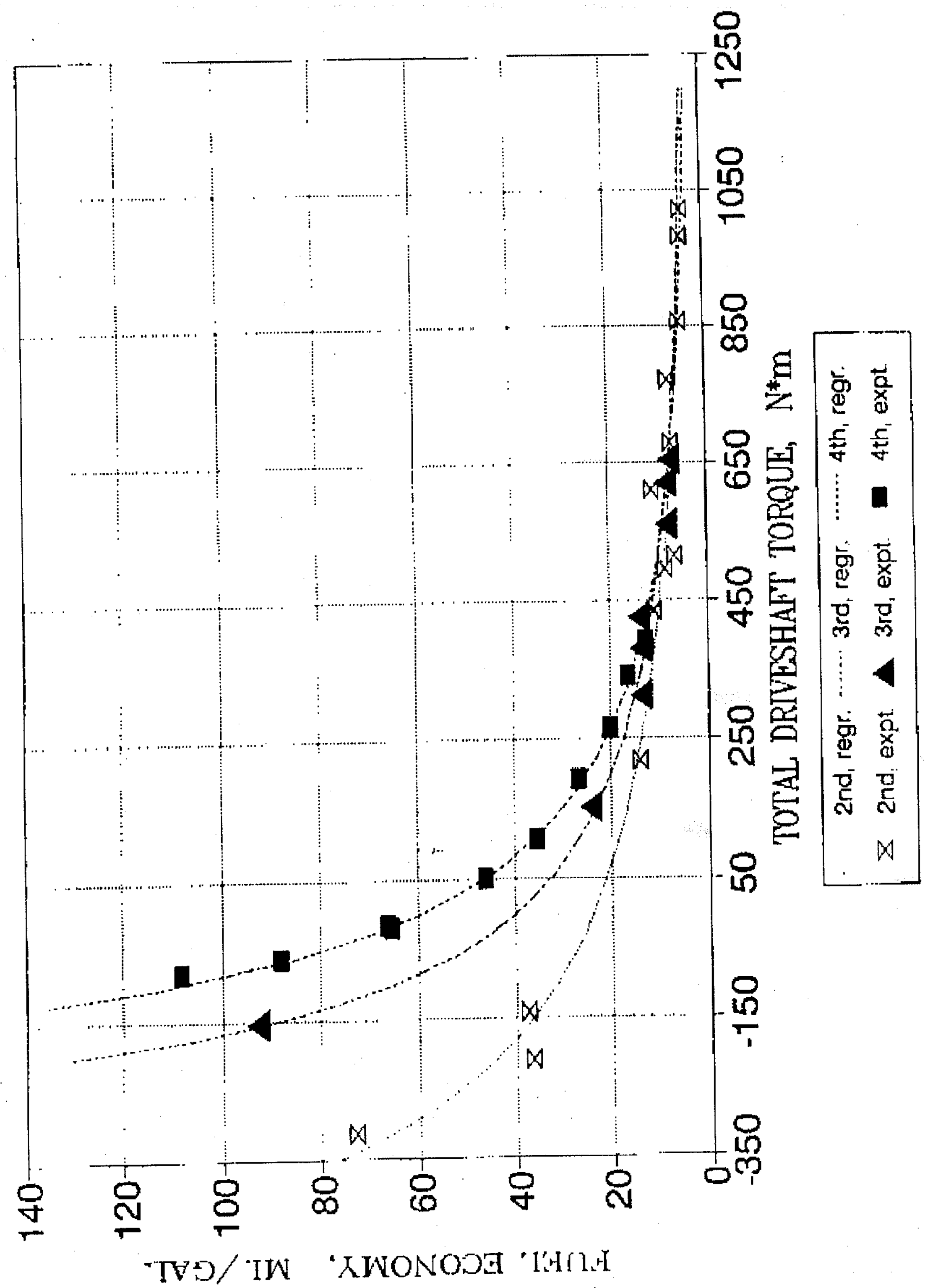
REGRESSION EQUATION:

FUEL ECONOMY, FE, MPG = $a\,(T/2 + b)^c$ (T in N.m)

1 mi/gal =.42 km/L

DATA:

GEAR	SPEED RANGE km/h	COEFFICIENTS a	b	c	R^2
4th (high), + 3rd & 2nd for T > 360 N.m	95 - 120	5.96E06	200	-2.1708	0.978
3rd	55 - 120	3.39E05	200	-1.7072	0.990
2nd	20 - 75	4.58E06	400	-2.0306	0.961
1st	Not tested; see text				



1 mi/gal =.42 km/L

Figure 6. Citation fuel economy vs. driveshaft torque and gear.

C.1.5 TRACTIVE FORCE LOSS IN DRIVE AXLE AND TIRES—PONTIAC

This section deals with an analysis of data made only for one vehicle, the 1980 Pontiac; therefore, it is reported in the "Data for One Vehicle" category although the rest of this appendix portion addresses tests on the 1981 Citation.

The exploration of the dissipation of tractive energy in drive axle and tires as a function of total propulsive torque (in driveshaft) or tractive force (at drive tire surface) was discussed succinctly but adequately in Section 3.7 of the main report. The empirical results of this limited study are shown in figure 7. The original data were taken in terms of driveshaft and dyno torques. However, in figure 7, these data have been transposed into tractive force at the dyno rolls and the loss of tractive force from the quantity corresponding to driveshaft torque.

From a total of 63 data points spanning 7 speeds from 15 to 117 km/h, only 8 points are outliers and excluded from the regression. These eight points are evident in figure 7 since they are most distant from the regression curve. The agreement of the bulk of the experimental data with the regression curve can be seen in the graph in figure 7. The equation of the regression curve, and the coefficient of determination, are shown in the figure.

This completes the discussion of experimental data for a single test vehicle. In the following sections, similar final results for two additional cars are presented and related to the vehicle discussed above.

C.2 EMPIRICAL DATA: COMPARATIVE DATA ON THREE CARS

The preceding sections (C.1.1 - C.1.5) discussed the development of test data for a single test vehicle. This section will append similar data for two other vehicles and, where feasible, combine the data for these vehicles into a single relationship.

C.2.1 DERIVED TRACTIVE FORCE FOR THREE VEHICLES

Section C.1.1 and figure 3, presented earlier, described the determination of road load vs. speed, in which road load was expressed in terms of driveshaft torque. In general, for any given vehicle, propulsive demand can be expressed equally well in terms of either driveshaft torque or tractive force. Within the VNTSC test program for a particular vehicle, distinct advantages resulted from using driveshaft torque; this was a primary operating parameter that was measured both on the road and on the chassis dyno. Therefore, there was no uncertainty in relating the two test conditions, as there could have been if a value of tractive force had been calculated from a measured driveshaft torque on the road and from a dyno torque on the dyno.

However, when such data for different vehicles are to be compared, it almost always is necessary to use tractive force as the operative variable; e.g. for two RWD cars such as the Chevette and the Pontiac, both the final drive (differential) ratio and the drive wheel radius affect the relationship between driveshaft torque and delivered tractive force. For the Chevette, these quantities are 3.70:1 and 0.280 m, respectively, and result in a theoretical conversion factor from torque to force of 3.70/0.280 = 13.21 m^{-1}; corresponding values for the Pontiac are 2.41/0.314 = 7.68 m-1. Consequently, equal shaft torque in the two cars produces calculated tractive forces in the ratio of 13.21/7.68 = 1.72:1. The difference is even greater when the FWD Citation is compared with either of the other two; torque is measured in the half-shafts of the Citation,outboard of the differential. Here, the effective differential ratio is 1.00:1, and the comparable conversion factor is 1.00/0.302 = 3.31 m^{-1}. The quantity calculated by use of this factor, presumed to be the road load tractive force, actually is a derived quantity; it is measured within the self-propelled vehicle and should represent the sum of all propulsive loads known and unknown.

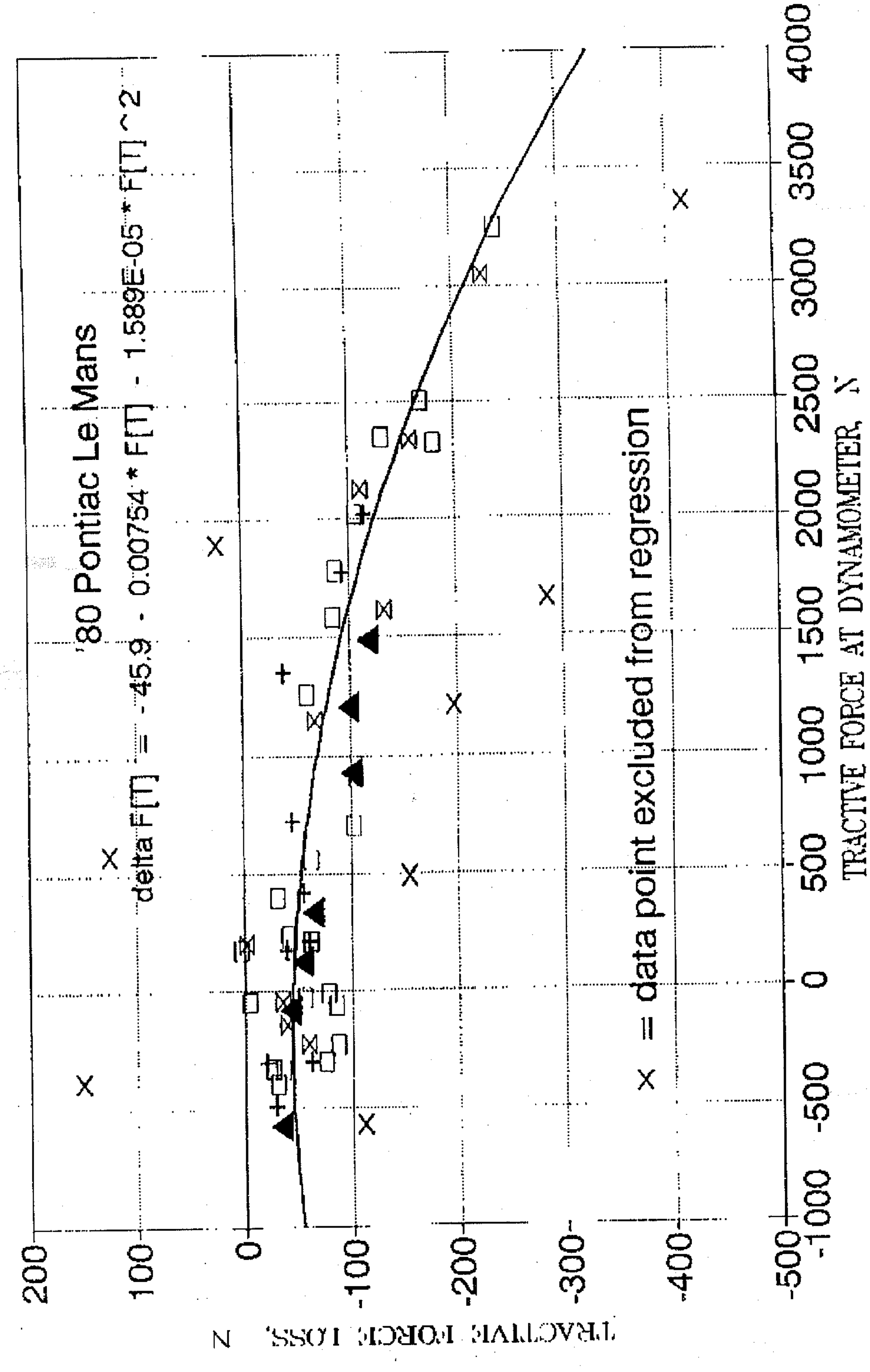


Figure 7. Tractive force loss vs. total force.

However, it has not been compared experimentally with the tractive force determined for this vehicle by coast down as many other experimenters would measure this parameter. The precise degree of equivalence between VNTSC's derived tractive force and the more conventionally measured value is not known; accordingly, the VNTSC parameter is labeled "derived tractive force."

Figure 8 presents graphs of derived tractive force in newtons (N) vs. speed in km/h for the three passenger cars characterized under this project. The source data for these curves are given in table 4; these include the road load torque equations, the equations for converting from torque to derived tractive force, the coefficients for the torque equations found by regression, and the operative axle ratio and wheel rolling radius values.

Table 4. Derived tractive force for three passenger cars.

(See figure 8)

EQUATION FOR ROAD LOAD DRIVESHAFT TORQUE:

$$T_{RL}, \text{N.m} = a_0 + a_1 V + a_2 V^2 \quad \text{(V in km/h)}$$

EQUATION TO CONVERT DRIVESHAFT TORQUE TO TRACTIVE FORCE:

$$F_T, \text{N} = T_{RL}\, AR/R_{wheel}$$

DATA:

	PONTIAC	CHEVETTE		CITATION
Speed range, V, km/h	0 - 140	0 - 42.8	42.8-140	0 - 140
Coefficients:				
a_0	19.33	11.0	15.48	46.7
a_1	0.0800	0.05833	-0.1277	- 0.1159
a_2	0.00579	0.00220	0.00410	0.01317
Correlation coeff., R^2	------	0.999	0.97	0.960
Axle ratio, AR	2.41	3.70	3.70	1.00 *
Rolling radius, R_{wheel}, m	0.314	0.280	0.280	0.302

* Final drive ratio not applicable because torque was measured outboard of differential.

C.2.2 NORMALIZED CURVE TORQUE INCREMENT FOR THREE CARS

Figure 9 presents graphs of the regression equations of normalized curve-torque increment for each of the three cars. Also shown is a composite curve combining the three vehicles into one relationship that is proposed to be used for all modern passenger vehicles unless curve-test data for a specific vehicle are available. The regression coefficients for each of the three cars, and for the composite equation, are given in table 5. It will be noticed, in table 5, that each car has two sets of coefficients for specified ranges of lateral acceleration.

Table 5. Normalized curve-torque increment: comparison of three passenger cars.
+
(See figure 9)

REGRESSION EQUATION:

$$\Delta T_c/T_{RL} = a_0 + a_1 (A_{Lat}) + a_2 (A_{Lat})^2$$

DATA:

VEHICLE	RANGE		COEFFICIENTS			
	$\leq A_{Lat} \leq$		a_0	a_1	a_2	R^2
CHEVETTE	0	0.25	0	2.165	1.357	---
	0.25	0.6	0.687	1.692	2.15	0.837
PONTIAC	0	0.3	0	0.427	4.96	---
	0.3	0.6	0.188	-0.758	6.82	0.727
CITATION	0	0.147	0	0.957	5.695	---
	0.147	0.6	-0.1412	2.495	1.812	0.893
COMPOSITE	0	0.6	0	1.349	3.37	0.986

As discussed in Section C.1.3, the two sets of coefficients were necessitated by the fact that regressions of the test data did not pass through (0,0). Therefore, in each case, it was necessary to fit a supplemental curve to the lower end of the data to pass through (0,0), since the physics of the phenomenon require this behavior.

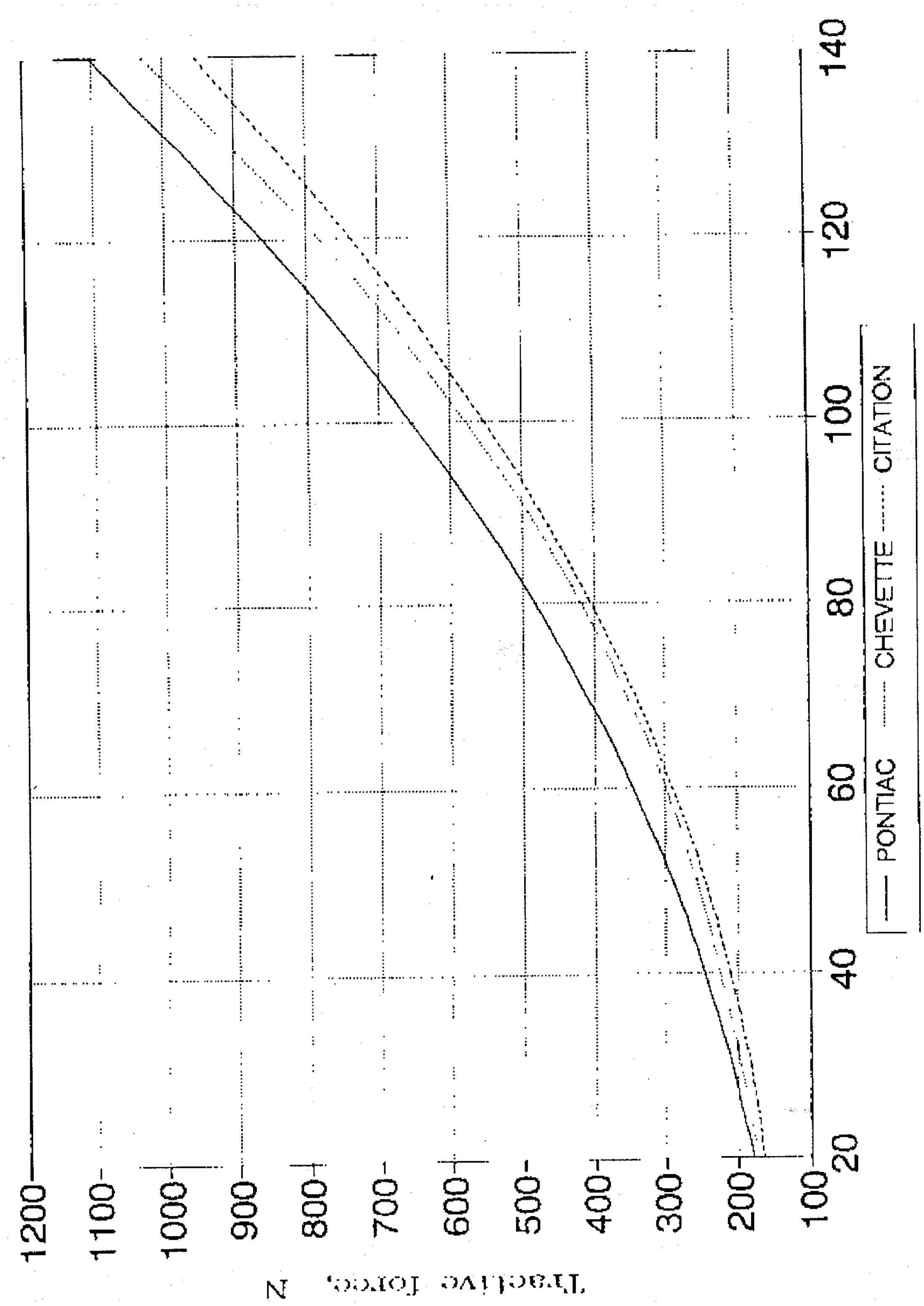


Figure 8. Road load derived tractive force vs. speed - three cars.

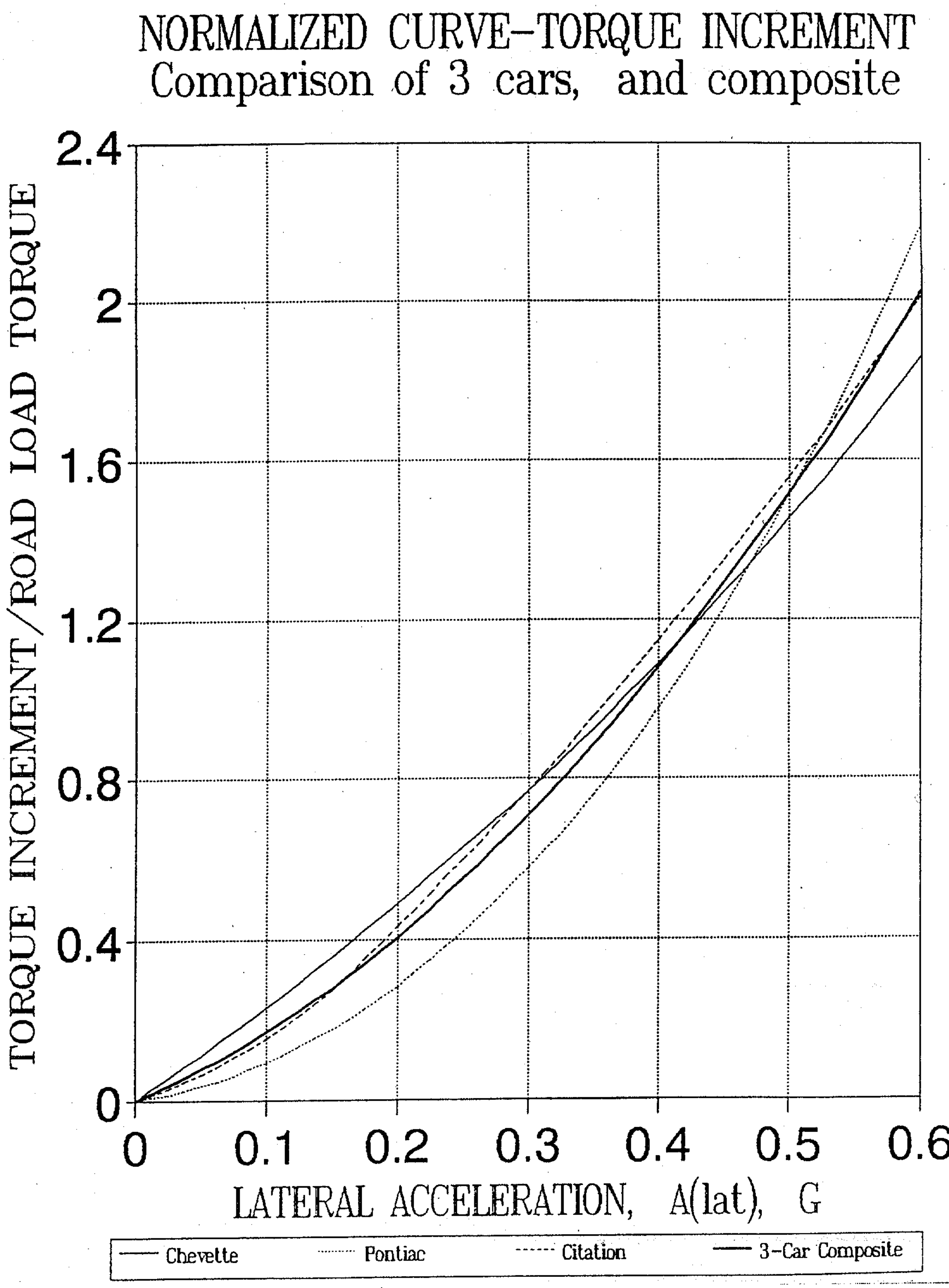


Figure 9. Normalized curve-torque increment - three cars.

The composite equation was determined by regression of values calculated from each of the individual regression equations at corresponding values of lateral acceleration. In this case, it can be seen that the overall composite equation does pass through (0,0) without adjustment. The coefficient of determination for this overall composite equation, $R^2 = 0.986$, is not indicative of the dispersion of original data points; rather, it is a measure of the agreement between the three individual regression curves. The composite normalized curve-torque-increment curve has been repeated, and the regression equation is listed in figure 9 to facilitate reading T_c/T_{RL} from a graph rather than calculating a value (figure 10).

In relation to this agreement between the three cars, it is interesting to recall the prediction of Segel and Lu (1) that cornering drag is independent of RWD or FWD configuration. Comparison of the curves for the Chevette (RWD) and the Citation (FWD) in figure 9 shows that the two curves remain close to each other throughout the test range.

The overall composite function of normalized curve-torque increment can be used to estimate a curve-torque increment for a vehicle that has not been curve-tested, if the road load relationship for that vehicle has been determined. The normalized torque increment, T_c/T_{RL}, is dimensionless; it can be applied equally well to driveshaft torque or to tractive force. The computation sequence would be as follows:

1. The road load force or torque is calculated for the speed of interest.

2. The lateral acceleration at that speed and road curve radius is computed.

3. ÄT_c/T_{RL} can be found either from its equation or by reading its value off a graph of this function vs. lateral acceleration.

4. The cornering-drag increment is found by multiplying road load or force by T_c/T_{RL}.

5. The total propulsive force or driveshaft torque is found by adding the quantities found in Steps 1 and 4.

C.2.3 FUEL ECONOMY FOR TWO ADDITIONAL VEHICLES

Section C.1.4 and figure 6 presented fuel economy data for the 1981 Chevrolet Citation. This section contains similar data for two additional vehicles, the 1980 Chevrolet Chevette sedan and the 1980 Pontiac sedan.

In the case of the Chevette, the very wide dynamic range of fuel economy values for the different gears precluded effective exposition by combining data for all gears on a single graph, as was done in figure 6; rather, it was necessary to use a different graph for each gear (figures 11 to 14). As in Section C.1.4, the general form of regression equation is given, along with individual sets of coefficients and coefficients of determination, also before in a separate table, table 6. It will be seen that the form of equation for the Chevette is identical to that for the Citation.

The 1980 Pontiac was the first vehicle for which a detailed fuel economy map was prepared over the full range of positive and negative torques. This vehicle, like the 1975 Dodge Dart sedan used for prototype investigations, was equipped with an automatic transmission and torque convertor. In nearly all of the dyno tests for measurement of fuel economy, the transmission was allowed to select the gear in which to operate; only in a few cases where "hunting" between two gears was experienced was the gear selector lever used to force operation in a specific gear. This vehicle tended to group nearly all fuel economy data reasonably closely along a single curve, as had been observed initially for the Dodge Dart (unlike the more clearly separated curves for different gears in

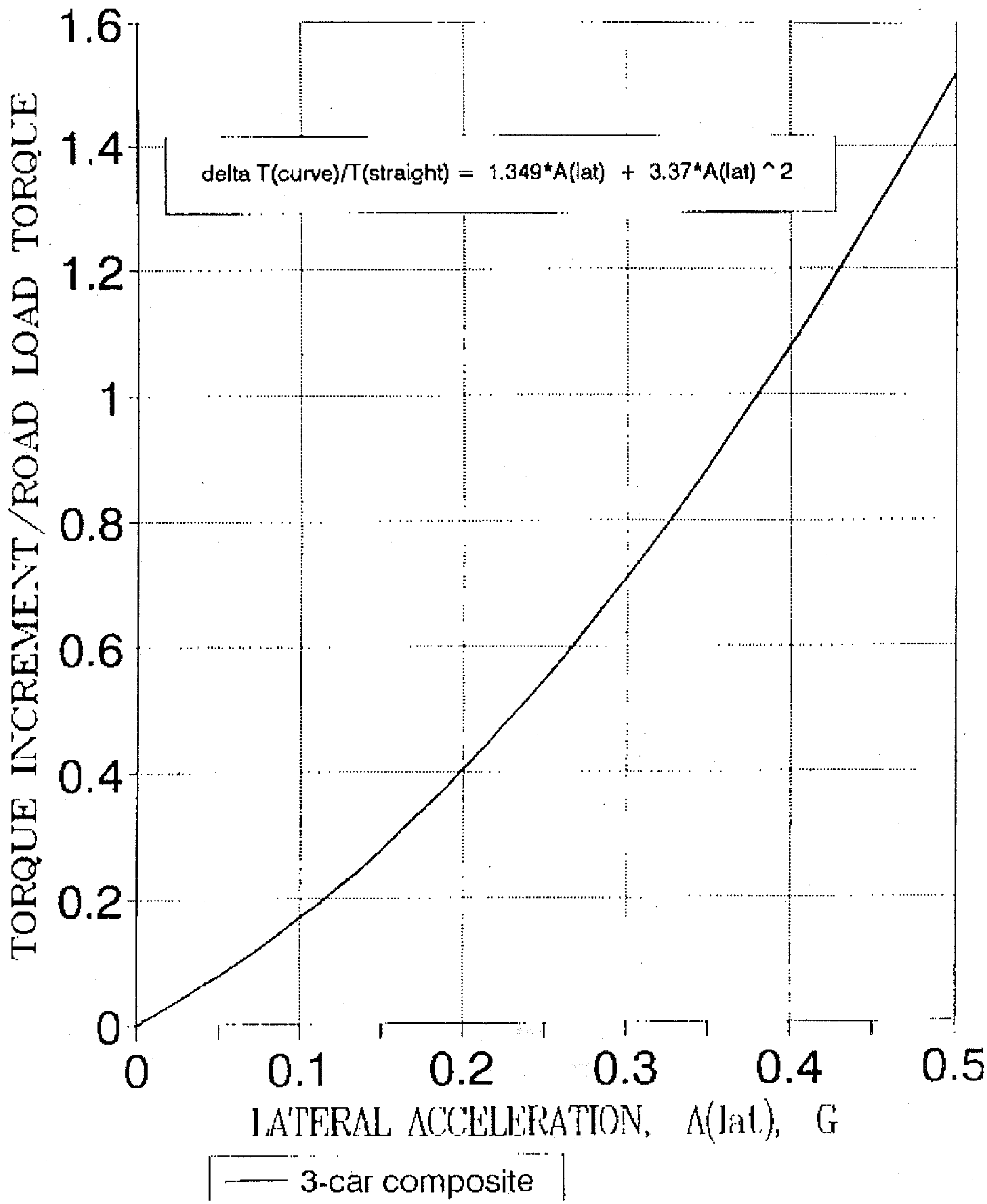
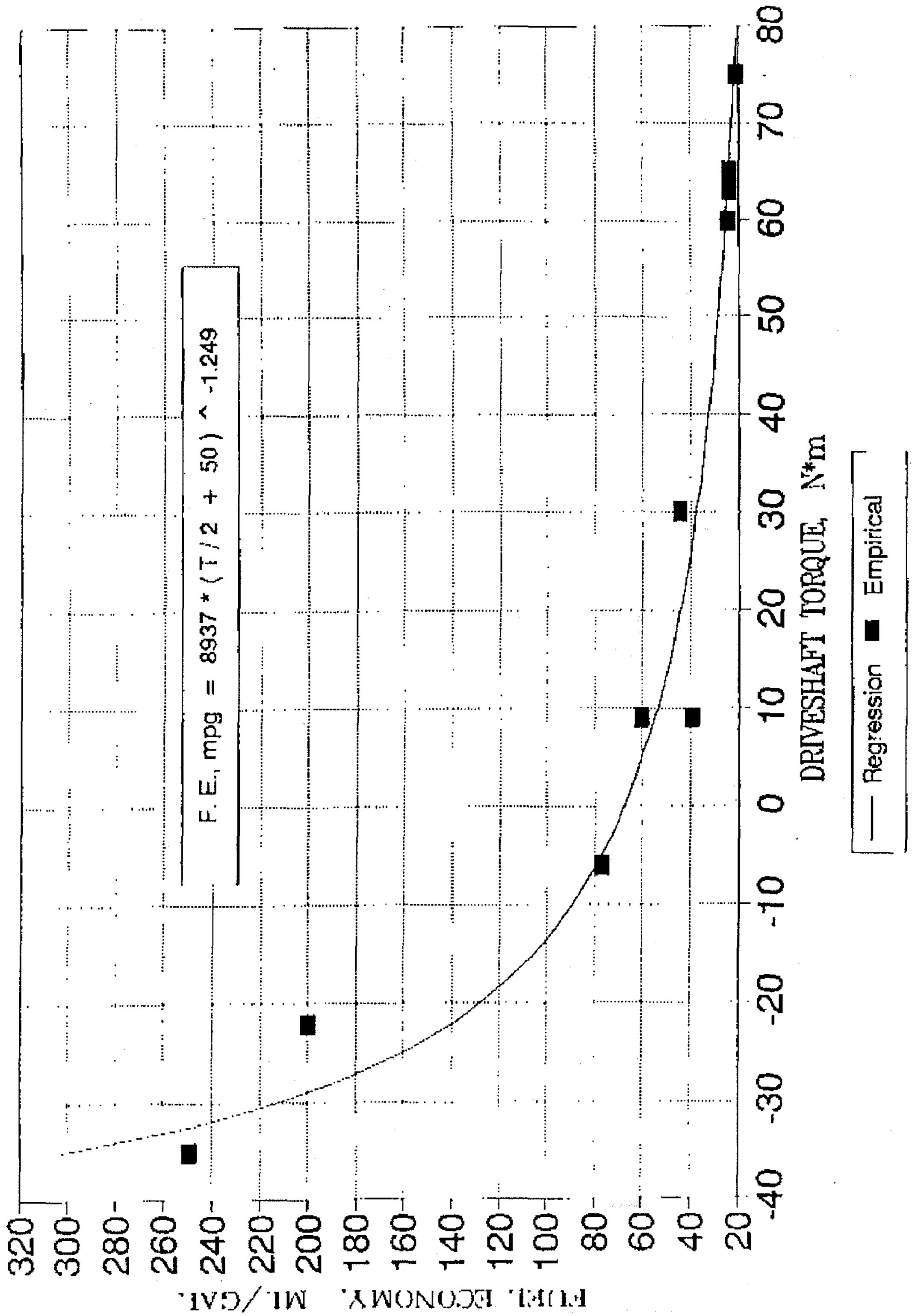


Figure 10. Composite normalized curve-torque increment - three cars.



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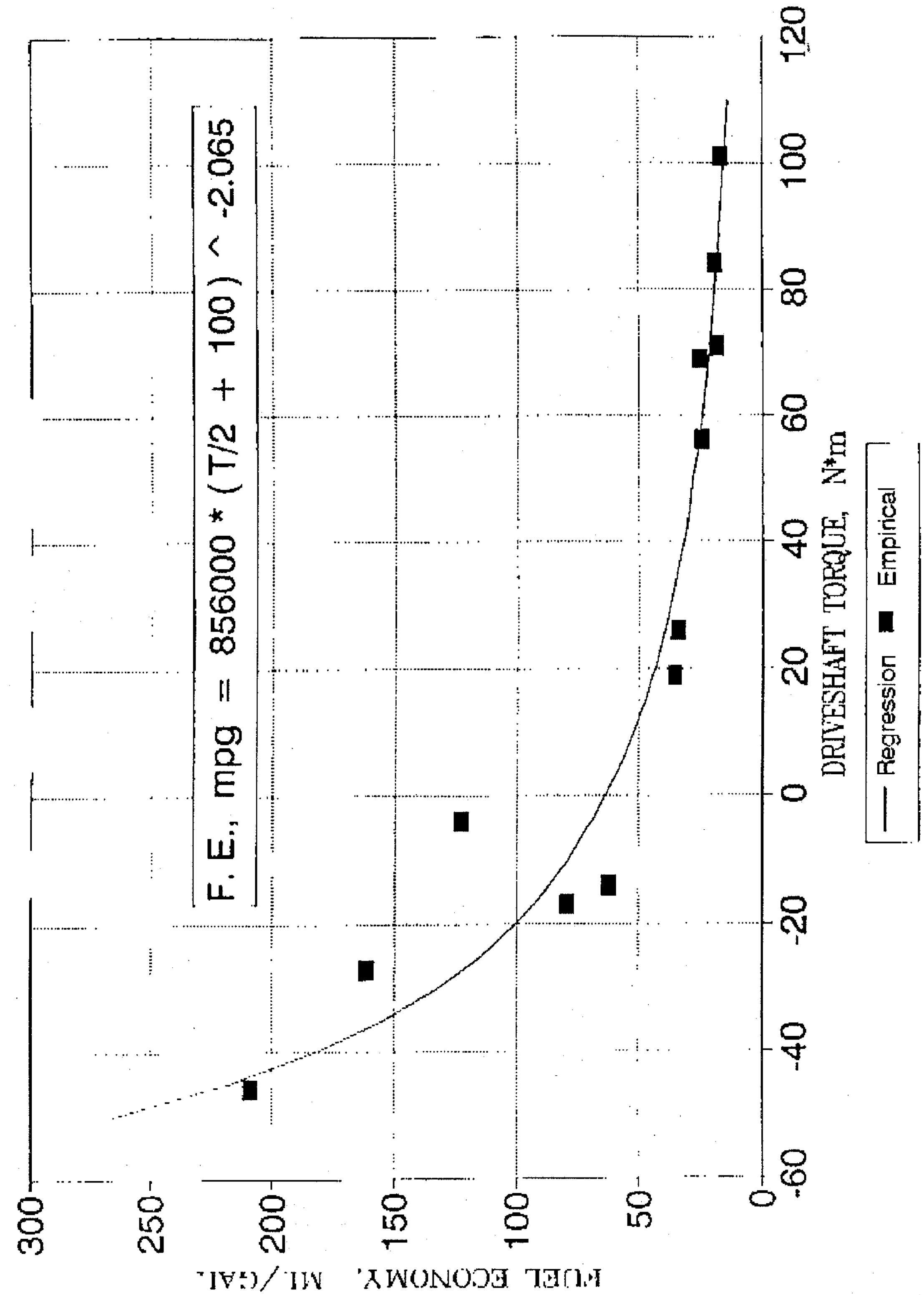
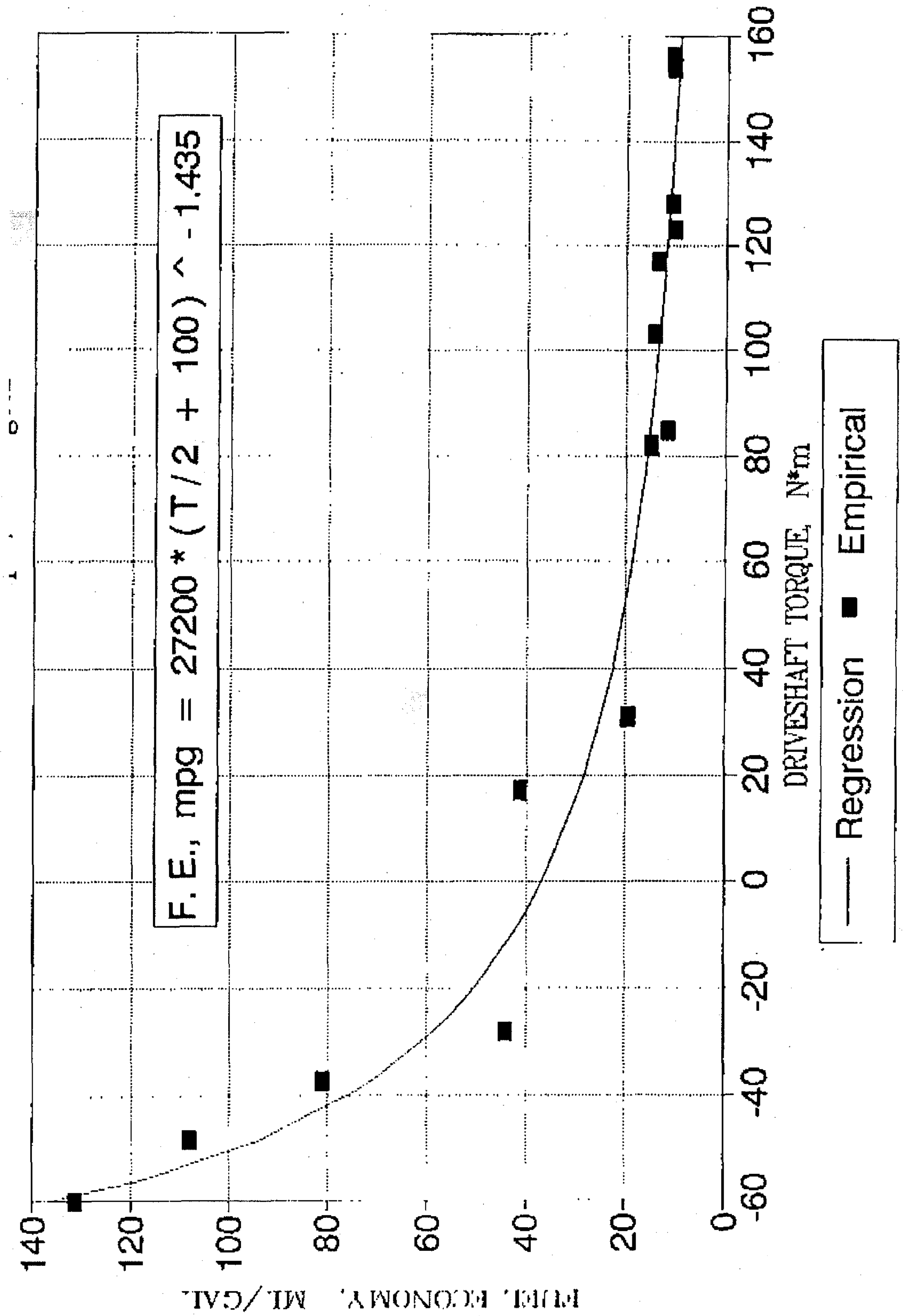
1 mi/gal =.42 km/L

Figure 11. Chevette fuel economy vs. driveshaft torque, fourth gear.



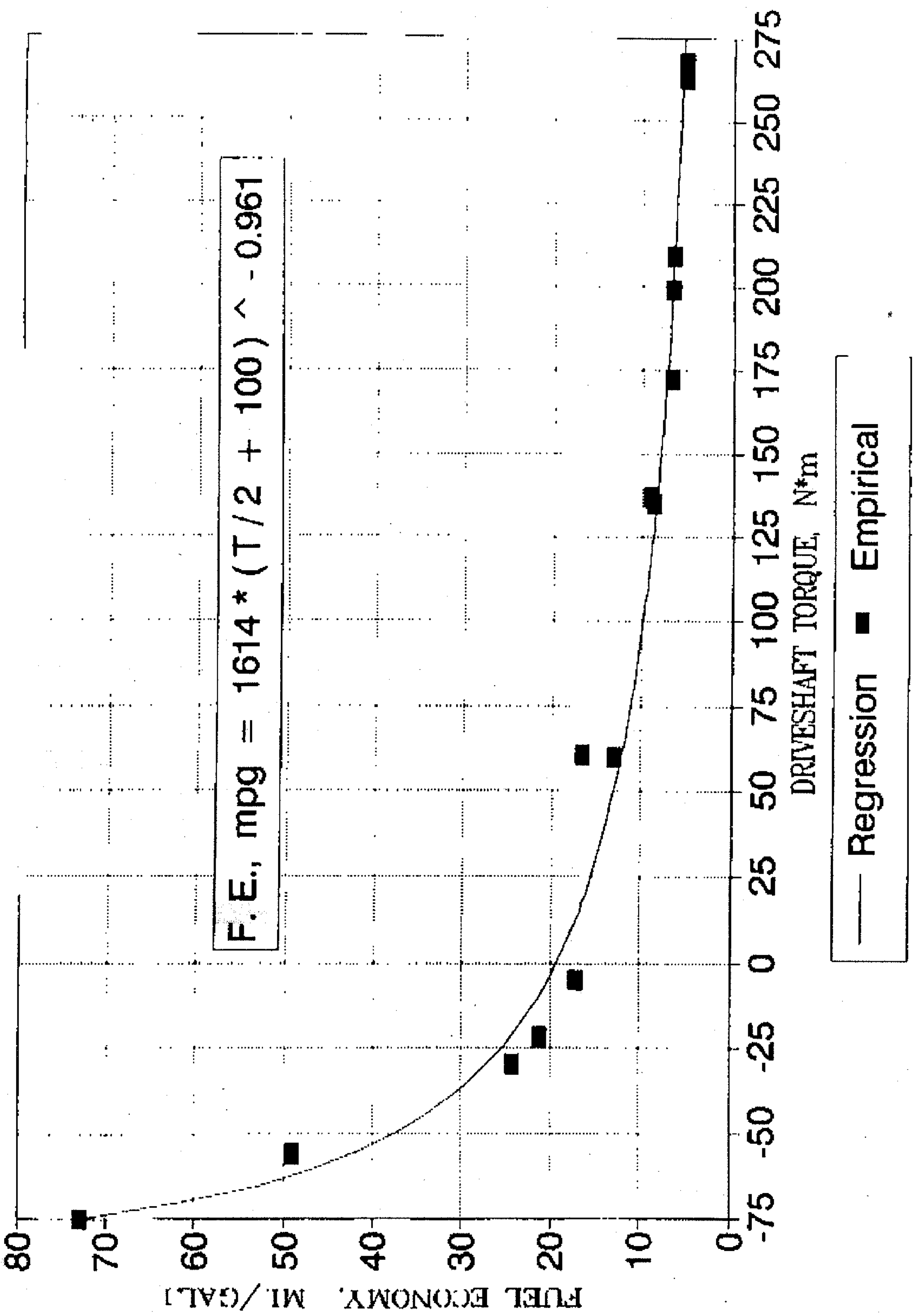
1 mi/gal =.42 km/L

Figure 12. Chevette fuel economy vs. driveshaft torque, third gear.



1 mi/gal =.42 km/L

Figure 13. Chevette fuel economy vs. driveshaft torque, second gear.



1 mi/gal =.42 km/L

Figure 14. Chevette fuel economy vs. driveshaft torque, first gear.

Table 6. Chevette fuel economy vs. driveshaft torque and gear.

(See figures 11 through 14)

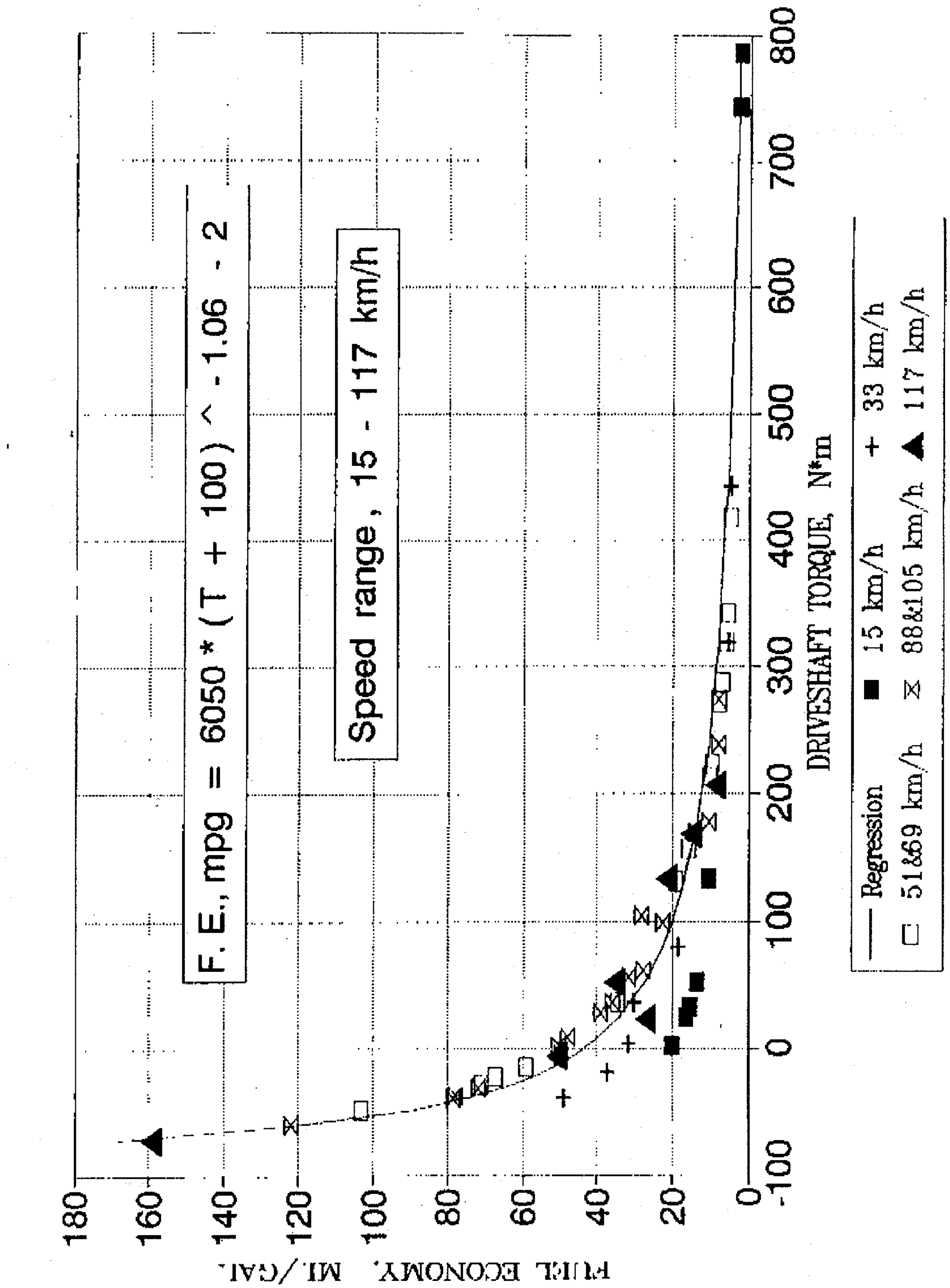
REGRESSION EQUATION:

FUEL ECONOMY, FE, MPG = a $(T/2 + b)^c$ (T in N.m)

DATA:

GEAR	SPEED RANGE km/h	COEFFICIENTS a	b	c	R^2
4th	100 - 125	8937	50	-1.249	0.955
3rd	60 - 105	856000	100	-2.065	0.925
2nd	20 - 70	27200	100	-1.435	0.962
1st	12 - 38	1614	100	-0.961	0.977

the Citation and Chevette that were found later). Consequently, a single regression equation was used to represent all of the experimental data, as is shown in figure 15; this equation, and the range of test speeds, are shown on the figure. This regression equation is similar in form to the fuel economy equations presented earlier, except that it was necessary to subtract a constant from the quantity obtained from the power equation. No coefficient of determination was calculated for this regression.



1 mi/gal =.42 km/L

Figure 15. Pontiac fuel economy vs. driveshaft torque, all speeds.

APPENDIX D
TEST FACILITIES AND PROCEDURES

This appendix describes the facilities used for road tests and the procedures utilized in vehicle testing and in analysis of the resulting data.

D.1 ROAD TEST FACILITIES

D.1.1 QUABBIN RESERVOIR

Initial pilot tests under this project were performed on roads within the Quabbin Reservoir of the (Massachusetts) Metropolitan District Commission (MDC). This reservoir has access roads that normally are open to the public but that can be closed by the MDC for any reason and without notice. A 4-mi (6.4-km) section of road was closed for the exclusive use of VNTSC for 5 days of testing.

The roads used in these tests are posted for public use at a maximum speed of 25 mi/h (40 km/h). One 2.5-mi (4.0-km) section was used for tests at 15, 30, 45, and 60 mi/h (24, 48, 72, and 96 km/h), while the other 1.35-mi (2.2-km) section could be run safely only at the lower three speeds because of sharper curves. The road was paved with bituminous asphalt concrete; grades ranged from level to 8 percent; and horizontal alignment included straight sections and curves with radii ranging from 240 ft (73 m) to 1900 ft (580 m), or curvatures from about 24 degrees to 3 degrees. Grade of the entire length of road was measured every 50 ft (15 m) by a calibrated DC servo accelerometer installed in a test support truck. The truck was stopped every 50 ft (15 m) to eliminate any influence of vehicle motion while the accelerometer signal was measured with a digital voltmeter.

D.1.2 SOUTH WEYMOUTH NAVAL AIR STATION

The South Weymouth Naval Air Station (SWNAS), South Weymouth, MA, has major runways of about 6000 and 7000 ft (1.8 and 2.1 km) in length, and is located within about 30 minutes' driving time from VNTSC. The shorter runway is extremely level over its entire length and is oriented approximately parallel to the prevailing good-weather wind direction. The longer runway intersects the other at nearly 90° near one end of the shorter and near midpoint of the longer; but the longer runway has an elevation change of about 10 ft (3 m) near one end. The 6000-ft (1.8-km) runway is about 150 ft (46 m) wide, while the other is about 200 ft (61 m) wide. For a time, the SWNAS suspended flight operations on Mondays, and since aircraft traffic was essentially nonexistent those days, VNTSC was allowed to conduct vehicle tests on the runways. Subsequently, air traffic increased and vehicle testing there was discontinued.

D.1.3 QUONSET POINT NAVAL AIR STATION

The Quonset Point Naval Air Station (QPNAS), Quonset Point, RI—now a civilian airport—had a main runway 8000 ft (2.4 km) long. The facility was located about 2 hours' driving time from VNTSC. Coast down tests were made on the runway, but the grade of the runway varied too greatly along its length to be suitable for this procedure. The airport also had a paved area well over 800 ft (244 m) x 1600 ft (488 m) that looked promising as a site for circle tests of up to 300-ft (91-m) radius; however, this area was used for parking of a large number of privately owned light aircraft and was unavailable for the intended circle tests.

D.1.4 BANGOR INTERNATIONAL AIRPORT

The Bangor International Airport, Bangor, ME, has one runway 11,438 ft (3.5 km) long; of this, the central 6200-ft (1.9-km) portion has a very uniform grade of about 0.58 percent, while the end segments of 2400 and 2800 ft (0.73 and 0.85 km) have different slopes. The airport also has a portland cement concrete-paved heavy-duty apron that is about 750 ft (229 m) wide with a length of about 1500 ft (457 m) available for general use. Arrangements were made with the airport management by the Maine Department of Transportation for the VNTSC to use both facilities for automotive testing. A portion of the heavy-duty apron was used to lay out concentric circles with radii of 100, 200, and 300 ft (30.5, 61, and 91.4 m); a transit survey of this area showed it was a plane with a slope of 0.79 percent (for drainage).

This facility was used for the major portion of vehicle testing under this project. The site is located approximately 6 hours' driving time from VNTSC.

D.2 VEHICLE TEST AND DATA ANALYSIS PROCEDURES

The principal vehicle tests performed under this project consisted of pilot tests on roads and on a chassis dyno (in the first year, to establish feasibility and validity of the project), of straight-road tests to determine vehicle road load vs. speed, of curved-road tests to characterize the influence of curves on propulsive demand, and of chassis dyno tests to map fuel economy and emissions vs. speed and tractive load. These tests involved four different automobiles. Supplemental supporting tests included fuel economy during accel/decel on a chassis dyno for one vehicle, increased drive train loss at higher tractive forces for one vehicle, and the influence of different drivers on road fuel economy. The principal tests and data reduction procedures are described in separate portions of this section.

D.2.1 PILOT TESTS ON ROAD

D.2.1.1 Test Procedure

The major objective of the pilot tests was to demonstrate that the vehicle propulsive demand imposed by principal highway geometrical features could be measured and could be related quantitatively to geometrical parameters. Further, it was to be determined whether the effects of such features were independent of the particular vehicle being operated on the road (except for such commonly known vehicle quantities such as weight, drive axle ratio, wheel radius, etc.). Driveshaft torque had been specified as the primary indicator of propulsive load, and the utility and sensitivity of this parameter were to be evaluated.

The pilot tests were performed on paved roads (see Section D.1.1) that offered grades from level to about 8 percent and horizontal alignments from straight to curves of 240-ft (73-m) radius. It was desired to measure the effects of grades and curves both individually and in combination; the test road provided several locations where curves and grades were superimposed and fewer sites where one geometrical feature existed in isolation. The experimental approach was to drive over the test road in both directions at each of four constant speeds while recording speed and torque continuously over the entire length; the resultant data were analyzed to determine the effects of selected road features by identifying positions along the road within the recorded data. Since the parameter of interest was driveshaft torque, not fuel economy (or consumption), the vehicle was operated with its automatic transmission fixed in whichever gear gave smoothest engine operation and best control of speed; consequently, the tests usually involved lower gears and higher engine speeds than would be used in normal driving (except at 60 mi/h where conditions were typical of highway driving). The use of lower gears was particularly necessary on downgrades at lower speeds, in order to develop sufficient engine braking to maintain the target speeds. Engine braking without use of the vehicle's service brakes was necessary to transfer the total braking torque from the engine through the instrumented driveshaft where it could be measured; any application of service brakes would generate a torque not measured by the driveshaft.

Target road speeds were 15, 30, 45, and 60 mi/h (24, 48, 72, and 96 km/h); on the steeper downgrades, it was not always possible to hold the 15 mi/h speed in low gear with closed throttle, and speeds sometimes reached 21 mi/h (34 km/h). The sampling speed of the data logger was varied to provide a data point every 4 to 9 ft (1 to 3 m) of travel at all speeds. It had been intended to use an after-market automobile speed control and fuel consumption "computer" device to maintain constant speed, but the device was found to respond much too slowly to sudden changes in grade; throttle control by the driver, while less precise than desired, provided the ability to anticipate grade changes and, overall, better stability of speed than did the speed control device. A total of 19 one-way runs of acceptable quality was made; these were divided approximately evenly among the four test speeds and both directions.

The road facilities available for the pilot tests were not well suited to determination of constant-speed, level-road driveshaft torque vs. speed, or to coastdown methods. Thus, the road load characteristic of the pilot test vehicle was not measured.

D.2.1.2 Data Analysis

All test data were printed in engineering units with cumulative distance tabulated for each scan. The chosen test segments were located first by cumulative distance from the start, then by inspection of torque level for the expected kind of change. Within an identified test segment, the data were scanned to select an interval, or group of consecutive data points, which exhibited essentially constant speed and torque. For each segment at each speed, data points were accumulated from additional runs until a stable mean value with reasonable standard deviation had been developed. A total of 120 such data points was chosen in this manner from the 19 runs.

For each speed, mean driveshaft torque was regressed against positive and negative grades; the data were well described by a straight-line relationship, as had been expected. The slope of the straight line, in units of (N.m of torque increase)/(percent grade), should be a measure of the difference between drive torque on a grade and drive torque on a level road at the same speed, and therefore should be constant and independent of speed. This was found to be true for the three highest speeds; the regression slopes of 23.75, 24.75, and 24.56 N.m/percent for 30, 45, and 60 mi/h had a mean of 24.4 N.m/percent and a standard deviation of 0.53 N.m/percent. This compared well with the calculated value of 24.1 N.m/percent for this vehicle determined from test weight, drive axle ratio, and tire rolling radius. At the lowest speed, 15 mi/h (24 km/h), the regression slope of 21.07 N.m/percent differed significantly from the consistent slopes at higher speeds; however, as noted earlier, the test runs at nominal 15 mi/h were not all at nearly constant speed, but deviated by as much as 40 percent from the target speed on steep downgrades—while the higher-speed runs each had been at reasonably stable speeds. Had the engine and transmission been able to develop higher braking torques sufficient to hold speed down to near 15 mi/h, the slope of this regression curve would have been steeper and therefore would have tended to approach the regression slopes observed at higher speeds.

Efforts made to identify a torque component associated with road curvature in the pilot tests were unsuccessful. Many of the sharper curves were combined also with moderate to steep grades where the grade torque component was large and dominant. Also, the shorter-radius curves necessarily were of short arc length and were traversed quickly; during the sudden vehicle maneuvers, the driver was unable to maintain a desired level of stability in either speed or torque. (This problem of maintaining stability during steering transients was to be encountered again, later, on plane curves as described below.) The order of magnitude of cornering losses and the dependence on speed and curvature were unknown, at the time. The net result was that it was not possible to observe any quantitative effect of curvature on driveshaft torque in those tests, and it was determined that a more sensitive and better-controlled test method for this relationship would have to be developed.

D.2.2 VEHICLE ROAD LOAD VS. SPEED

As mentioned elsewhere in this report, within this project the term "road load" is restricted to mean the relationship between driveshaft torque (or tractive force) and speed at different constant speeds on level, straight road in the absence of wind (i. e., of ambient air motion relative to the ground). With the torque-instrumented vehicles used in this project, and given the test roads available, much better determinations of road load were made by measuring torque at several constant speeds than by using coastdown procedures employed frequently by others for uninstrumented vehicles. Only the multiple- constant-speed method is described here.

D.2.2.1 Test Procedure

The test "road" for these measurements was the main runway at Bangor International Airport (see Section D.1.4). Test data were taken only on the constant-slope central portion; the end portions were used for acceleration to and stabilization of test speed, and for deceleration and exit from the runway. Road load tests usually were performed in late evening hours when aircraft traffic was relatively light (but still far from non-existent). The vehicle was warmed up to approximately normal condition prior to starting the tests by making a 25-mi (40-km) round trip on the nearby Interstate I-95 at 55 mi/h (88 km/h). All test operations on the runway and taxiways were under radio control of the airport control tower. Target speeds generally were 70, 55, 40, and 25 mi/h (113, 88, 64, and 40 km/h); speeds under 25 mi/h (40 km/h) were not used on the runway because of the inordinately long time the runway would be occupied. The sequence of test speeds usually was from high to low in order to have the drive train preconditioned as nearly as possible to representative operating temperature; the delays incurred before each test, both for data system preparation and for runway access clearance, inevitably resulted in some loss in drive train temperature as testing progressed.

Tests at a given target speed always were made in pairs of one run in each direction on the runway with as little time as possible lost between the two runs. This time lapse ranged from a minimum of about 2 minutes, to reset the data system, to as much as 10 minutes while holding for an aircraft to land or take off (aircraft almost always were given priority). If the delay between two runs at the same speed exceeded 15 minutes, the first run was disregarded and two subsequent runs were made. These constraints admittedly were not what would be desired, but the consistency of the resulting data seemed to indicate that the effects of the time restrictions were not serious.

A typical pair of test runs would be performed as follows: The test vehicle is stopped at the edge of a taxiway near the end of the runway; the engine is idling. The data logger is preset for the run, and a 30-s scan of all data channels is recorded to establish zero-offsets. Clearance to run is requested from the airport control tower by radio. When clearance is granted, the vehicle is driven onto the runway and accelerated to test speed. Speed is stabilized as the vehicle enters the 6200-ft-long (1900-m) test section. When well into the test section (approximately 500 ft, or 150 m) so as to be well past any grade transition zone, data recording is started and continued to near, but short of, the end of the test section. The recorder is turned off and the vehicle is accelerated to the far end of the runway, driven onto a taxiway and stopped at the edge thereof. The tower is advised the car is clear of the runway. Total test run length while collecting data was about 5000 ft + 500 ft (1.5 km + 0.15 km).

Necessary data review and identification procedures, which require 2 to 3 minutes, are followed; another 30-s zero-scan is made, and runway clearance is requested from the tower. If clearance is granted within a few minutes, a return run is made in the opposite direction at as near to the first run's speed (whether it was on target or off by several km/h) as possible, as described above. However, if a hold of more than a few minutes is required, another zero-scan will be made immediately before the return run. If the hold extends beyond a total stopped time of 15 minutes, a new zero-scan is taken, the return run is made but now is considered the first of a new pair (the initial, "old" run is disregarded), and a third run in the same direction as the first is made to complete the run pair.

D.2.2.2 Data Analysis

The test data were printed in engineering units both for test runs and for zero-scans; the printouts included means and standard deviations for both torque and speed. The end-of-run summaries were examined first, and the entire printouts were inspected for anomalies, data dropouts, etc. Standard deviations in speed for the entire test series for a given vehicle ranged from about 2.5 km/h down to less than 1.0 km/h; while standard deviations in torque, on a similar basis, ranged from under 30 N.m to under 10 N.m. The number of data points per individual test run increased from about 130 in early tests to about 450 in later tests. In general, no editing of the data was performed to reduce the range of speed within a run or to correct observed torques for the torque increments caused by gradual speed changes. The mean net torque values were regressed against a quadratic function of mean speeds.

Since the tests had been performed on a road with a slope of about 0.6 percent, initial correlations were made separately for the uphill and downhill directions; coefficients of determination (R^2) for these individual directions ranged from more than 0.999 to worst-case values of about 0.96. To obtain a regression equation for level road, one of two approaches was used: torque values at equal speeds were calculated from the regression equations, averaged at each speed, and regressed against speed; or, the raw uphill and downhill data points were regressed jointly along with level-road torque data at very low speeds from curve-torque tests (see below) where lateral accelerations were at or below 0.10 G. In all cases, the coefficient of determination for the level-road load curve was 0.960 or better.

A minor difficulty was encountered when regression of the Chevette data was performed. Road-load torque had been measured only down to speeds of 25 mi/h (40 km/h). With no low-speed data entered into the regression, the regression values of level-road torque decreased as speed dropped to about 20 km/h, then increased again as speed decreased to zero. Clearly, this is an anomalous artifact of the regression procedure when extrapolating to outside of the region of data entered. Similar behavior was observed in regressions for the Pontiac and Citation when runway data for speeds only in excess of 40 km/h were used; but, when additional test data at lower speeds were added to the regression, monotonic decreases in regression torques were obtained down to zero speed (or, for the Citation, to less than 5 km/h). The road-load curve for the Chevette was observed to lie between those of the Pontiac and Citation throughout the runway speed range of 40 to 120 km/h; therefore, it was considered reasonable and valid to force the Chevette curve to remain between the other two down to lower speeds, also; this was done by finding an equation that lay in that region and blended approximately with the raw-data curve in the vicinity of 40 km/h. Consequently, either of two road-load equations will be used for the Chevette, depending on whether speed is above or below 43 km/h (26.7 mi/h).

It had been expected that refinement of the raw torque data, to correct the torque values for the torque increments associated with observed gradual changes in speed, would be necessary. This would have involved a significant effort in software development at a time when resources were needed in other areas. The excellent correlations of the raw data means indicated that such refinement would not have improved the quality of the data to any substantial degree and would not have been cost-effective.

D.2.3 HORIZONTAL CURVATURE EFFECT

D.2.3.1 Test Procedure

As indicated earlier, the inability to discern effects of horizontal curvature in the pilot tests demonstrated that a more sensitive test procedure with minimum operating transients was required for investigation of this phenomenon. In particular, the tests must be made in the absence of significant grade.

The first refinement was to lay out curves of different radii on an airport at the intersection of two perpendicular runways. One runway was 150 ft (46 m) wide, the other was 200 ft (61 m) wide. A complete circle with a radius of 100 ft (30.5 m) was defined; and arcs of 200-ft and 300-ft (61-m and 91.5-m) radius were laid out with contiguous tangent approaches along the 150-ft-wide runway. Multiple continuous laps were driven on the 100-ft-radius circle and produced stable data up to a lateral acceleration of 0.6 G (lateral acceleration = $(\text{speed})^2$ divided by curve radius; G = gravitation acceleration constant = 32.2 ft/s^2 = 9.81 m/s^2). The results of the tangent-arc tests were much more limited and inconclusive; it was just too difficult to maintain either constant speed or constant torque (throttle position) during the sudden transition from straight-line motion to the substantial lateral accelerations on the arcs.

Fortunately, the complete-circle tests guided interpretation of the limited longer-arc data; the results tended to support the initial thesis that lateral acceleration might be a meaningful independent variable that would incorporate the interacting parameters of speed and radius. The conclusion was that only tests on complete circles of different radii would afford data of adequate quality.

A maximum circle radius of at least 300 ft (91.5 m) was desired to permit speeds approaching highway levels at lateral accelerations within 0.5 G (a speed of 76.2 km/h, or 47.4 mi/h). A 600-ft-diameter circle, plus a 50-ft-wide safety margin outside of the circle, requires a paved area of 11.25 acres in a 700-ft-square format. (A 400-ft-radius circle with similar safety margin would require 18.6 acres of smooth, continuous pavement in square format.) Shopping malls have parking lots exceeding this area, but they are laid out in an unsatisfactory rectangular format, and usually are obstructed with curbs and light poles. Established auto test facilities with such skid pads were located far from VNTSC.

Fortunately, the Bangor, ME, International Airport has a heavy-duty apron (aircraft parking and service area) that is greater than 700 ft (213 m) wide and much longer, and the management was willing to allow testing of cars there. Concentric circles of 100-, 200-, and 300-ft radii were marked off with 12-in (0.4-m) lengths of 4-in-wide (10-cm) yellow traffic-lane marker tape spaced about 15 ft (4.5 m) apart. (For the guidance of others who may have cause to lay out similar paths, lengths of 18 to 24 in (0.4 to 0.6 m) would improve visibility from the driver's seat.) A transit was used to determine that the surface, pitched uniformly for drainage, had a slope of 0.79 percent. A reference line along a radius at the high point of the circle was laid out with road tape across all three circles to define the start/finish point for all laps.

Target speeds were calculated for each circle to give lateral accelerations of 0.1, 0.2, 0.3, 0.4 and 0.5 G. Precise values of lateral acceleration were unnecessary, however; it was more desirable to maintain the greatest degree of stability in speeds that were in the area of the target values. The actual values of lateral acceleration for whatever test speeds were used could be calculated later. Accordingly, the drivers were instructed to approximate the target speeds as closely as reasonable, but to emphasize stability at whatever final speeds seemed most manageable. The driver would first establish his path straddling the designated taped circle; then he would adjust his speed to a level which he felt he could maintain. In this way, he usually drove two or three practice laps of the circle, for each test, before recording any data.

When comfortable and as stable as feasible, he would start data recording just as he crossed the start/finish line. He would maintain stable speed and track for six laps, and stop the data logger just as he crossed the finish line. (Sometimes the driver would lose count and make five or seven laps, but this was detectable easily in the final data.) The higher-acceleration tests imposed severe wear, and presumably greater heating, on the outside tires on the circle; therefore, alternate directions of rotation were used for odd- and even-numbered tests (i.e., 0.1, 0.3, and 0.5 vs. 0.2 and 0.4 G). To provide the driver with lateral support from the car door in the 0.5-G turns, these were driven in the clockwise direction (as viewed from above). Only one series of laps was run at each speed and radius in order to minimize testing time; it was reasoned that, since several laps were being made at each condition, a bad lap could be deleted without invalidating the rest of the data for that test, if necessary.

In retrospect, it would have been useful to have run at least one acceleration level on each circle in both directions; this would have provided data in both directions that were directly comparable, and occasionally there were questions that could have been handled better if such data were available. The sequence of runs at five lateral acceleration levels on three radii was approximately randomized to minimize experimental bias effects.

The total number of scans (of all data channels for each scan) per lap averaged about 140. The number varied, depending on the scanning frequency chosen, the speed, and the curve radius; the minimum number of scans per lap in any test was about 60.

D.2.3.2 Data Analysis

Test data were printed out and inspected as described in Section D.2.2.2. Various dependent relationships of the test data means were calculated to assess the integrity and consistency of the data; these included the ratio of driveshaft RPM/road speed, the mean driven-circle radius based on distance traveled and number of laps run, etc. The mean speed for each test run, and the nominal circle radius, were used to calculate mean lateral acceleration. For each mean speed, the previously determined road-load equation (constant speed, straight and level road) was used to calculate the normal road-load torque; this torque value was subtracted from the mean experimental torque to determine the mean absolute torque increment imposed by the circular path. The absolute torque increment also was divided by the road-load torque at that speed to obtain the normalized, dimensionless relative torque increment associated with the curve.

Both the absolute and the normalized torque increments for each circle radius were regressed against a quadratic function of lateral acceleration. The results for the first vehicle tested (Chevette) provided very strong support for the hypothesis that lateral acceleration indeed was an excellent independent variable, and that the absolute and normalized torque increments were valid dependent parameters, for characterization of this effect of horizontal curvature. The absolute torque increment of the Chevette exhibited a coefficient of determination (R^2) of more than 0.98, while the normalized increment's R^2 was 0.84. Subsequent vehicles produced results of the same or even higher correlation, in nearly all cases; the Pontiac yielded results of this level when either radial-ply or bias-ply tire data were considered separately, but the coefficient of determination fell to 0.73 when the data for normalized torque increment for both types of tires were combined. In all but the bias-ply tire case with the Pontiac, absolute torque increments at equivalent lateral accelerations but on different radii (hence, at different speeds) tended to group very tightly; the data tended to disperse more when the corresponding normalized torque increments were considered.

It was desired that a generalized curve-effect relationship be developed that could be applied also to other vehicles that had not been curve-tested. Accordingly, the regression curves for all three cars were plotted on the same graph (Appendix C, Section C.2.2, figure 10). The differences between the curves were observed to be considerably less than the dispersion of the experimental data, and it was considered reasonable, at least for the purposes of this project, to calculate a composite regression of all three vehicles. Data for this composite regression were obtained by using the regression equations for normalized torque increment of each of the vehicles to calculate normalized torque increments at the same values of lateral acceleration. When these three sets of data were regressed together against lateral acceleration, the composite equation had a coefficient of determination, R^2, of 0.986. While this is not descriptive of the scatter of the actual raw data points, it is an indication of the degree of agreement between the individual regression equations for the three cars.

In retrospect, perhaps the above data should have been treated in a somewhat different manner. This would be especially true if one wanted the most quantitative analysis of the phenomenon of curve effect on individual vehicles. (As noted in the preceding paragraph, the emphasis in this project was to try to find a reasonably valid approximation that could be applied to vehicles in general.) Subsequent to testing of the two RWD cars (Chevette and Pontiac) but before testing of the FWD Citation, a theoretical analysis of the problem of "cornering drag" on plane curves was published by Segel and Lu (3). This paper chose the same quantities as used above, viz., lateral acceleration and normalized torque increment, for independent and dependent variables, respectively, and predicted similar quadratic behavior. (The paper expressed normalized torque increment as percent increase in drag, where "drag" is the total drag on straight, level road at the same constant speed; the percent increase in drag is dimensionless, as is normalized torque increment, and the former equals 100 times the latter.)

The analysis, in the absence of empirical data (which Segel and Lu found almost non-existent), predicted that "the percent increase in drag depends on the radius of the turn as well as on the lateral acceleration level." A review of the raw data from this project shows just such a dependence, which was very small at lateral accelerations typical of normal highway driving but became larger at higher levels. A more discriminating evaluation of the present data, in light of the referenced paper, would produce a set of equations rather than one single, general-purpose function. The choice of greater rigor and complexity vs. simplicity, economy in data reduction and facility of use thus resides with the user and should be governed by the needs and intended application.

If one prefers the rigorous approach, one needs—in addition to speed and radius of curvature—also the "cornering compliance" coefficient for the tires. This factor will vary with the degree of wear of the tires, and with the inflation pressure. Further, the tires on a car a few years old likely will be of a different make than any of the possibly several types of tires supplied by the manufacturer on a specific model when new. For the level of approximation required by the purposes of this project, and to enhance the ease of use of this procedure to the point where a highway designer may be persuaded to use this tool at all, a single, cruder relationship appeared preferable and adequate.

D.2.4 CHASSIS DYNAMOMETER TESTS

The main purpose of chassis dyno testing in this project was to map the fuel economy (and, early in the project, exhaust emissions) of the test vehicles over the normal operating range of the vehicle. This mapping was done by operating the vehicle at each of a number of steady-state speed and torque test points. Fuel consumption was measured (and emissions, if applicable). Tests were made also at idle; the transmission, if automatic, was in Drive position. All tests were made with the vehicle, drive train, and tires fully warmed up, and in a test cell where temperature was controlled to 70 °F ± 5 °F (21 °C ± 3 °C); no control of barometric pressure or relative humidity was provided (but, if emissions tests were made, these results were adjusted for ambient conditions).

The sequence of speed and torque levels during each test day was randomized to minimize any systematic bias; however, the randomization was controlled to preclude several consecutive tests occurring at combinations of speed and load that would produce excessive heating or cooling of the engine and drive train. For each vehicle, a single combination of speed and load representative of normal highway operation was selected as a "check point"; this check point was repeated at least once, and usually twice, per day to verify that vehicle and test system condition did not change significantly during the test sequence.

D.2.4.1 Test Point Determination

Selection of target values of speed and torque for test points was conducted in the following manner. An X-Y recorder was connected to the analog signal outputs of the chassis dyno for speed and torque. A simple RC filter with a time constant of about 2 s was connected in series with the torque signal to minimize torque fluctuations entering the chart recorder. The recorder scales were adjusted to span 0 to 120 km/h on the speed (X) axis and maximum negative to maximum positive capability of the vehicle on the torque (Y) axis. Calibration marks at known speed and torque levels were made on both axes. The test vehicle was warmed up and the dyno placed in adjustable constant-speed mode.

If the car had an automatic transmission, the dyno was started at 15 km/h (9 mi/h). The transmission was shifted into Drive, the recorder started, and the throttle gradually increased to wide-open-throttle (WOT) position and held there. This produced maximum or near-maximum torque for this vehicle. The dyno speed control was slowly and smoothly increased until a speed of 120 km/h (75 mi/h) was reached; the transmission was allowed to shift normally, always at WOT. Dyno speed then was decreased smoothly back to 15 km/h (9 mi/h), still with the throttle wide open, to define the points at which the transmission downshifted. The throttle then was released to full closed-throttle (CT) position, and the above sequence was repeated to determine the CT, engine-braking-torque curve. The chart recorder was stopped, and the product was a graph of the dyno torques corresponding to WOT and CT operation over the stated speed range.

If the vehicle had a manual transmission, the above procedure was modified slightly. For each gear, the dyno speed that would allow an initial engine speed of 800 RPM was calculated; also, the maximum speed for that gear, at engine redline, was determined. For each gear, the test would be started by setting the dyno to the calculated minimum speed, and the transmission would be engaged. The vehicle operator then increased throttle to WOT, the recorder was started, and dyno speed was dialed up to near redline. At that point, the operator released the throttle to closed position and dyno speed was smoothly dialed back to minimum speed for that gear. This process was repeated for each forward gear.

Either of these methods produced an experimentally determined maximum performance envelope for the vehicle at both WOT (wide open throttle - maximum positive torque) and CT (closed throttle - maximum negative, or engine-braking, torque), vs. road speed. This performance envelope provided the basis for selecting test speeds and torque levels. For each gear, either three or four speeds were selected, depending on the shape of the torque curve: one speed each near upper and lower limits, and one or two to define the peak and curvature. In general, the spacings of speeds were non-uniform.

At each speed, for every gear, five torque levels were selected on the following basis. Operation at WOT usually produced unsatisfactory data because of excessive torque fluctuations; also, heavy knocking frequently was encountered and could have damaged the engine during sustained test durations. Since engine efficiency (e.g., in terms of brake-specific fuel consumption, BSFC) changes relatively slowly with torque at medium to high torques, but very rapidly at very low torques, test points were planned to emphasize the low-torque region. At each speed, the total torque range, WOT minus CT (the latter usually was a negative value) torque, was calculated. The test torque levels were established as decrements of this torque range below WOT: WOT minus 15, 37, 60, and 85 percent of torque range, and at CT. Thus, at medium to high speeds, at least two test points were at negative (motoring) torque levels. This process resulted in selection of 65 to 80 potential test points for a given vehicle.

D.2.4.2 Performance Mapping

Before each test, the dyno was stopped and zero-scans of about 30 s duration were recorded on both the lab (dyno) data acquisition system and on the data logger used in the vehicle, to determine zero-offsets. The dyno, operating in constant-speed mode, was set to the speed specified for the test. The vehicle's transmission was set to the desired gear; if automatic, usually Drive position was used, unless "hunting" between two ranges occurred or a particular gear had been specified for some special reason. The vehicle's throttle pedal was controlled from the dyno console by a remote-control servo actuator. The throttle was adjusted to develop the approximate value of specified torque and the system was allowed to run for 1 to 3 minutes to stabilize; if throttle readjustment was required, the stabilization period would be extended. However, when operating at near maximum engine speed or maximum torque, the stabilization period was kept as short as possible. If emissions were being monitored, the appropriate sample bags were prepared.

At the start of the actual test period, the vehicle's data logger and the central data acquisition system for the dyno lab were started. The vehicle data logger recorded driveshaft torque and speed, and also the road speed indicated by a fifth wheel running on the dyno roll beside the vehicle drive wheel; sampling frequency was selected to accumulate an adequate number of data points without consuming excessive amounts of recording tape. The lab data acquisition system recorded dyno torque and speed at fixed 1-s intervals.

The usual test duration was 400 s if emissions were being monitored; this allowed accumulation of a "comfortable" quantity of exhaust gas for analysis. However, at heavy throttle conditions where high exhaust temperatures threatened to overheat the sampling system's main blower, the test period was shortened to 250 s. If only fuel consumption was being measured with an in-line fuel flowmeter, the test duration usually was determined by observing the fuel counter; a minimum of 200 fuel counts (at approximately 1 ml/count) was desired to provide a resolution of 0.5 percent in fuel quantity. In this mode, test duration could be less than 1 minute for high-speed, high-torque conditions; or it could extend to about 10 minutes for closed-throttle, negative-torque situations, and during idling tests.

When the test was completed, the data systems and dyno were stopped and zero-scans were repeated. Vehicle operation was returned to about 50 mi/h (80 km/h) at typical constant-speed, road-load torque to keep the engine and drive train at operating temperatures. The data summaries provided by both systems were scanned briefly to see that the mean test values were sufficiently close to the specified conditions and that no serious anomalies had occurred. The next test run identification number was entered into the vehicle data logger and the next test condition was established.

If emissions were being monitored, only three test runs could be made in a series before all sample bags were filled. Test operations then had to be halted for at least 20 minutes while the exhaust samples were analyzed, the bags emptied, purged, and emptied again. However, if only fuel consumption was being monitored, test runs could be continued without such interruption.

During these chassis dyno tests, the vehicle data logger was powered from a lab 120 VAC power line rather than from the DC/AC inverter in the vehicle. Thus, the inverter did not impose any load on the engine (and affect fuel consumption) during the performance mapping. Cooling air was discharged against the front of the vehicle from a duct 3.5 ft high x 7 ft wide (1.1 x 2.1 m) resting on the floor and ending within 1.5 to 3 ft (0.5 to 1 m) of the front of the vehicle. Air discharge velocity was slaved to the road speed of the vehicle's drive wheels. This provided engine operating pressure and temperature, and drive train/exhaust system temperature, conditions approximating those that would occur during similar on-road driving.

Dyno rotational inertia (either simulated electrically or augmented by inertia flywheels) usually was set to correspond to vehicle test weight, although this should not be particularly important in constant-speed operation. However, with one vehicle (the Chevette), torque oscillations sometimes developed. These seemed to involve torsional vibrations between engine and dyno inertias coupled by a long, springy, two-piece driveshaft; also between body mass and axles coupled by suspension springs (pitching of the vehicle body imparted an angular rotation disturbance to the drive axle); and between dyno stator (a substantial rotational inertia), the torque load cell spring member, and the electrical time constant of the dyno control system. These oscillations were reduced to an acceptable level by increasing the flywheel inertia coupled to the dyno so that total rotational inertia was equivalent to a vehicle mass of about 3200 kg, almost three times the 1134-kg test weight of the car. This increased inertia was used only for constant- speed tests.

In the early tests (Dodge Dart and Pontiac), tests were run at all of the test points that had been planned; this involved approximately 65 to 70 points per vehicle, plus replications of the single check point. Time and resources did not allow replications of other test points unless a result clearly was grossly aberrant; even here, if the apparently bad point fell in a torque/speed region where sufficient other data had been taken and the bad point was redundant, it simply was excluded from subsequent analyses. During final data compilation, it was observed that many test points had produced data that were multiply redundant in the low- to mid-torque region.

Consequently, in tests of the last two vehicles (Chevette and Citation), a "test optimization" strategy was adopted to reduce redundancy and to minimize test costs. The initial tests for each vehicle were conducted mainly at the maximum and minimum speeds and maximum-positive and maximum-negative torques in low and top gears. These were followed by a modest number of tests at intermediate speeds, torques, and gears. As data accumulated, certain of the analysis computations described in the following section were performed and the values of fuel economy were plotted against torque. When sufficient data had been accumulated to define adequately the fuel economy curve and to yield any other data of particular interest, testing was discontinued before completing the entire sequence of planned points.

D.2.4.3 Data Analysis

The vehicle data logger results were printed in engineering units and inspected as described in Section D.2.2.2. The lab data acquisition system printouts were evaluated similarly. Tabulations were made of mean values and standard deviations of actual torque and speed, fuel used, test duration, and distance "traveled" during the run. Net driveshaft and dyno torques were calculated. For the Pontiac, driveshaft torque values were converted to equivalent torques as would be seen by the dyno if no losses occurred in transfer through the differential and tires to the dyno rolls. These theoretical output torques at the dyno were subtracted from the actual dyno torques (the latter always were the lesser quantity) to obtain the values of torque loss, or dissipation, between driveshaft and chassis rolls. (Note: The decision to evaluate loss of torque, rather than of power, was made when it was observed that the loss of torque appeared to be dependent principally on the magnitude of torque transmitted—or of tractive force produced—and to be essentially independent of road speed.) If emissions had been measured, pollutant rates were expressed in units of g/mile. Fuel economy in MPG, and time rate of fuel flow in gallons/hour (GPH) were calculated. The ratios of driveshaft RPM/road speed in km/h were determined.

For the first vehicle (Dodge Dart), test results were plotted against driveshaft torque to assess what kind of relationship and what degree of complexity were involved. It had been expected that a kind of three- dimensional topographic map involving performance parameter (fuel economy or pollutant), torque, speed and/or gear, would be encountered. Instead, especially for fuel economy, to a very good first approximation, the result was a monotonic function of driveshaft torque; functions that appeared suitable included a second-order polynomial, a power equation, and a hyperbola.

Data from the second vehicle (Pontiac) showed similar behavior; apparently because this car had been tested at negative torques as well as positive ones, fuel economy seemed to be described best by a power equation. For both vehicles, the scatter of the data in the mid-torque region was greater than would be desired for, say, competitive sales comparisons; but the scatter was considered acceptable for the overall highway design evaluation purposes of this project. Agreement of regression curves with data points at the very low (or negative) and very high torque regions was good, and the mid-torque region was reasonably well-represented by the curve. (The pilot tests of the Dodge Dart had been completed prior to refinement of the dyno mapping procedure and principal development of the concept of highway torque demand/fuel consumption computation; consequently, mapping of this vehicle at negative torques was not as extensive as was done for later vehicles.) Both of the first two vehicles were equipped with automatic transmissions and torque convertors; the extent to which these variable-drive-ratio systems contributed to the approximately monotonic, single-independent-variable relationship between fuel economy and torque was unclear. A number of other vehicle characteristics, especially those related to carburetion, spark timing, and vehicle weight/engine displacement ratio, also could exert significant effects.

The last two vehicles mapped (Citation and Chevette) were equipped with manual four-speed transmissions. As for the first two cars, at higher torque levels the regression curves of fuel economy vs. torque tended to merge into a common curve. However, in the low- to mid-torque region representative of constant-speed driving and of engine braking, the fuel economy-torque relationship clearly was dependent on which gear was engaged. For any single gear, these parameters correlated as well as had been observed for the first two cars.

The number of test points at similar values of speed and torque but in different gears was substantially smaller for these last two cars than for the first two; this resulted from the test optimization strategy that had been developed to reduce the total number of test points. One less desirable consequence was that, in general, the number of data points in any one isolated sub-set (e.g., fuel economy at different torque levels at one speed in one gear) was insufficient to support detailed quantitative analysis. However, when the data were evaluated over the full torque range for a given gear, information contained in data points well removed from the region contributed guidance and stability to interpretation of points within the particular region.

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