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.

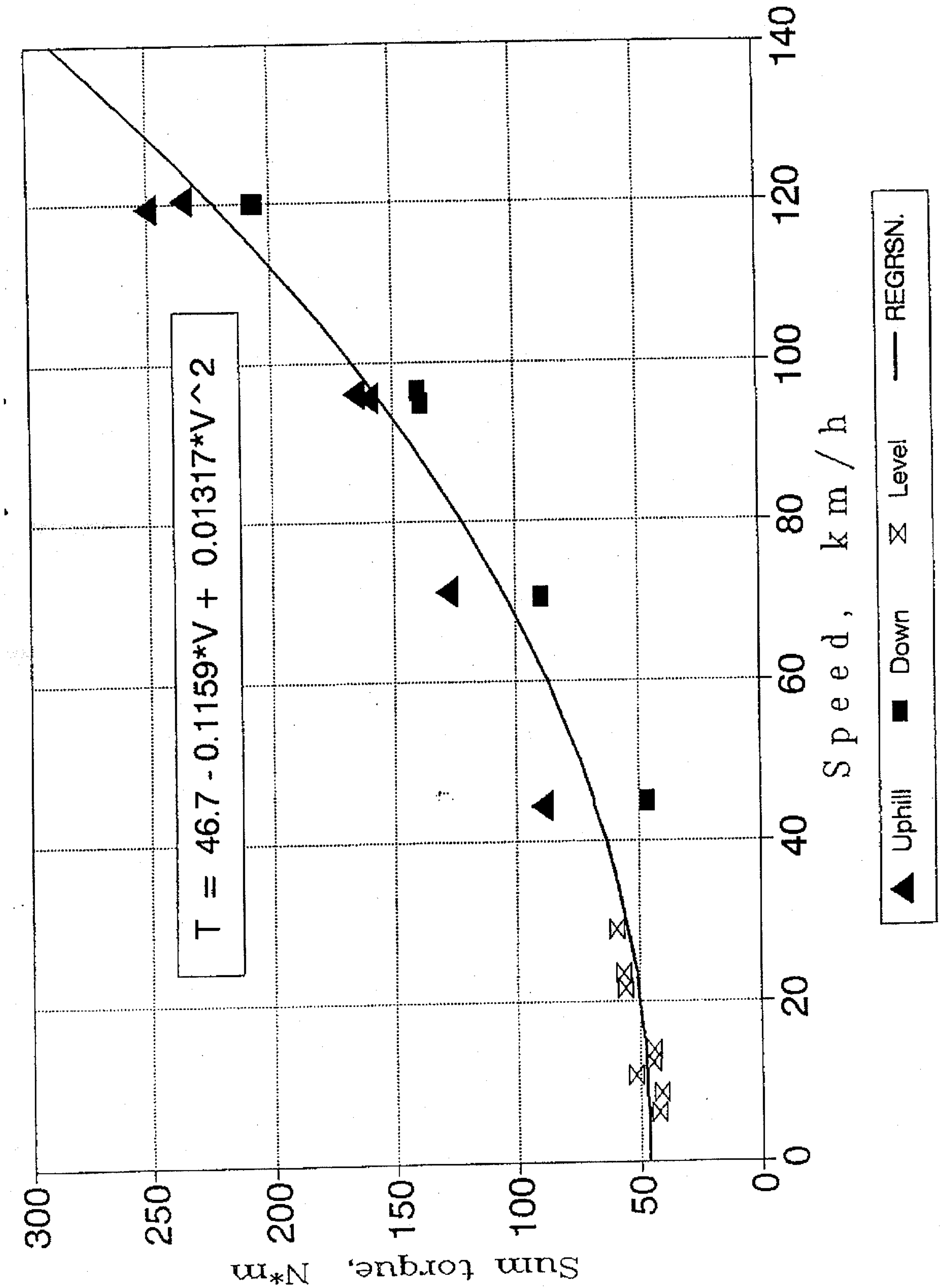


Figure 3. Citation road load torque vs. speed.

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	COEFFICIENTS			
	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.

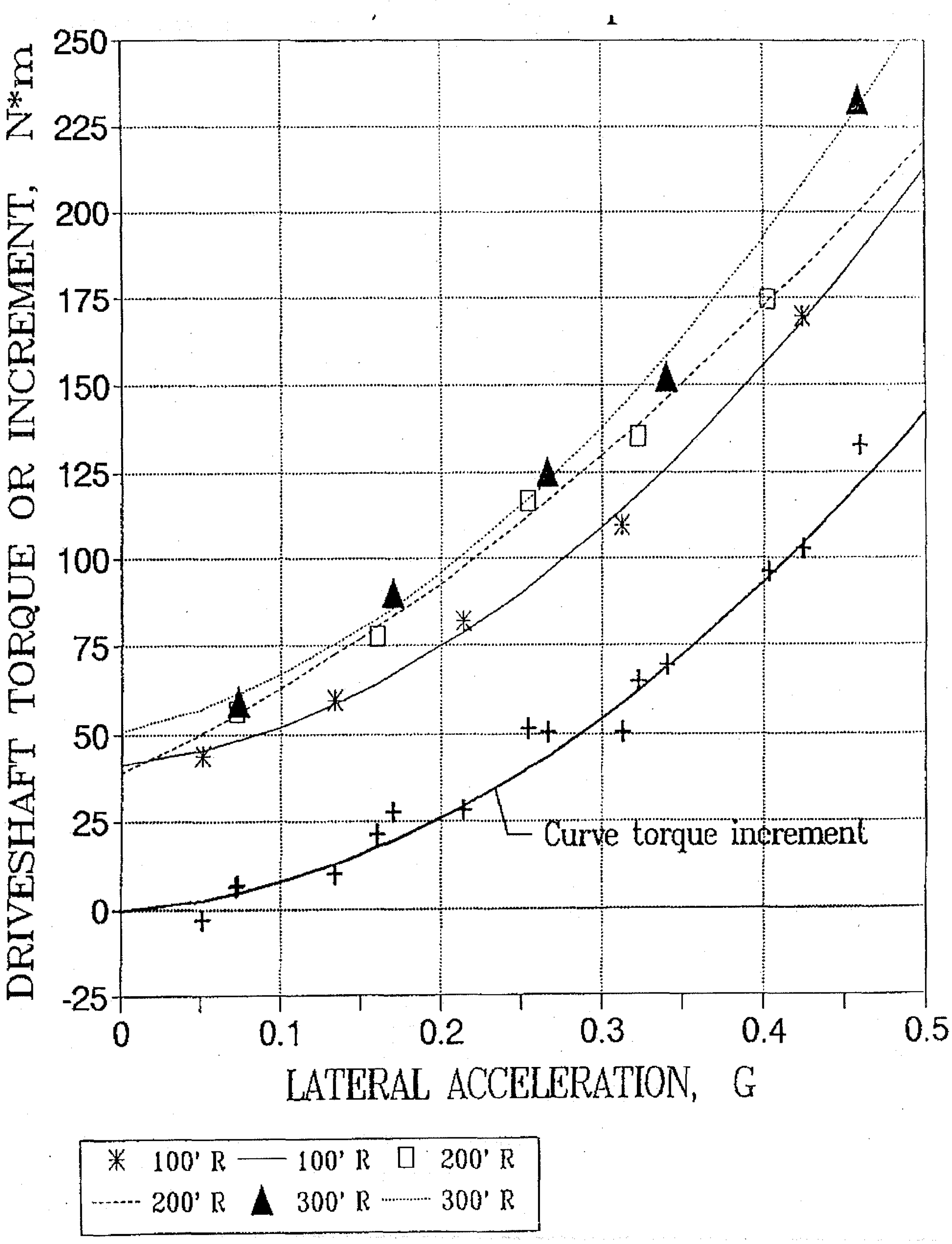


Figure 4. Total torque and torque increment on plane curves.

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 =

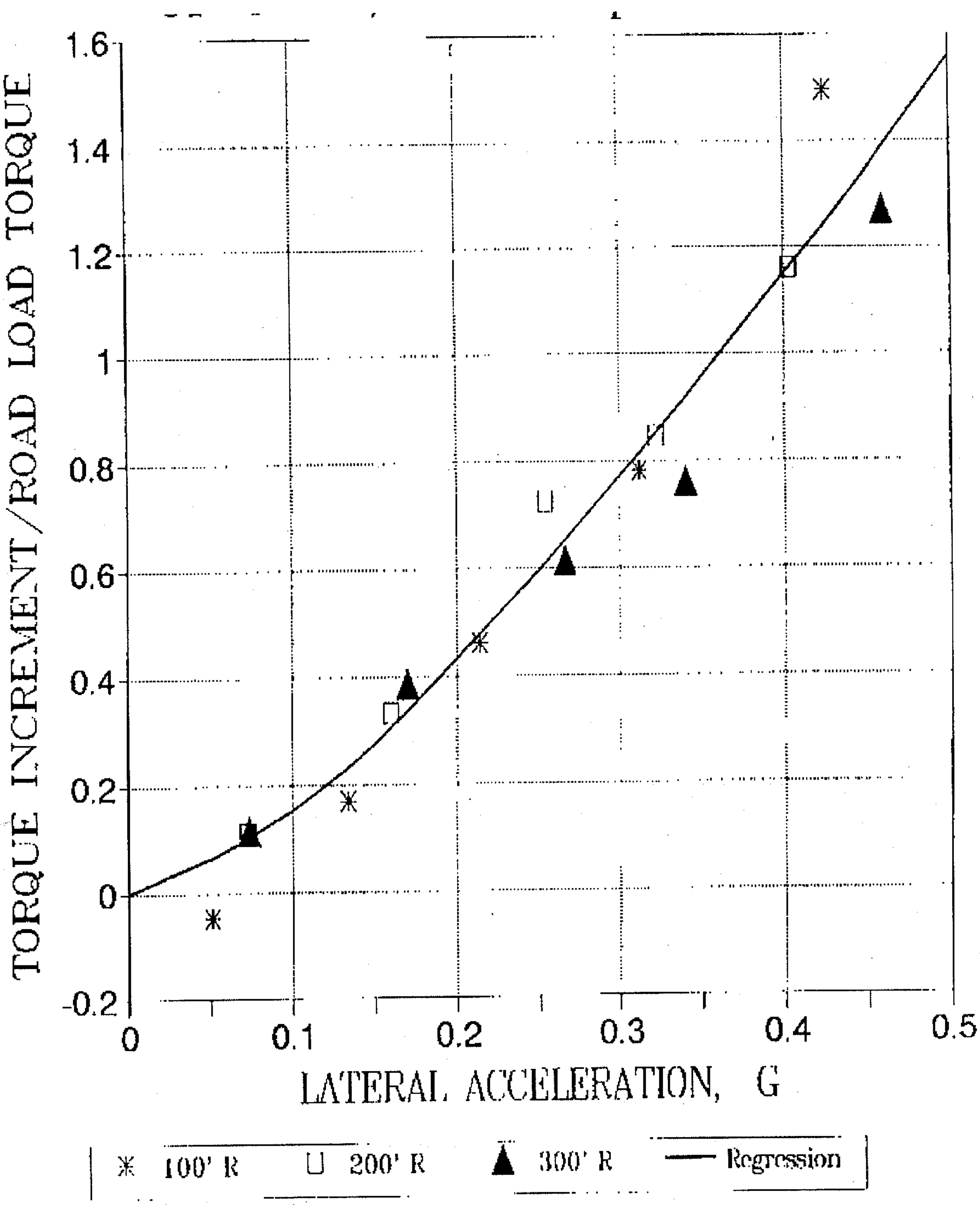
$28.1 A_{Lat} + 512 (A_{Lat})^2 : r^2 = 0.986$

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 ΔTc = 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

$A_{Lat} > 0.147$ 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)

$0 \leq A_{lat} \leq 0.147$ 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)

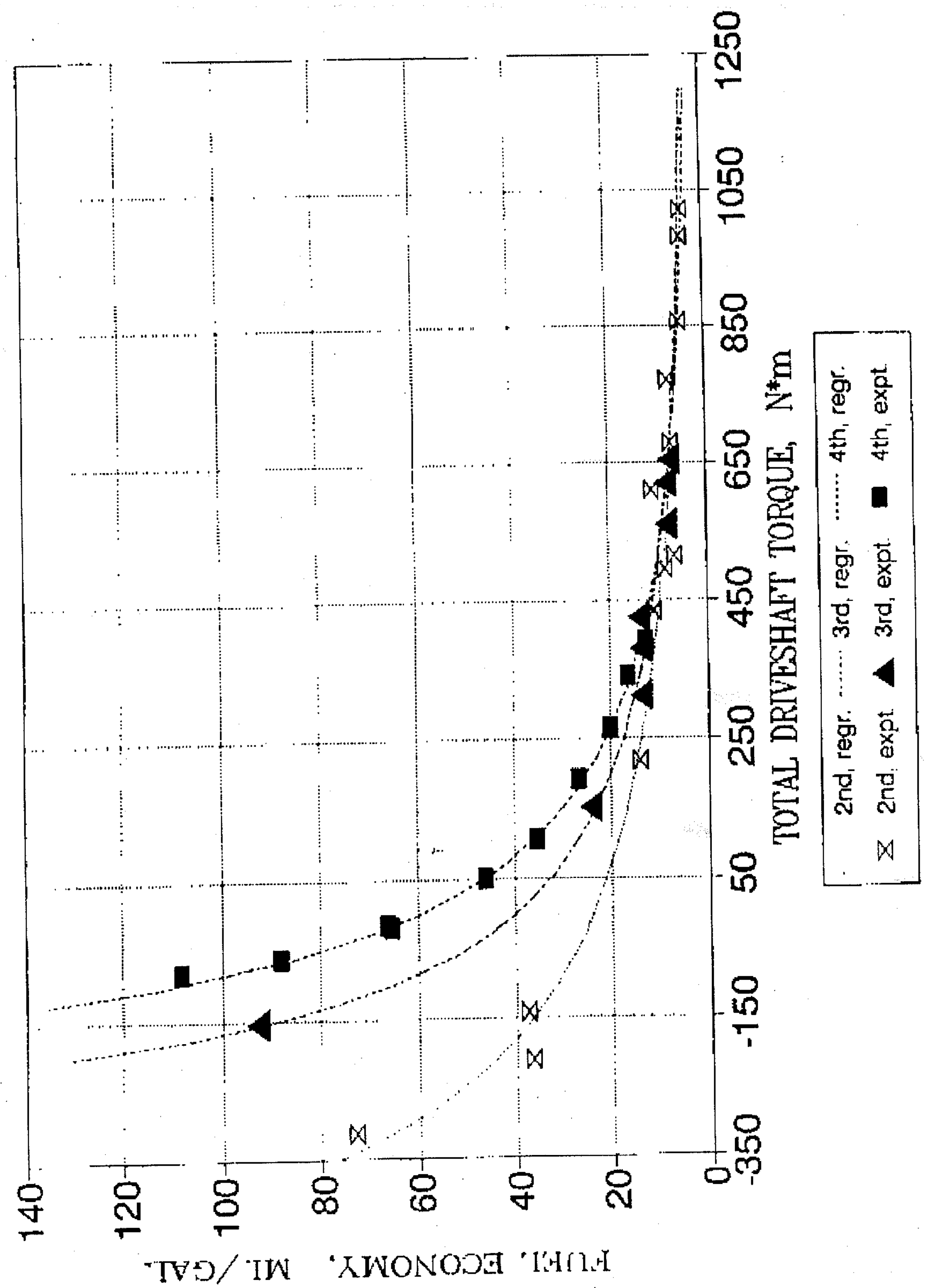
REGRESSION EQUATION:

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.

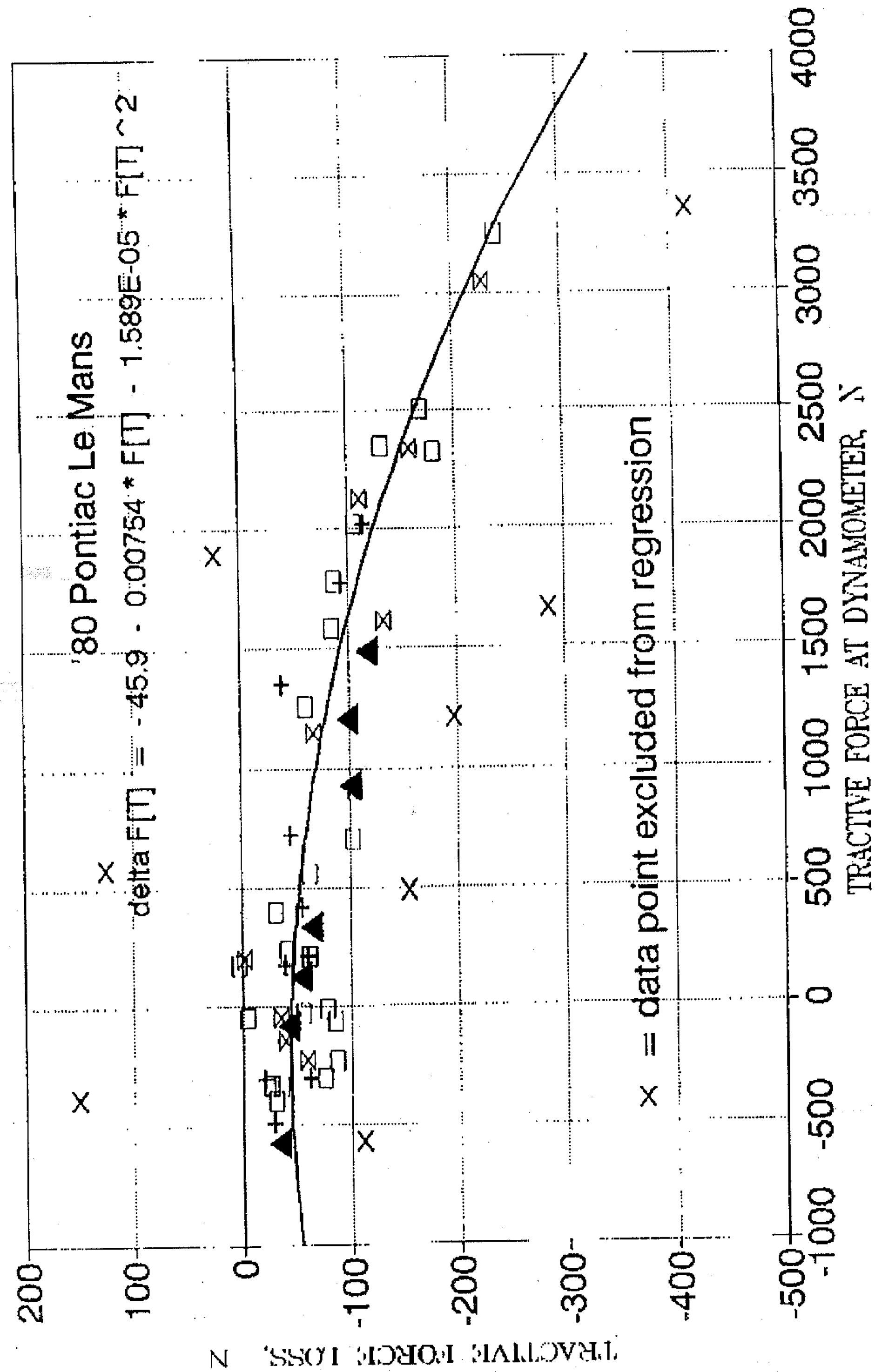


Figure 7. Tractive force loss vs. total force.

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:

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

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.

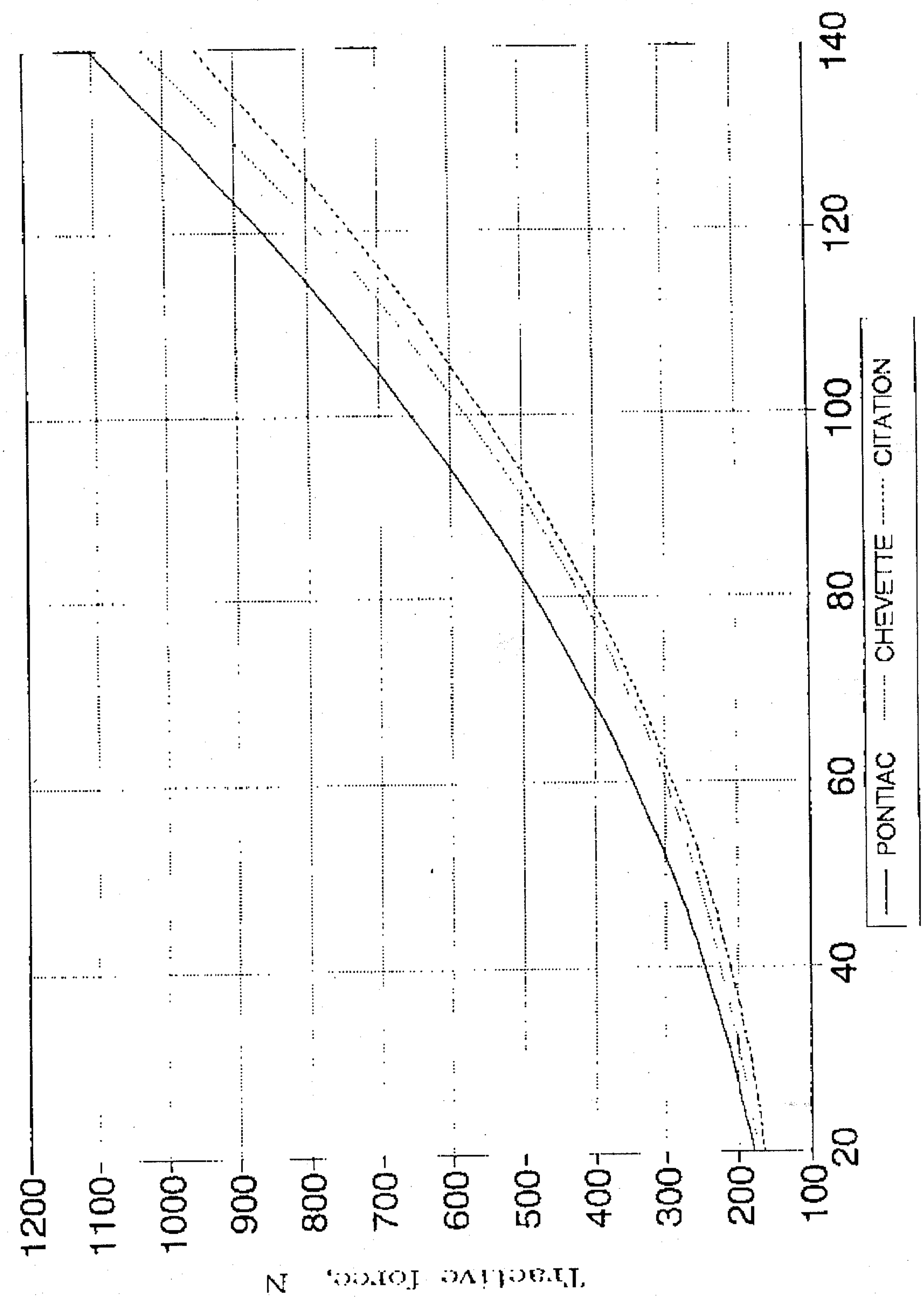


Figure 8.   Road load derived tractive force vs. speed - three cars.

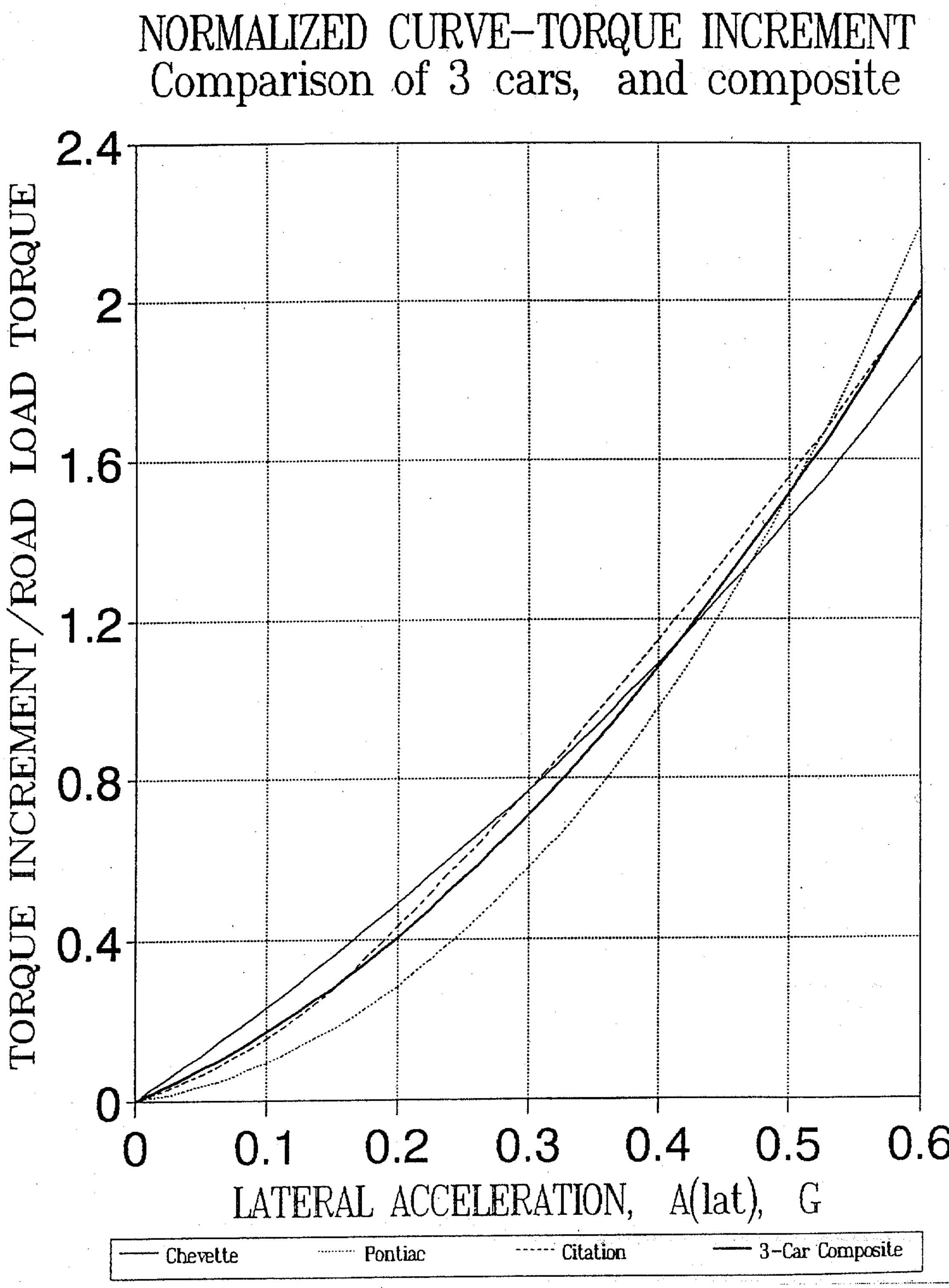


Figure 9. Normalized curve-torque increment - three cars.

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

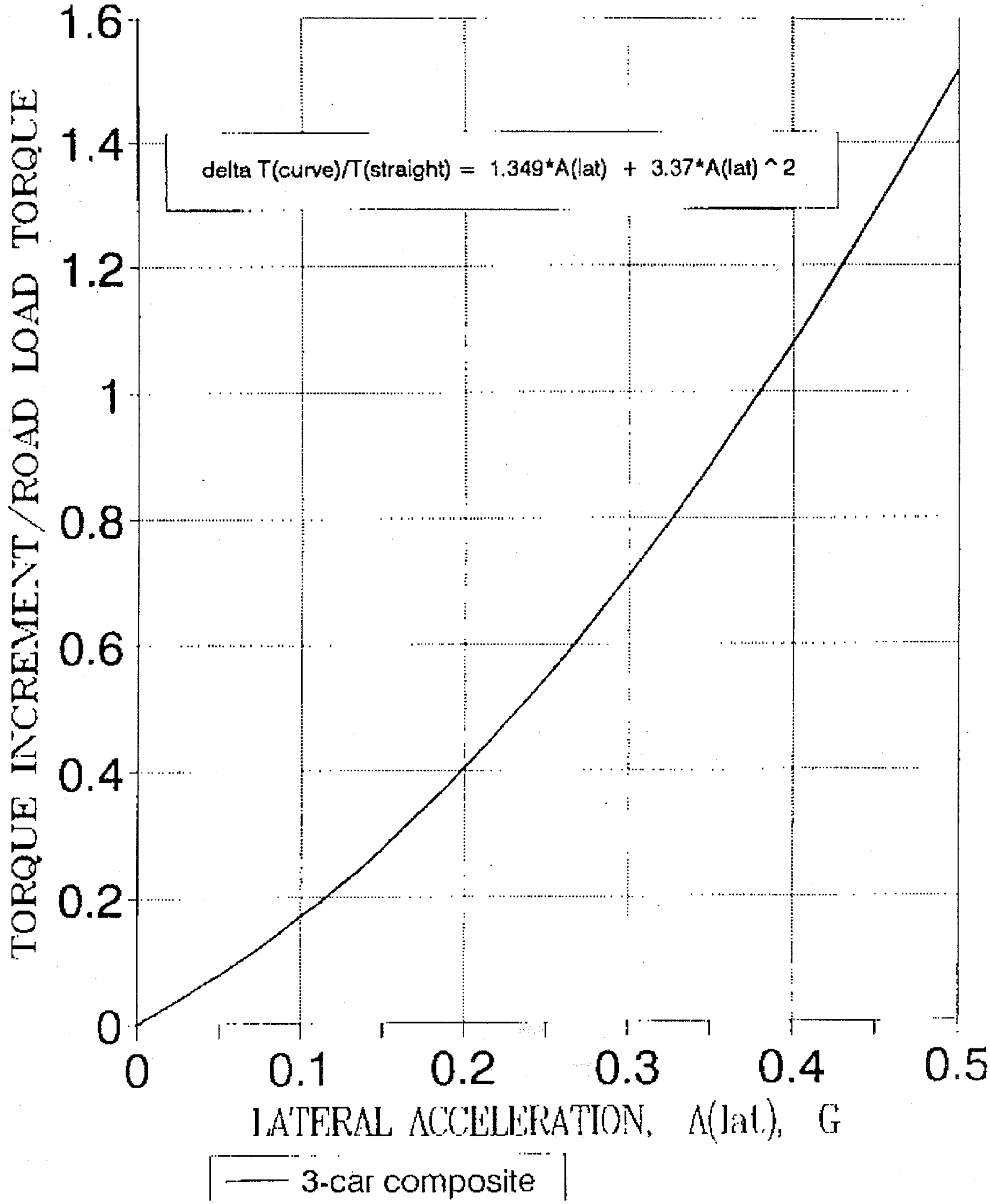
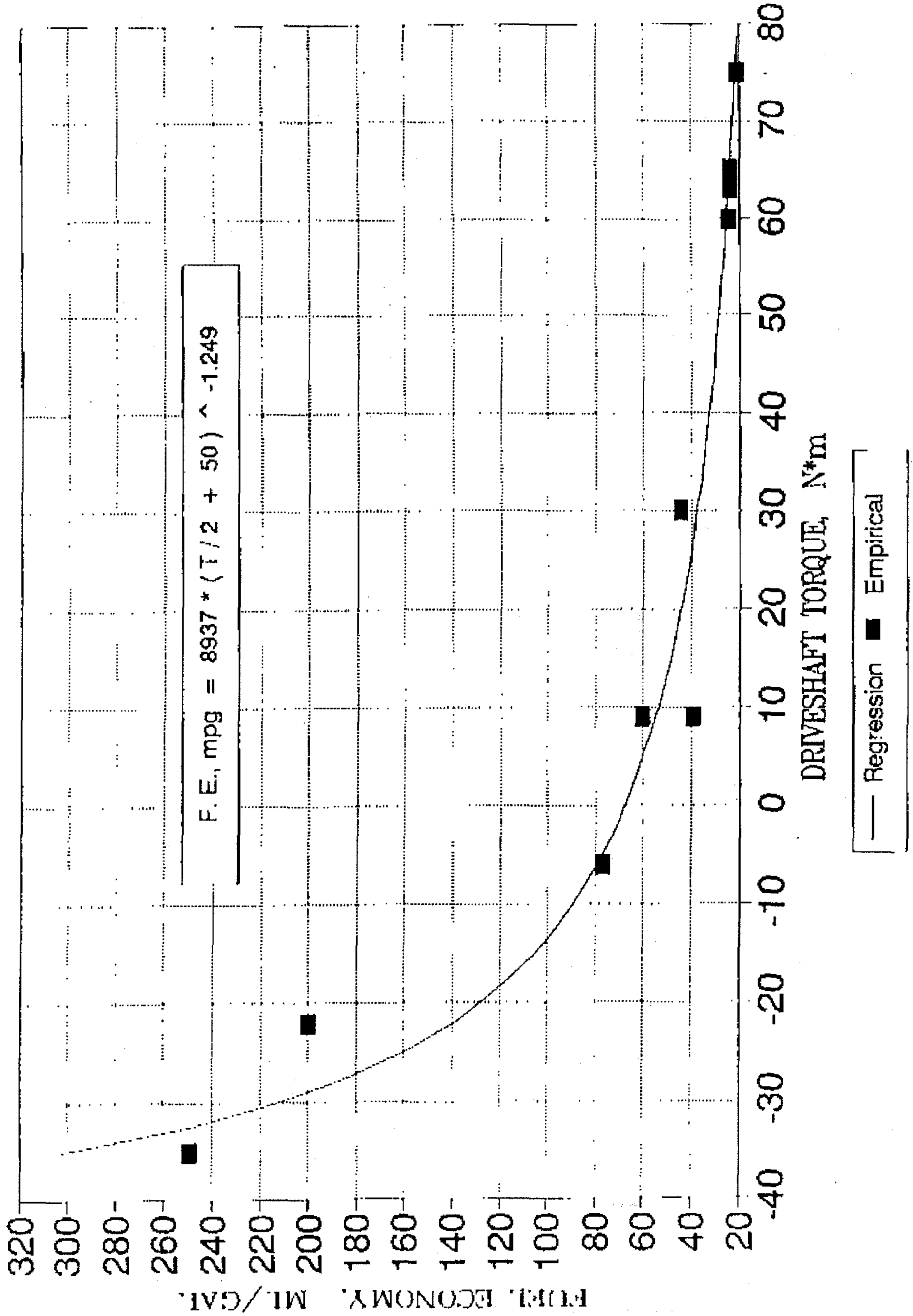


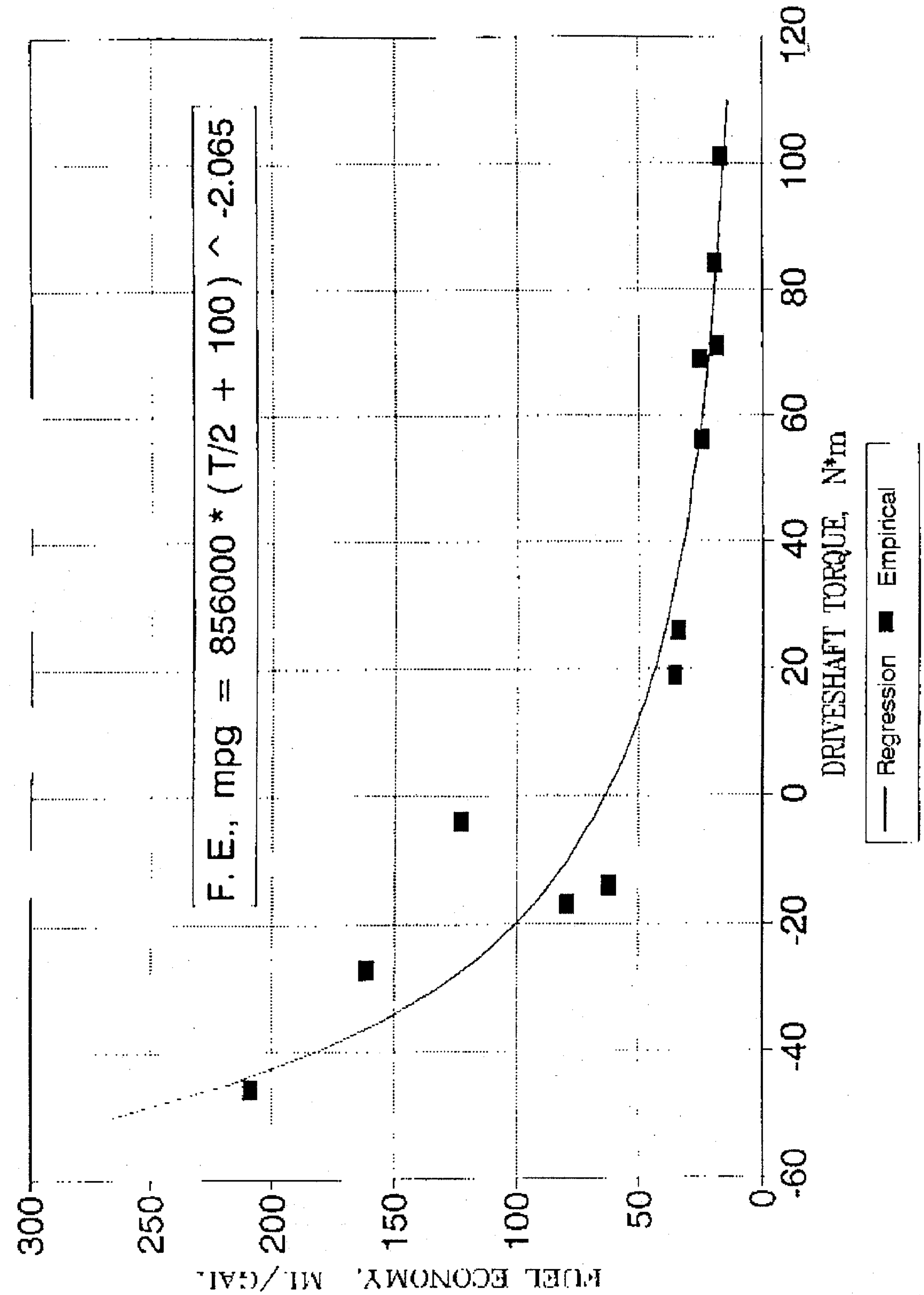
Figure 10. Composite normalized curve-torque increment - three cars.



1

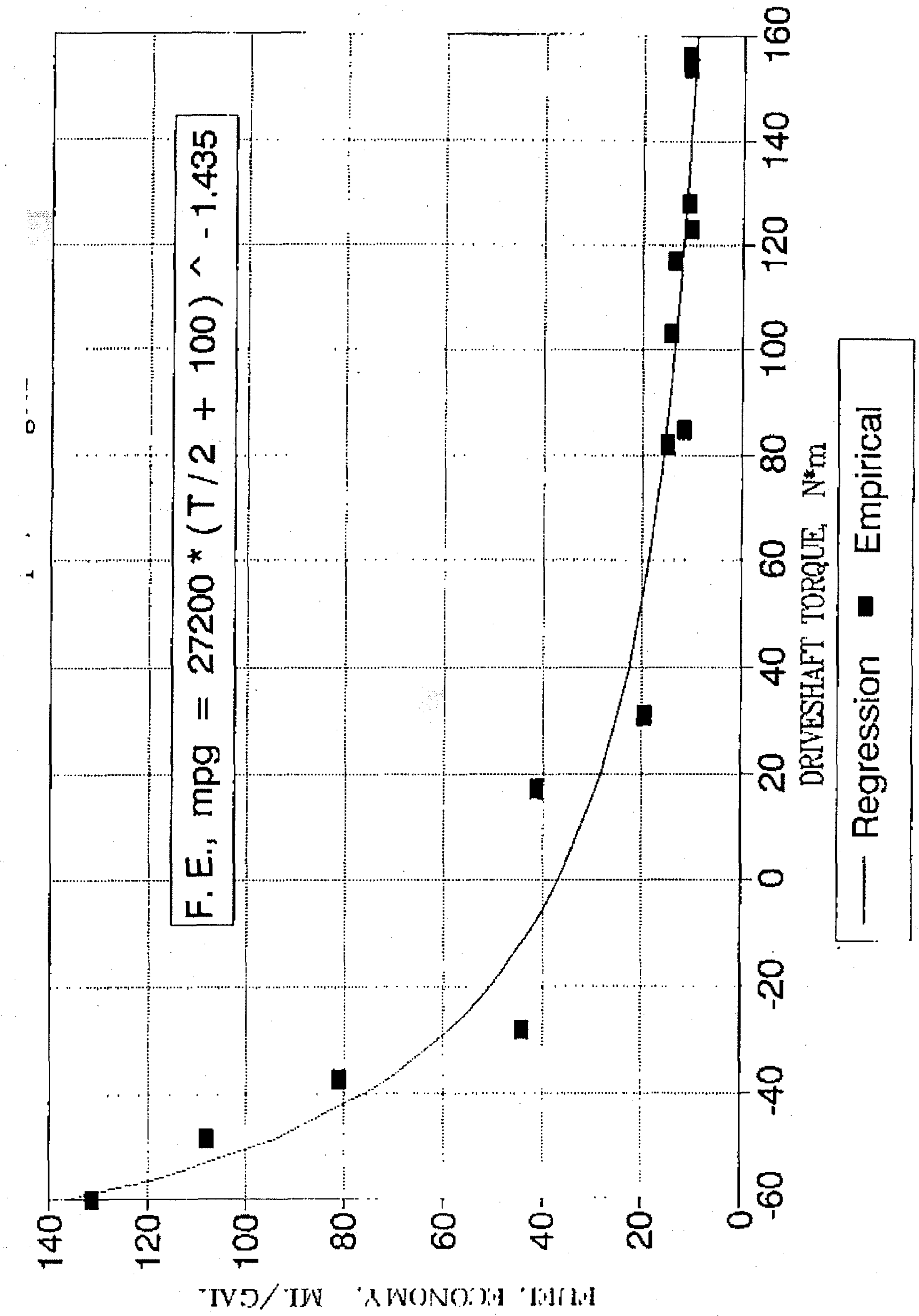
1 mi/gal =.42 km/L

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



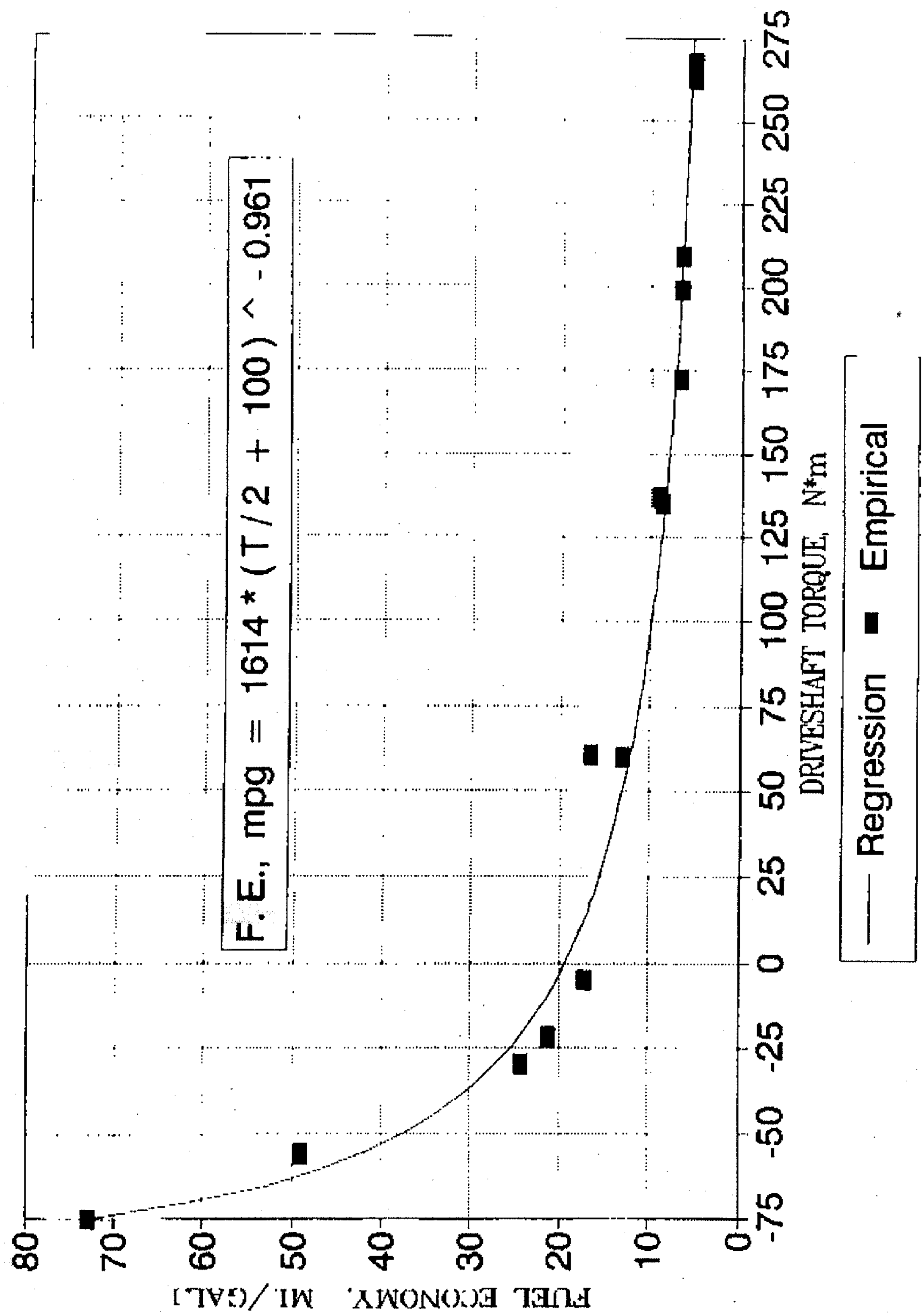
1 mi/gal =.42 km/L

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)

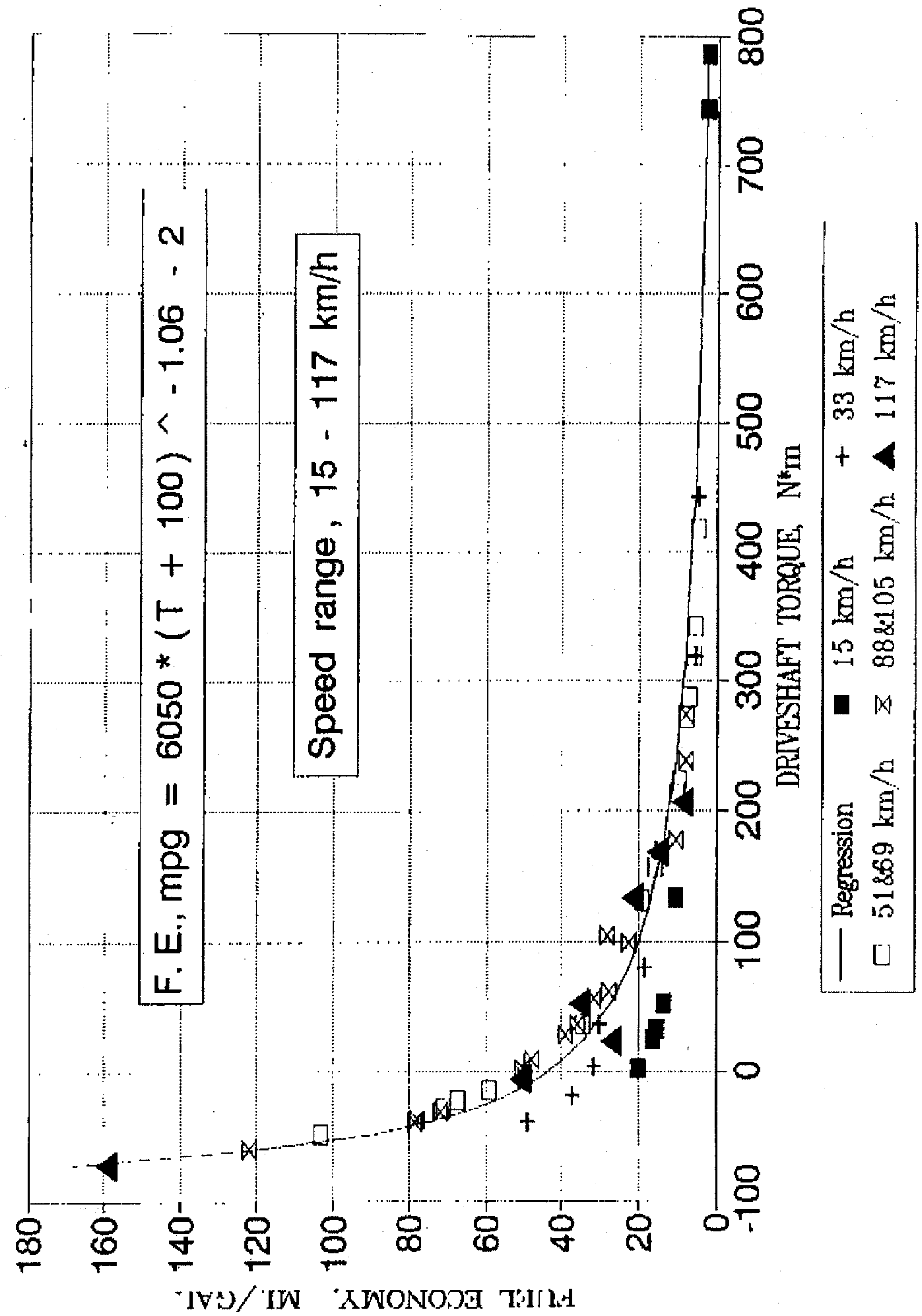
REGRESSION EQUATION:

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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