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LIGHT VEHICLE NOISE: VOLUME III — IDENTIFICATION OF NOISE SOURCE COMPONENTS AND EVALUATION OF NOISE REDUCTION TECHNIQUES

For

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1.0 INTRODUCTION

As part of the evaluation of the noise produced by motor vehicles, EPA has sponsored the development of a light vehicle noise test procedure[1] and the acquisition of a comprehensive noise data base on 66 new (1977 model year) light vehicles.[2] This data base defines the noise levels associated with various types of light vehicles, and identifies vehicle types which are noisier than others. Among the noisier vehicle types are those expected to be more prevalent in the future due to increased fuel efficiency. If federal action is taken to limit the noise emissions of light vehicles, it must be demonstrated that any required noise reduction is technically feasible. Accordingly, a study has been conducted to define in detail the noise source emission characteristics of several light vehicles, identify methods of reducing the noise, and demonstrate the feasibility of these methods.

In the present study, five light vehicles were selected from those reported in References 1 and 2. They consisted of three subcompact automobiles, one Diesel powered automobile and one light truck.* Vehicle selection criteria and specification of the test vehicles are presented in Section 2.0.

Testing was conducted on the inertial dynamometer located at Wyle's Norco, California, test facility. This dynamometer permits the simulation of an acceleration load on a stationary vehicle. The use of a stationary facility permitted detailed source identification procedures which would not be practical in a moving test, and also provided a degree of repeatability which cannot normally be achieved with a moving test. The inertial dynamometer, instrumentation, and experimental arrangement are presented in Section 3.0.

The five test vehicles were operated over a range of throttle settings from no load to full throttle, and speeds up to 90 percent of rated-engine speed. The test matrix included the test condition defined in Reference 1. A variety of microphone positions were used, including 7.5m (25 ft) and 15m (50 ft) sideline, 0.5m (20 in) from the exhaust outlet, and

*Wyle would like to thank General Motors Corporation for providing the Diesel automobile, and Ford Motor Company for the light truck.

a variety of near-field positions. Vehicles were first tested in their stock condition, then tested in a variety of modified configurations. The modified configurations included various degrees of wrapping and component removal for source identification, and demonstration configurations of noise reduction modifications. The test procedures, vehicle configuration, and data collection and reduction are discussed in Section 4.0.

The noise source characteristics of the vehicles and their components are presented in Section 5.0. This includes identification of major source components and the rank-ordering of their importance. It was found that the engine was usually the dominant noise source. Exhaust noise was generally not important at most operating conditions.

Noise reduction techniques are presented in Section 6.0. Methods specifically investigated and demonstrated included flow-through engine enclosures, cooling fan changes, and exhaust system modifications.

In addition to the five vehicles selected for testing, the opportunity arose to test a turbocharged Diesel powered automobile.* Moving tests were conducted on this vehicle at EPA's Noise Enforcement Facility in Sandusky, Ohio. Data for this vehicle are presented in Appendix A.

The feasibility of a stationary unladen test, similar in concept to the IMI test used for heavy trucks, was briefly investigated. Unladen tests were conducted on three vehicles, and compared to measurements under acceleration. These data are presented in Appendix B.

*We would like to thank Volkswagenwerk AGB for providing this vehicle.

Table 1. Specifications of Test Vehicles*

Vehicle	Test Vehicle No.**	Size	Curb Weight	Engine	CID	BHP	Torque	Trans.	Rear Axle	Tire Size
Ford F-150	015	LT	4,590	V8	351	168 @ 3800	270 @ 2600	4M	4.11:1	L78-15
Toyota Corolla	005	SC	2,325	L4	97	75 @ 5800	83 @ 3800	5M	3.91:1	165 SR 13
Chevrolet Chevette	019	SC	1,958	L4	97	63 @ 4800	82 @ 3200	3A	3.70:1	155-13
Oldsmobile Diesel	--***	LS	4,120	V8	350	120 @ 3600	220 @ 1600	3A	2.41:1	GR78-15
Ford Pinto	047	SC	2,477	V6	171	93 @ 4200	140 @ 2600	3A	3.00:1	BR78-13

1. *Units are: Curb Weight - lbs; CID - cu. in., BHP - BHP @ RPM; Torque - ft. lbs.
2. **See Reference 2.
3. ***Similar to Vehicle #053.

2.0 TEST VEHICLES

Table 1 lists the specifications of the five vehicles selected for testing. Selection was on the following basis:

1. Models selected from the 66 vehicles for which moving tests had been performed.[2]
2. Noise levels measured according to the EPA light vehicle noise test procedure were above average.
3. Include one light truck.
4. Include models expected to be more common in the future due to fuel economy considerations.
5. Include both automatic and manual transmissions.
6. Front engine/rear drive layout only, due to the mechanical constraints of the flywheel dynamometer.

Four of the five vehicles were the same vehicles previously tested in the study described in Reference 2. The Oldsmobile Diesel was mechanically similar to a model tested in that study. Table 2 summarizes the moving test results for these five vehicles.

In addition to these five vehicles, tests were conducted on an experimental turbocharged Diesel subcompact. The test results for this vehicle are presented in Appendix A.

Table 2. Moving Test Noise Levels (L_A at 15 m (50 ft), dB) For Test Vehicles from Reference 2

Vehicle	No.	EPA Urban Test	SAE J986a
Ford F-150	015	70.7	74.3
Toyota Corolla	005	67.5	72.4
Chevrolet Chevette	019	69.1	--
Oldsmobile Diesel	053	69.7	74.5
Ford Pinto	047	68.6	78.1

3.0 TEST FACILITY AND INSTRUMENTATION

3.1 Inertial Dynamometer

Load simulation on stationary test vehicles was accomplished using an inertial dynamometer. This consisted of a 1.22 m (4-foot) diameter, 200 kg (450-pound) steel flywheel mounted on a horizontal shaft, as shown in Figure 1. The vehicle was installed by disconnecting the driveshaft universal joint at the rear axle and connecting it to the dynamometer input shaft. A pair of short shafts provided clearance under the differential. Two heavy-duty, 4-speed manual automobile transmissions were used for reduction gearing between the dynamometer input shaft and the flywheel. Table 3 summarizes the operating capabilities of the inertial dynamometer, and Table 4 lists the gear ratios available. Figure 2 is a photograph of the dynamometer. To eliminate gear and bearing noise from the dynamometer, an enclosure was constructed of 1/2-inch plywood lined with fiberglass. Figure 3 is a photograph of the site, showing this enclosure in place and a vehicle installed.

The dynamometer load was adjusted to each vehicle by selecting appropriate gearing so as to match the test weight of the vehicle. The matching condition is that the kinetic energy of the flywheel equal the kinetic energy of the vehicle at the equivalent road speed.

The equivalent road speed is

$$V = \omega_s R_T / G_R \qquad (1)$$

where ω_s = driveshaft rotational speed;

R_T = tire rolling radius;

G_R = vehicle rear-end ratio.*

The flywheel speed is

$$\omega_F = \omega_s / G_D \qquad (2)$$

where G_D = dynamometer gear ratio.

* Gear ratios in this report are written as step-down ratios, i.e., input speed divided by output speed.

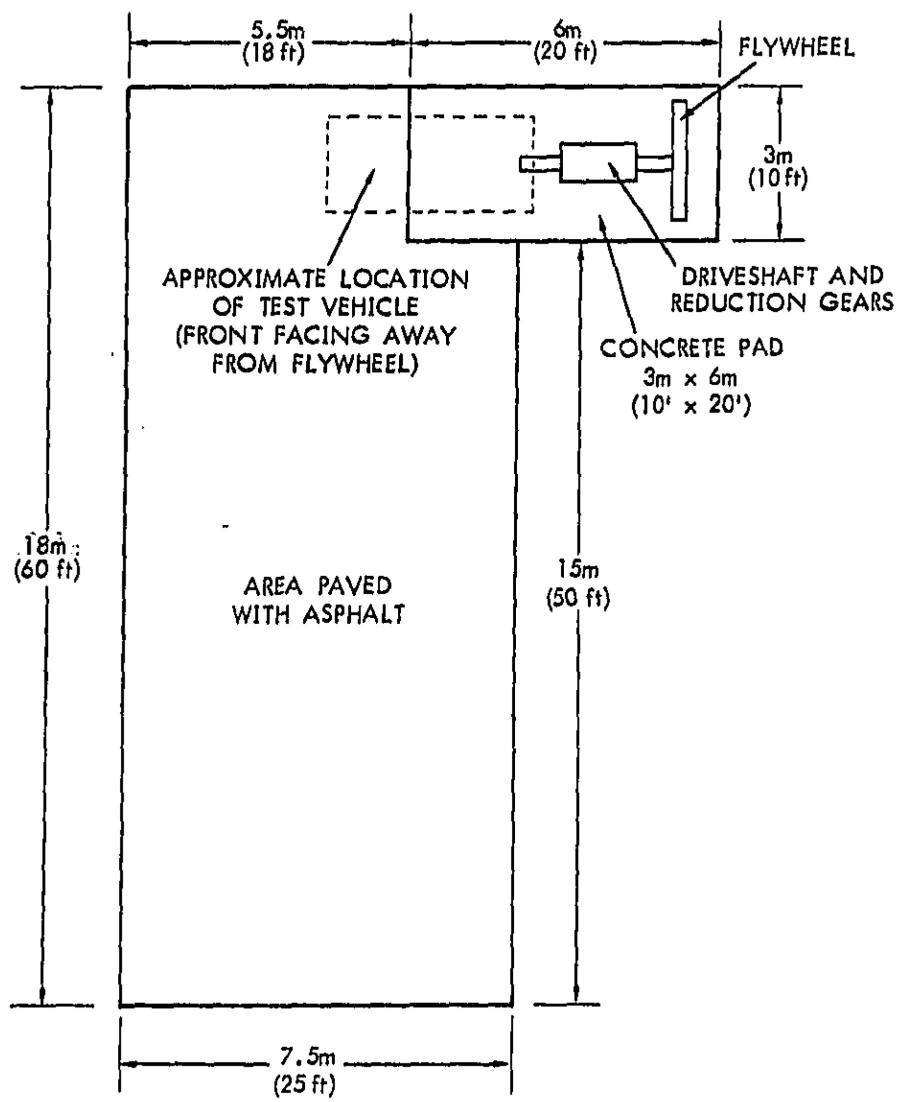

Figure 1. Layout of Inertial Dynamometer Facility.

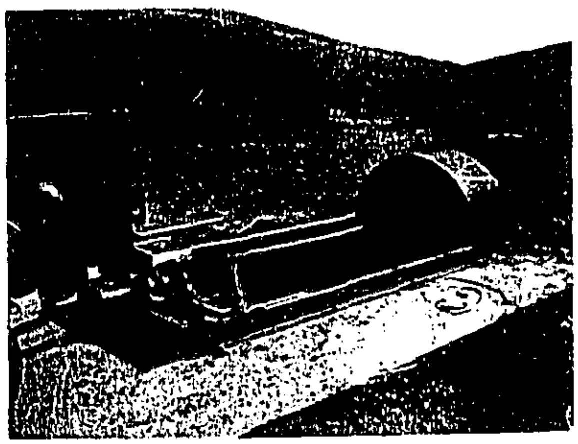

Figure 2. Inertial Dynamometer.

Figure 3. Test Site and Dynamometer, With Vehicle Installed.

Table 3. Inertial Dynamometer Specifications

Maximum Speed:	2000 RPM (Equivalent to 6800 kg Vehicle at 56 km/h)
Maximum Input Torque:	800m - N (600-foot-pounds)
Flywheel Diameter:	1.22 m (4 feet)
Flywheel Mass:	200 kg (450 pounds)
Momemt of Inertia:	37.2 kg-m^2 (27.95 slug-ft^2)

Table 4. Gear Ratios Available

		Front Transmission Gear			
		1	2	3	4
Rear Transmission Gear	1	5.825	4.337	3.461	2.656
	2	4.182	3.114	2.485	1.907
	3	3.033	2.258	1.802	1.383
	4	2.193	1.633	1.303	1.000

Matching vehicle kinematic energy $\frac{1}{2} m_v V^2$ to flywheel kinetic energy $\frac{1}{2} I_F \omega_s^2$,

$$\frac{1}{2} m_v \omega_s^2 R_T^2/G_R^2 = \frac{1}{2} I_F \omega_s^2/G_D^2 \tag{3}$$

Solving for the dynamometer gear ratio,

$$G_D = \frac{G_R}{R_T} \sqrt{\frac{I_F}{m_v}} \tag{4}$$

When applying Equation (4), the vehicle mass was based on the curb weight plus 175 pounds. This represents a test weight with one person. If the exact ratio required was not available (see Table 4), the next higher value was taken. This would give a slightly higher inertial loading than is required, which compensates somewhat for the neglect of load from rolling and air resistance.

The equivalent road speed, as a function of flywheel speed, is given by:

$$v = \omega_F R_T \frac{G_D}{G_R} \tag{5}$$

The equivalent acceleration is the first derivative of Equation (5); namely:

$$a = \dot{\omega}_F R_T \frac{G_D}{G_R} \tag{6}$$

When applying Equations (5) and (6) to calibrate the data recording system, the actual value of G_D was used if it differed from the ideal value computed from Equation (4).

At its maximum speed of 2000 RPM, the energy in the flywheel is equivalent to that of a 6,800 kg (15,000-pound) vehicle at 56 km/h (35 mph). This capacity is more than adequate for testing light vehicles. There is a slight restriction due to the 800 m - N (600-foot-pound) torque limit. Full-throttle accelerations with the vehicle in first gear (typically 3:1) are limited to vehicles with a net torque of less than 270 m - N (200-foot-pounds). This restriction permits full-throttle testing of most, but not all, light vehicles in first gear. Full-throttle acceleration tests can be conducted in second gear (typically 2:1 or less) for all light vehicles without exceeding the torque limit. All tests at the throttle setting corresponding to the EPA urban acceleration mode were well within the capability of the dynamometer.

3.2 Instrumentation

Basic data acquired consisted of:

- Two acoustic channels
- Engine RPM
- Flywheel RPM
- Flywheel acceleration
- Manifold pressure
- Automatic transmission oil temperature

Figure 4 is a block diagram of the instrumentation system. Basic recording was on a Gulton 8-channel chart recorder. A-weighted sound levels were recorded on the chart recorder. Spectra of events of interest were obtained during testing using a real-time analyzer. Acoustic data were also recorded on magnetic tape in the event that further spectral analysis was desired. An X-Y plotter was used to plot one noise channel versus engine RPM for field evaluation of noise modifications.

Specific instrumentation consisted of the following:

- Two 1-inch General Radio crystal microphones were used for sound measurements. Unweighted noise was recorded on a two-channel Kudelski Nagra SJ-IV tape recorder, at 3-3/4 ips. Two GR1933 sound level meters were used to provide A-weighted levels which were recorded on the chart recorder. A B&K 3347 real-time analyzer was used to obtain spectra from one acoustic channel during tests. A Tektronics 31 programmable calculator was used to trigger the RTA at desired engine RPM during acceleration tests.

- Engine RPM was obtained by a frequency-to-voltage converter connected to the ignition system. For the Diesel, an Electro 3011 HT magnetic pickup sensed the passage of four metal pieces epoxied to the crankshaft pulley.

- Flywheel RPM was detected by an Electro 3011 HT magnetic pickup adjacent to a 60-tooth gear on the flywheel end. A frequency-to-voltage converter was used to process the signal. The voltage output was electronically differentiated to give acceleration.

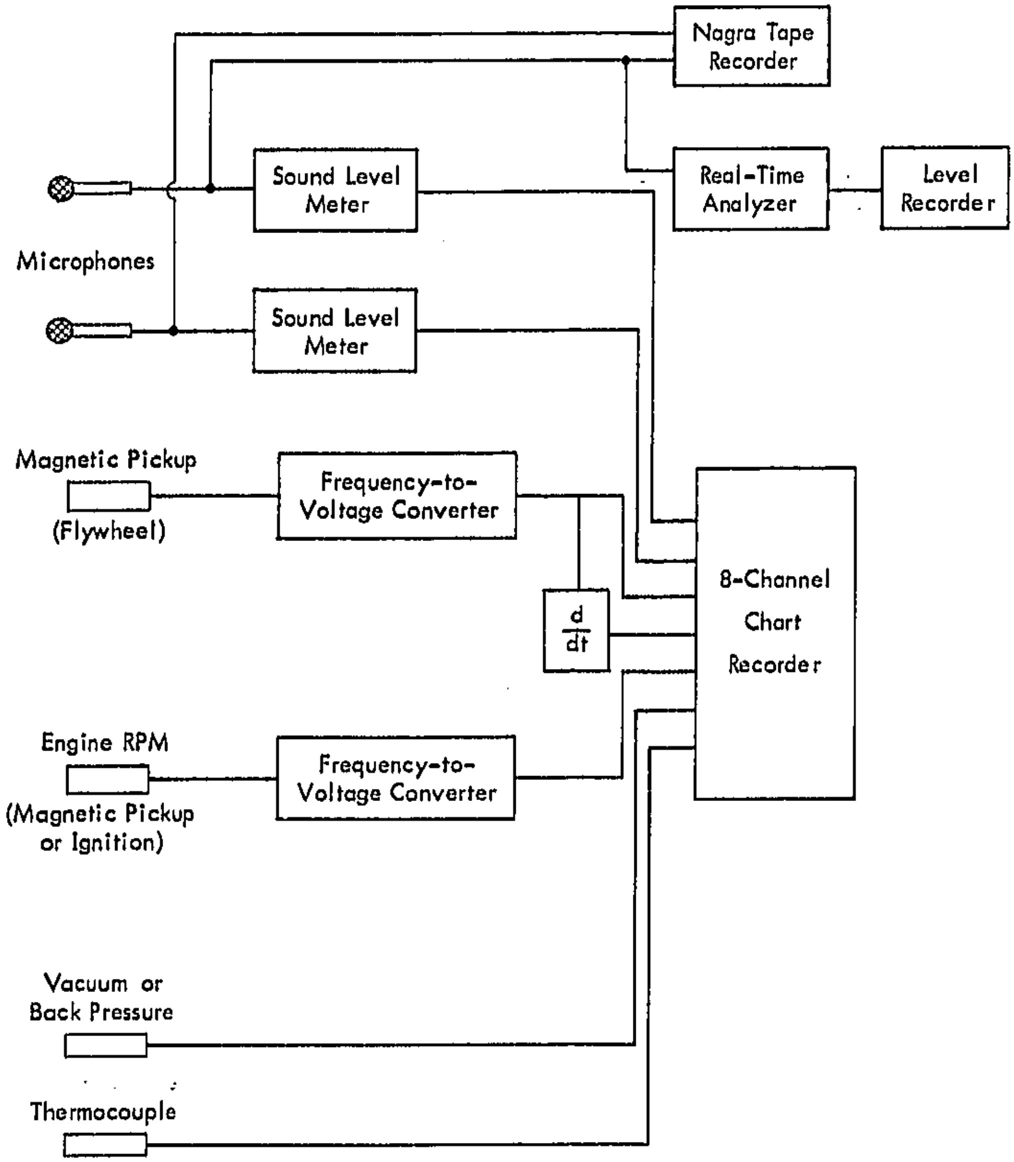

Figure 4. Block Diagram of Instrumentation and Data Acquisition System.

- Manifold pressure was detected using a Microsystems PA 0055-1001 pressure transducer. It was connected to the engine via an existing vacuum tap in the manifold.

- Automatic transmission oil temperature was monitored with a thermocouple attached to the end of the dipstick.

On the Diesel engine tested, exhaust back pressure was monitored. A fitting was brazed to the exhaust headpipe near the manifold flange, and the pressure transducer connected. Also for the Diesel, vibration measurements were taken by replacing one microphone with an Endevco model 2242 piezoelectric accelerometer.

All of the non-acoustic data were smoothed via filter circuits with a 100 ms time constant. This is the same smoothing as was used in the moving tests described in References 1 and 2.

Figure 5 shows the recording instrumentation as set up in the control room.

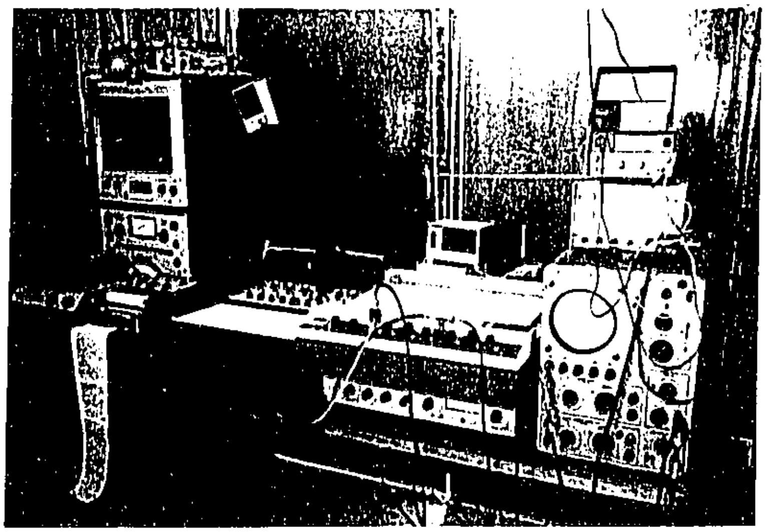

Figure 5. Instrumentation in Control Room.

4.0 TEST PROCEDURE

4.1 Vehicle Operating Modes

Noise data were collected under the following conditions:

- At steady speeds of 50, 60, 70, 80, and 90 percent of rated RPM, with transmission in neutral.
- Under acceleration for five different throttle settings. These settings consisted of:
  - That corresponding to the EPA urban acceleration test condition.[1]
  - That producing an acceleration of about 0.03g at the same engine RPM as that achieved in the EPA urban acceleration test. This simulated the load of a cruise condition.
  - Full throttle.
  - Two intermediate settings, one between the 0.03g and the EPA acceleration, and one between the EPA acceleration and full throttle.

Accelerations were performed with the vehicle in first gear and in drive (automatic transmission) and in first* and second gears (manual transmission).

When conducting tests at steady RPM, the throttle was adjusted to achieve the desired speed. For the acceleration tests, a hand-throttle mechanism with an adjustable stop was used (Figure 6) and the stop adjusted to positions determined in the initial setup. The acceleration throttle settings thus consisted of five specific opening angles of the carburetor throttle plate (or the equivalent for the Diesel).

A baseline series of measurements was performed for each vehicle using all throttle settings and two gear ratios. It was found that noise was not a strong function of throttle setting, so that five throttle settings were more than needed. Operating modes for tests after each baseline series were therefore limited to steady speed, partial throttle corresponding to the EPA acceleration test condition, and full throttle.

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*For the Ford F-150, first gear was not normally used in ordinary driving. Second and third gears were therefore used for the test.

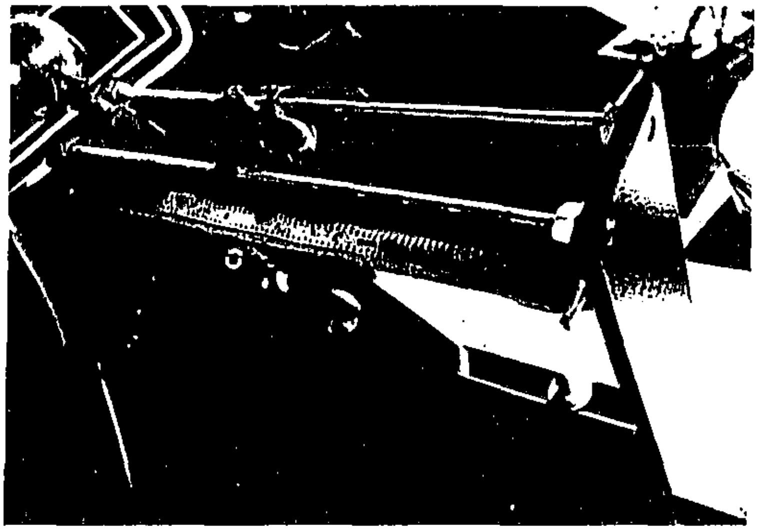

Figure 6. Hand Throttle and Stop Mechanism.

It was also found that noise at a given engine RPM and throttle setting did not depend on which gear was used. Acceleration at a given RPM and throttle setting was also found to be proportional to the ratio between the two gears. The accelerations encountered were sufficiently low for the power and noise characteristics to be quasi-static. Acceleration test conditions after baseline were therefore done in first gear only. Limited tests were also conducted with the transmission in neutral to see if quasistatic conditions occurred for inertial load by engine inertia alone. These tests are described in Appendix B.

4.2 Microphone Positions

The following microphone positions were used:

1. 7.5 meters (25 feet) to the left side of the vehicle, centered on the wheelbase. This was the basic far-field position and was used for all tests.
2. 15 meters (50 feet) to the left side of the vehicle, centered on the wheelbase.
3. 1.5 meters (5 feet) in front of the vehicle, aligned with the vehicle centerline.
4. 0.5 meter (20 inches) from the exhaust pipe outlet, at the same height as the outlet.
5. 0.5 meter (20 inches) from the muffler.
6. Various positions under the hood, 8 centimeters (3-1/4 inches) from the surface of various accessories and engine components.

Microphone positions 1, 2, and 3 were 1.2 meters (4 feet) above the ground. Only positions 1, 2, and 4 were used for quantitative determination of source components. Data from the other positions were used for qualitative assessments and to determine spectral shapes of individual components. Figures 7, 8, and 9 show typical microphone positions.

4.3 Vehicle Configurations

Each vehicle was first tested in its stock configuration. This was then followed by a series of tests with various components removed or shielded, and various degrees of engine compartment enclosure. Quieter components were also substituted. The vehicle was then restored to stock configuration and a final baseline series of measurements made as a consistency check. Modified vehicle configurations included:

Figure 7. Sideline Microphone Positions.

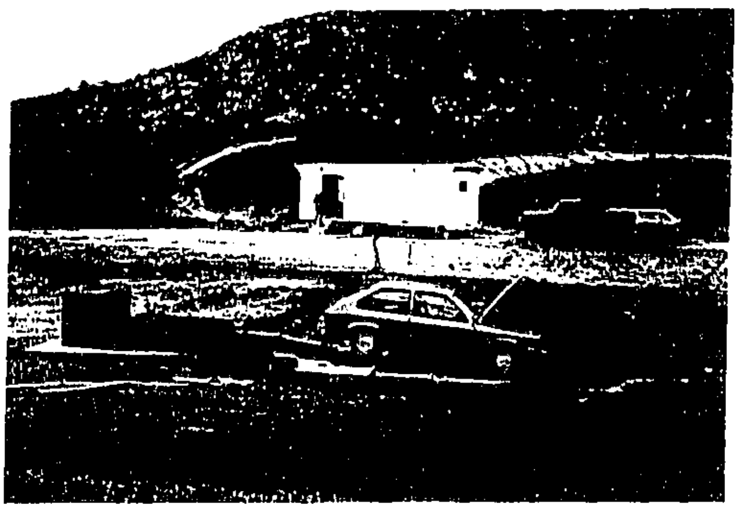

Figure 8. Sideline and Front Microphone Positions.

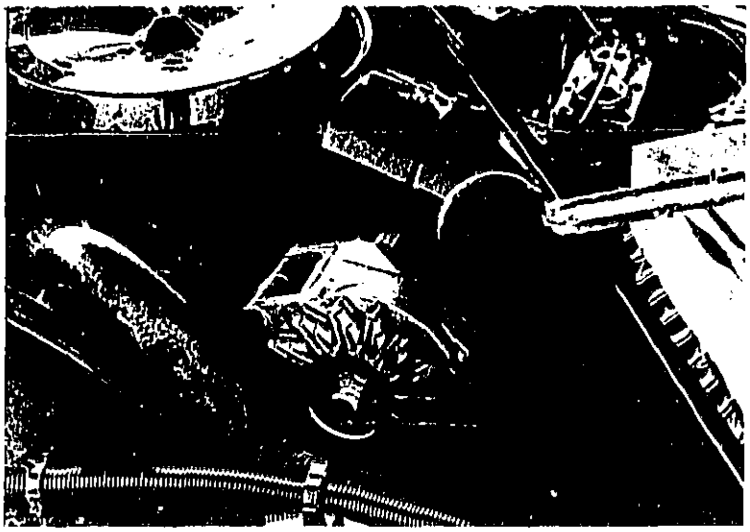

Figure 9. Typical Under-Hood Near-Field Microphone Position.

- Removal of the cooling fan. An external water supply was used to cool the engine when the fan was off.

- Installation of a clutch fan on vehicles equipped with a direct drive fan. Where available as an option, the unit supplied by the vehicle manufacturer was used. If not available, a unit from an aftermarket manufacturer was used.

- Removal of the alternator and/or other belt-driven accessories.

- Wrapping of exhaust system components with sheet lead (approximately 9.8 kg/m^2 (2 lb/ft^2)) and Kaowool. The Kaowool was used in pieces about one to three centimeters thick to provide padding between the lead and the exhaust components. Figure 10 shows a typical wrapping.

- Addition of an external muffler over the tailpipe. This consisted of a 1.8-meter (6-foot) long flex pipe, a large truck muffler of straight-through design, and 3-meter (10-foot) straight pipe. The internal diameter of these components was about 13 cm (5 inches). Figure 11 shows this muffler system.

- Modification to the original exhaust system. This was done on two of the vehicles. The Toyota's original exhaust system was removed and replaced with one having a larger muffler of type similar to the original, plus a glasspack straight-through muffler. On the Oldsmobile, a second muffler similar to the stock one was added in series, replacing a section of straight pipe ahead of the stock muffler.

- Addition of absorptive material in the engine compartment. This consisted of fiberglass and/or Kaowool about 5 cm thick (with NRC of about 0.75) placed over about half the surface area.

- Construction of engine compartment enclosures. These were constructed from lead sheet with wooden supports. Areas covered were the openings in the fender wells and the underside of the engine compartment. The underside of the engine compartment was covered with a lower enclosure, attached to the frame rails and the bottom of the radiator, and extending rearward as far as the firewall. The enclosures were left open at the rear to permit

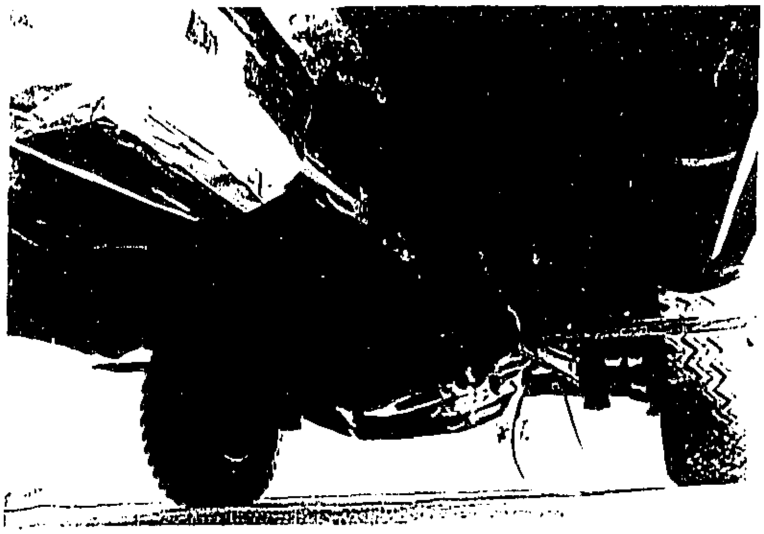

Figure 10. Wrapped Exhaust System, Ford F-150

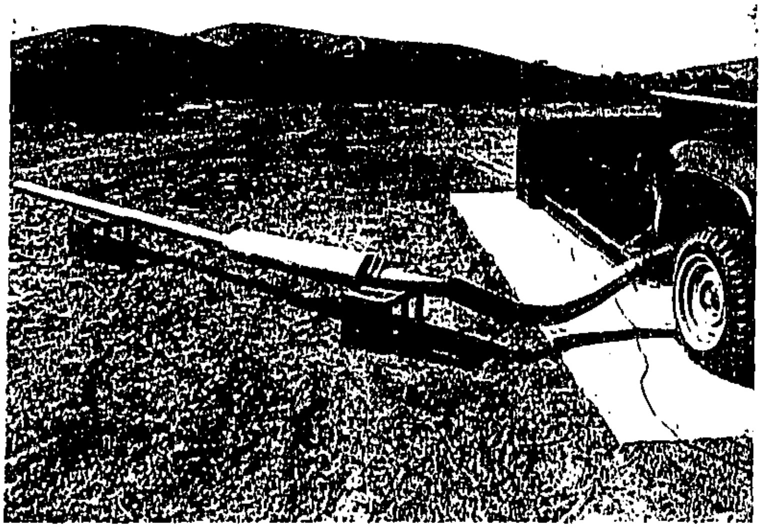

Figure 11. External Muffler System

cooling air to flow through. Figure 12 shows a typical enclosure installation. The geometry of the enclosures was restricted to that possible on production vehicles.

- Vinyl drapes over fenders, under the vehicle, etc., to help identify transmission paths. A vinyl material was used with mass of about 4.9 kg/m^2 (1 lb/ft^2), backed with open cell foam about 6mm thick.

These changes to the vehicle configuration were applied generally in order of source strengths. Table 5 shows the configuration matrix for the Toyota. The sequence of modifications was similar for the other vehicles.

4.4 Test Procedure and Data Reduction

Each vehicle was checked to ensure it was tuned to manufacturer's specifications, then installed on the test stand. The instrumentation system was calibrated and the throttle stop positions established. The vehicle was warmed up to normal operating temperatures, before noise measurements were conducted.

Steady, unloaded noise tests were performed at stablized engine speeds. In general, at least ten seconds of data were collected at each speed. Acceleration tests were performed four times in succession. This provided a test for run-to-run consistency.

Figure 13 shows a typical strip-chart recording for an acceleration test at partial throttle. The vacuum trace clearly shows when the throttle opened, and when a steady acceleration was established. The acceleration trace follows this closely. The small oscillations in the acceleration trace are due to torsional vibration in the flywheel. These are caused by unsteadiness in the engine at idle causing torque reversals; note the large spikes before opening the throttle. To minimize these, an operating technique was developed in which the throttle would first be slightly opened for long enough to smooth the flywheel motion, then fully opened to the stop.

The two noise channels on this figure are 7.5-meter sideline and exhaust outlet microphone positions. Note that there is some unsteadiness in the noise signals. Some of this is a characteristic of the engine, especially the exhaust noise at closed throttle,

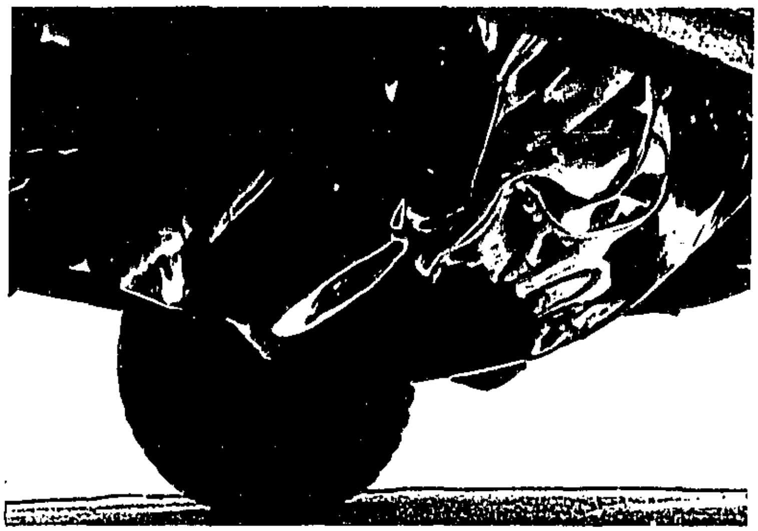

Figure 12. Lower Enclosure, Ford F-150.

Table 5. Test Configuration Matrix, Toyota Corolla

DESCRIPTION	Configuration Number												
	1	2	3	4	5	6	7	8	9	10	11	12	13
Stock	X												
Fan Off			X	X	X	X	X	X	X	X	X	X	X
Alternator Disconnected				X	X		X	X	X	X	X	X	X
Alternator Lead Wrapped (w/Absorption)													
Air Pump Disconnected					X	X	X	X	X	X	X	X	X
Water Pump Disconnected				X	X		X	X	X	X	X	X	X
Air Conditioning Disconnected													
Absorption Underhood										X	X		X
Heavy Vinyl Underhood										X	X		X
Lead Covering Gaps Above Frame								X	X	X	X		
Belly Pan								X	X	X	X		
Stock Fan Locked		X											
Modified Muffler							X						
Front of Car Wrapped in Lead									X	X	X	X	
Front Muffler Wrapped											X		

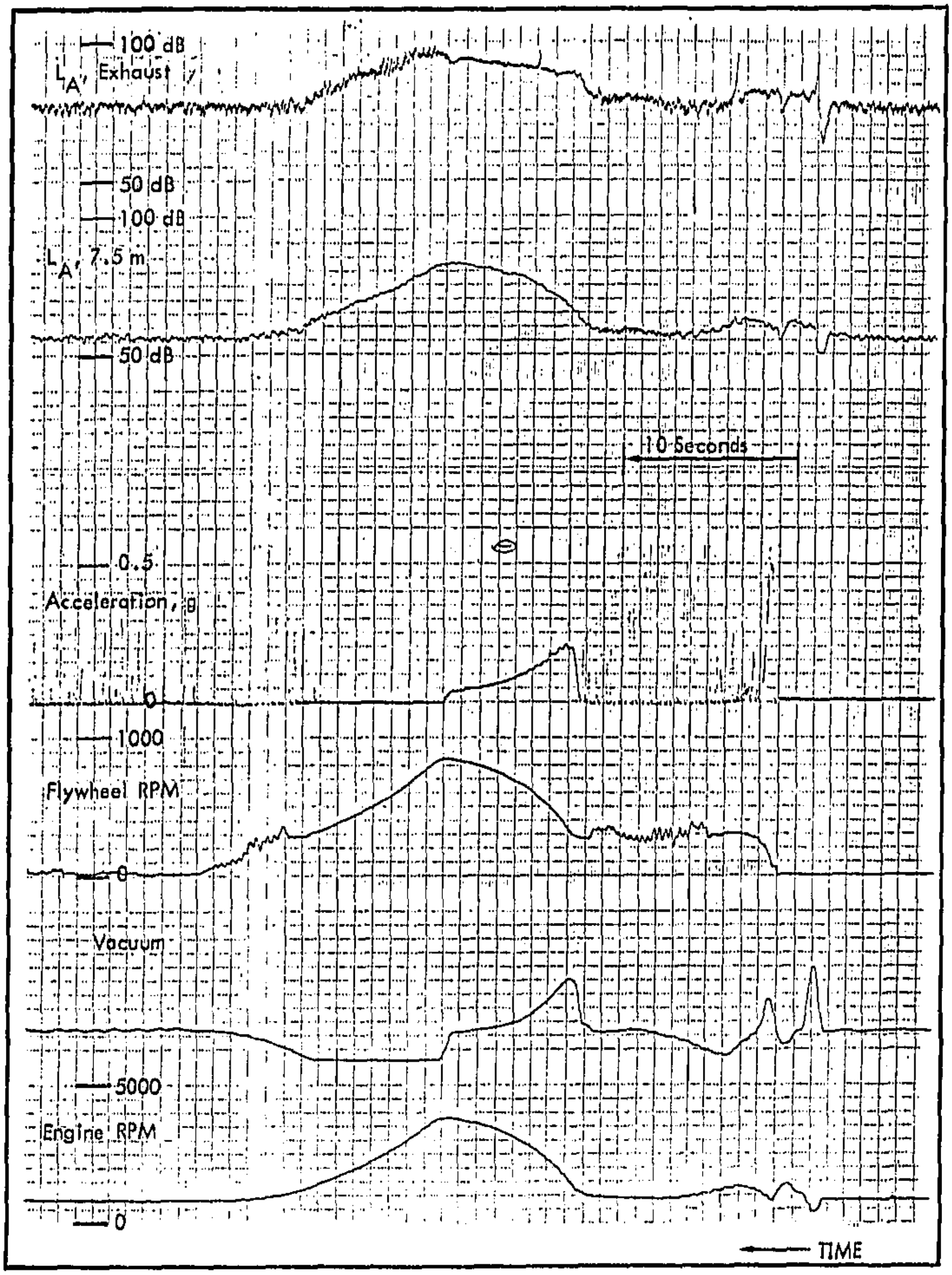

Figure 13. Typical Strip Chart Record, Ford F-150, Partial Throttle (Time – right to left).

while some at the 7.5-meter microphone may be due to atmospheric fluctuations. The fine-scale fluctuations seen in the figure are essentially random. To avoid variations due to random peaks at engine speeds of interest, data reduction for the present program was accomplished by smoothing the traces. This was the appropriate approach for the present program, where differences of noise between various vehicle configurations are of interest. It provides different levels, however, from the usual convention of taking the maximum fast response level. Referring to the noise traces during acceleration and during engine idle periods in Figure 13, the smoothed data reduction gives levels lower by about 1/2 dB and 1 dB, respectively, than maximum levels for the 7.5-meter microphone, and about 1/2 dB and 2 to 3 dB lower for the exhaust microphone. This difference should be kept in mind when comparing the present results with data from moving tests where the maximum fast response level is used.[2]

Using the smoothing convention for selecting levels at a given engine speed, data from the strip charts were read at the five steady speeds for unloaded tests, and at four to five speeds for the acceleration tests. One speed was that corresponding to the EPA test condition, one was the highest RPM achieved in the accelerations (usually about 90 percent of rated RPM), and one was the lowest RPM for which steady throttle was achieved (usually about half rated RPM). Values were also taken at one or two intermediate RPM, as needed, to provide smooth plots of noise level versus RPM. Levels for the acceleration runs were taken as averages of the values for four repetitions.

5.0 NOISE SOURCES AND CHARACTERISTICS

5.1 Noise Characteristics of the Test Vehicles

Figures 14 through 18 show the 7.5-meter sideline noise levels as a function of engine speed at various loads for the five test vehicles. The following characteristics may be seen:

- The relation between noise level and RPM is generally linear. This is in contrast with intuitive scaling laws which would be logarithmic with RPM.
- The effect of load on noise is less than the effect of RPM. The greatest difference between full and partial load noise at a given RPM is about 7 dB. Differences between level at full and half rated RPM are between 10 to 15 dB.
- The variation of noise from the Diesel — both with RPM and throttle change — is less than for the spark ignition engines.

Table 6 lists several quantitative parameters of Figures 14 through 18. These include average slope of the curves (dB/1000 RPM), slope times one-half rated RPM (giving ΔL from 50 to 100 percent RPM), noise differences above no-load for partial and full throttle at one RPM (corresponding to the EPA test mode), and the values of acceleration at that RPM.

Figures 14 through 18 are useful in that they define the noise emissions of vehicles as functions of throttle and RPM. They do not provide insight to methods for quieting, however, as they do not identify the sources of noise. Similar charts are needed for source components.

5.2 Noise Source Components

Noise source components were isolated through a sequence of removal and wrapping. As each component was removed or wrapped, its noise level was determined from the difference between before and after levels, measured at the 7.5-meter (25-foot) microphone position. To ensure meaningful differences, components were eliminated beginning with the noisiest. The following qualitative means were used for preliminary rank ordering of sources:

- Direct subjective judgement. Overall, this was very unreliable. Observers inside the vehicle, for example, always felt that exhaust noise was significant.

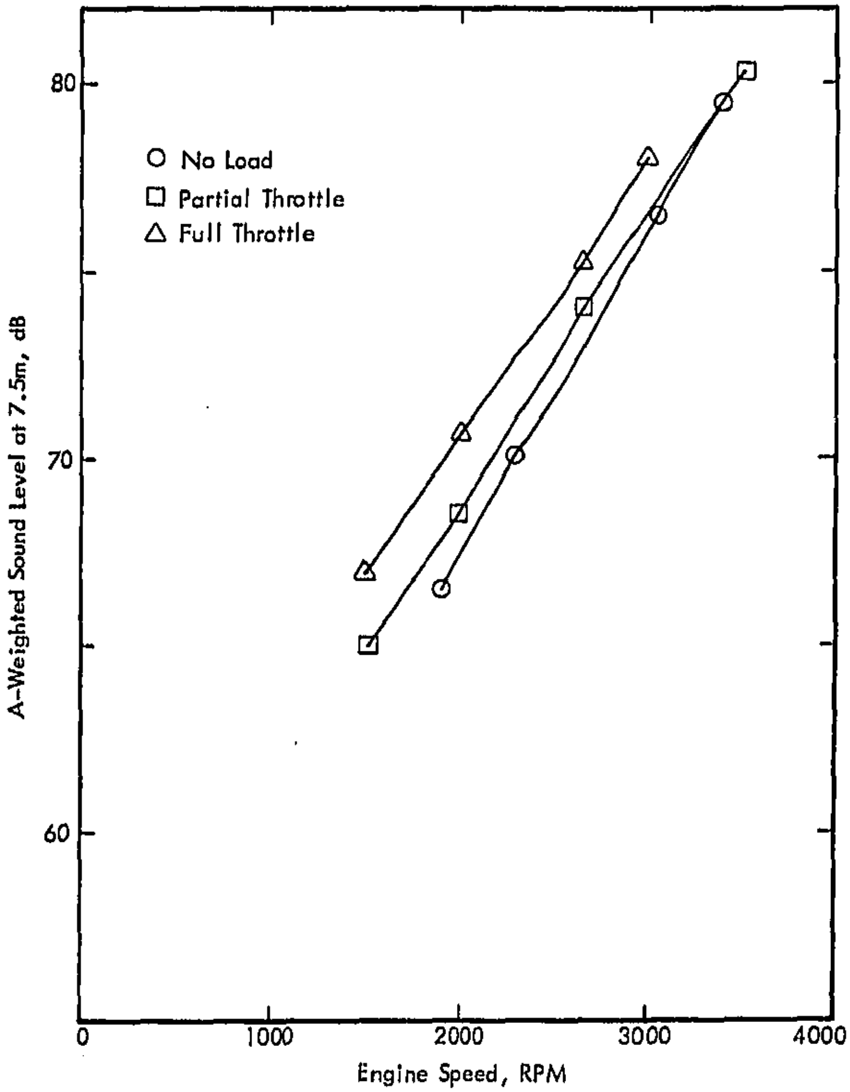

Figure 14. Sideline Noise Characteristics of Ford F-150.

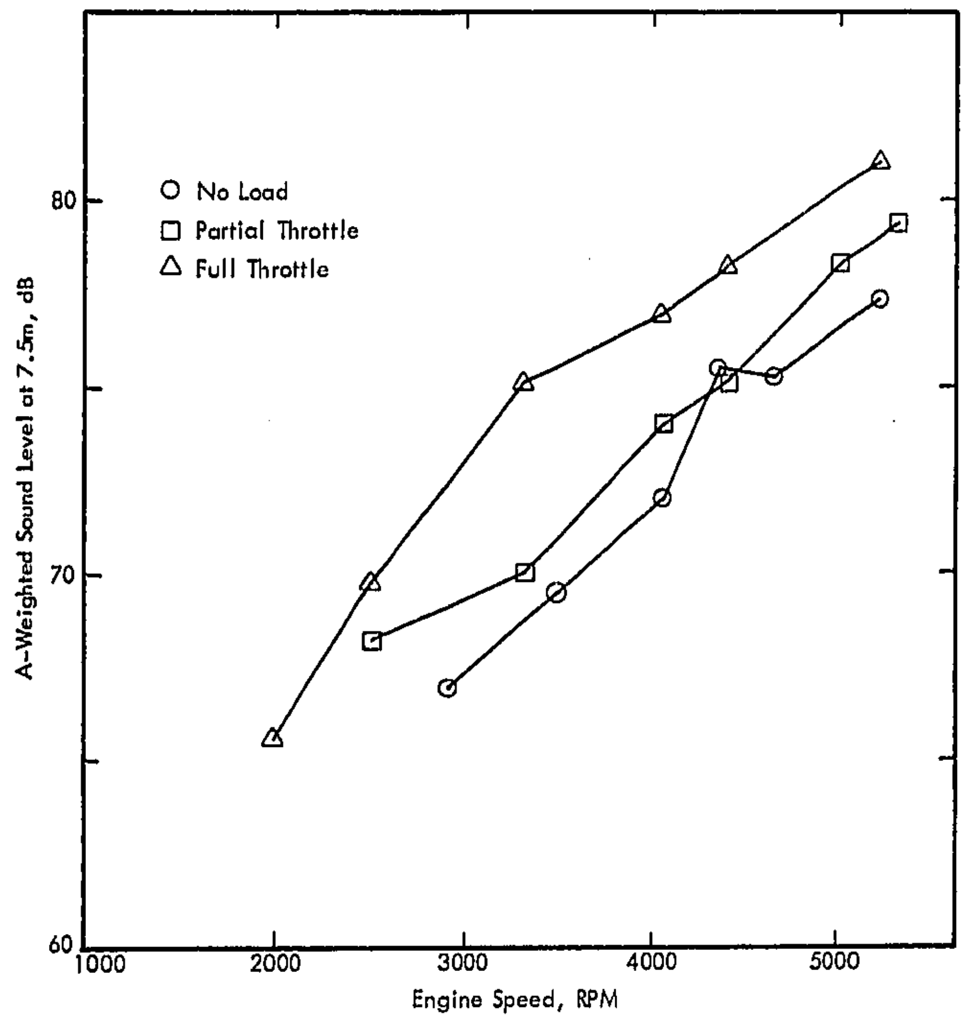

Figure 15. Sideline Noise Characteristics of Toyota Corolla.

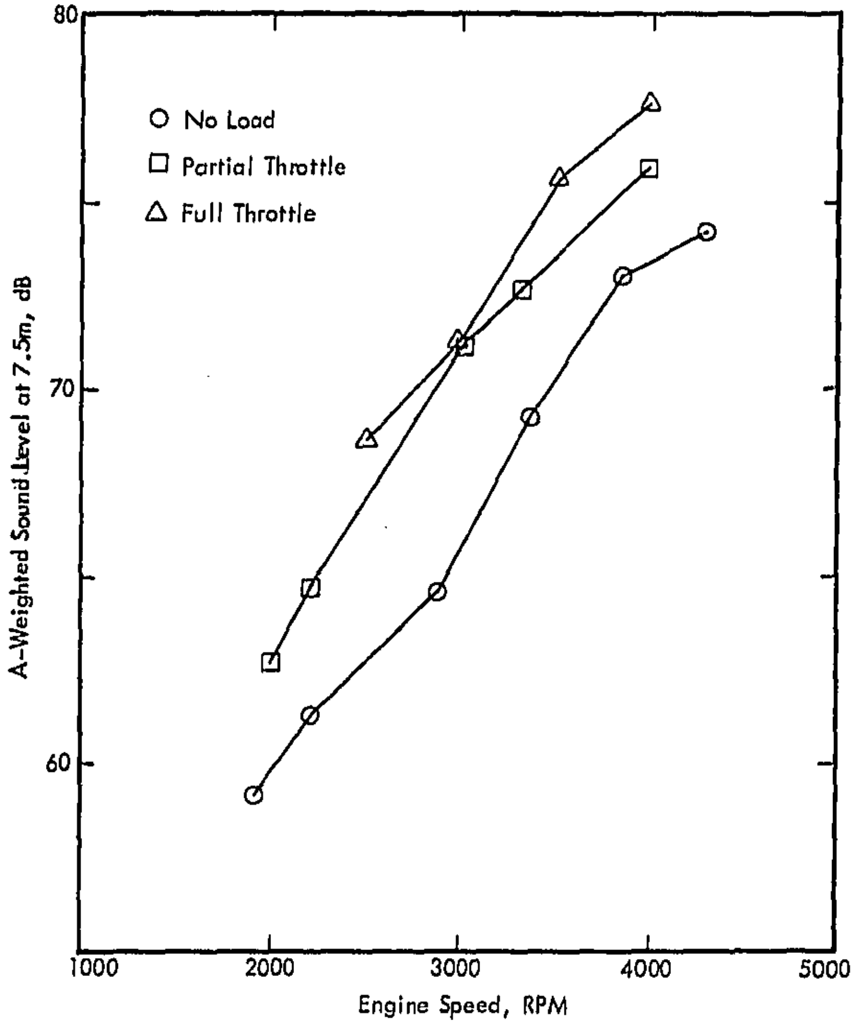

Figure 16. Sideline Noise Characteristics of Chevrolet Chevette.

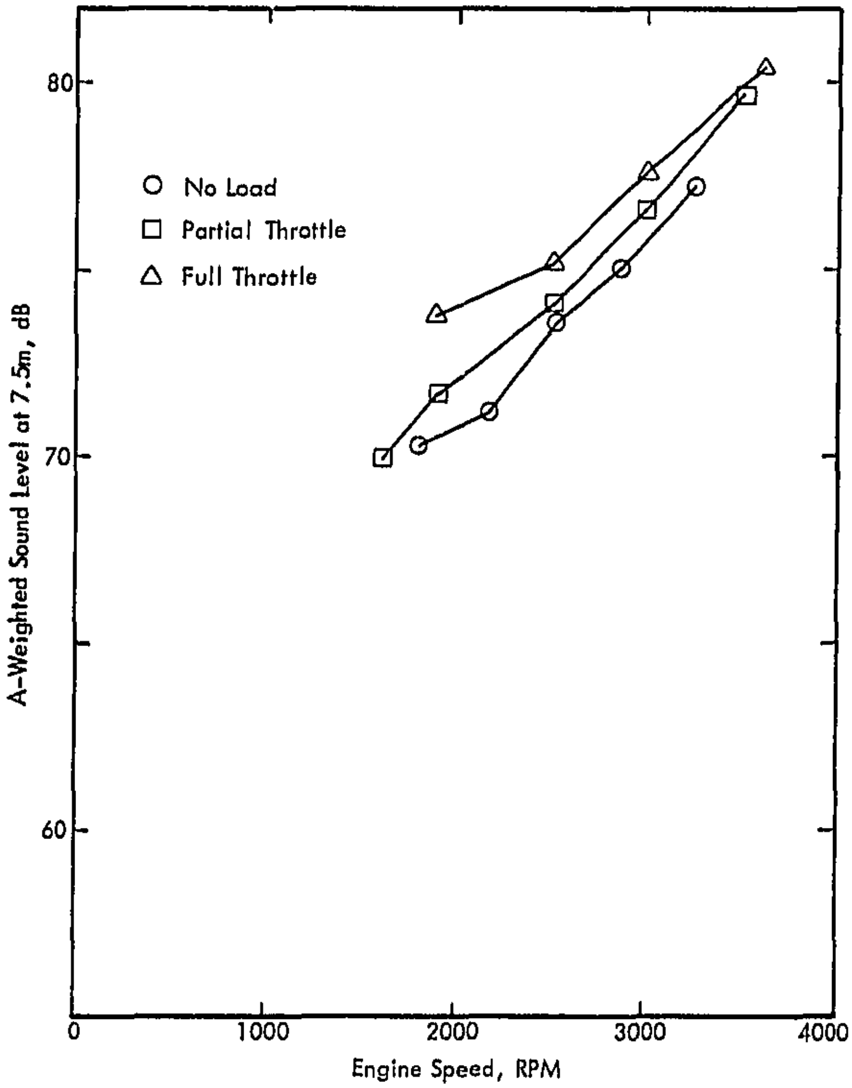

Figure 17. Sideline Noise Characteristics of Oldsmobile Diesel.

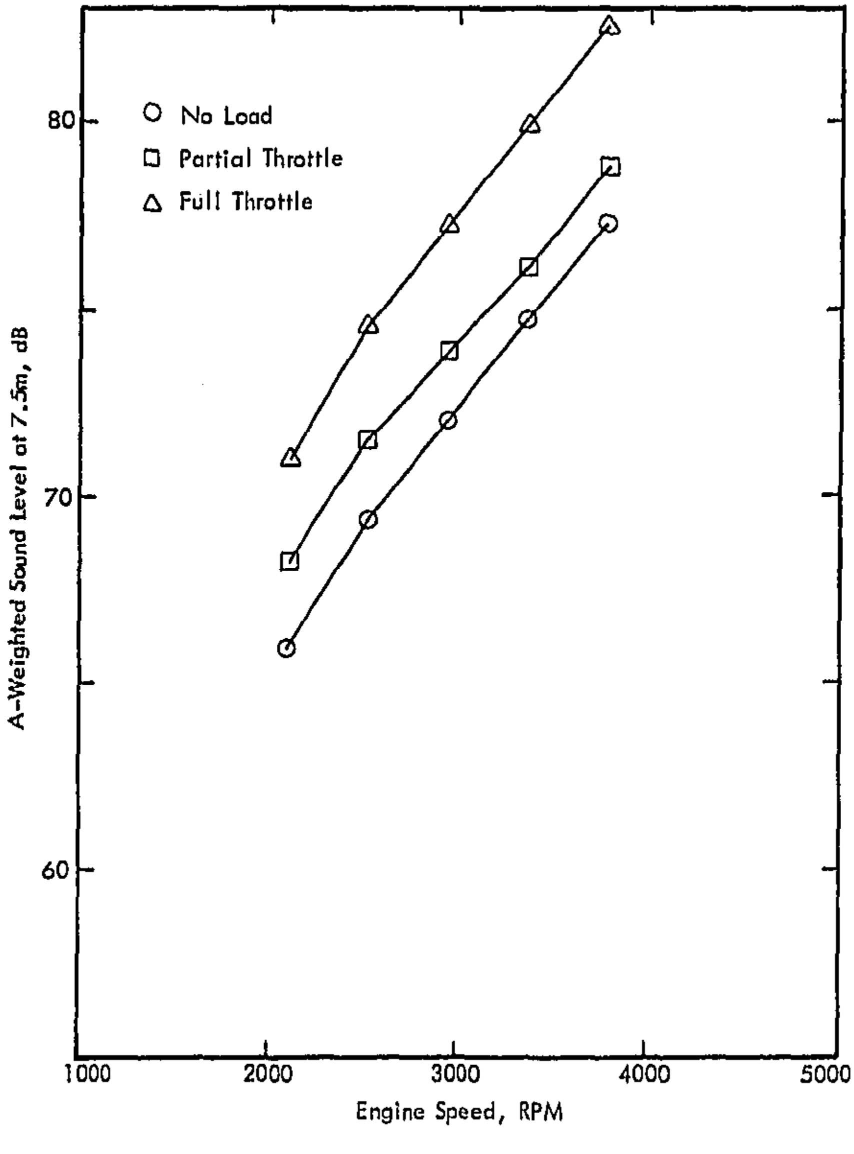

Figure 18. Sideline Noise Characteristics of Ford Pinto.

Table 6. Summary of Noise Characteristics

Vehicle	Slope dB/1000 RPM	Difference from Half to Full RPM, dB	EPA RPM*	Difference Above No Load, dB		Acceleration, g	
				Partial	Full	Partial*	Full
Ford F-150	8.0	15.2	2660	1.0	2.2	0.14	0.38
Toyota Corolla	4.2	12.1	4060	2.0	5.0	0.16	0.32
Chevrolet Chevette	6.5	15.6	3000	5.5	5.5	0.09?	0.32
Oldsmobile Diesel	4.9	8.8	2500	0.5	1.6	0.15	0.25
Ford Pinto	6.4	13.5	2520	2.2	5.2	0.21	0.26

*The part throttle tests were set up following the test procedure defined in Reference 1. The Chevette and Pinto, both equipped with automatic transmissions, shifted at significantly different speeds here than they did in the moving tests. These were the same vehicles, but had seen several months of rental service. For consistency with data in Reference 2, data here are presented for the RPM obtained in the moving tests. The corresponding acceleration rates in this table therefore do not correspond to 0.15g.

Subjective judgment was found to be useful only when a direct comparison of separate sources was involved. For example, an observer under the vehicle near the exhaust pipe could tell whether the section of pipe near him was significant compared to an adjacent section. Directional perception, as opposed to recognizing particular sounds, was the discriminatory mechanism.

- Near-field measurements. A microphone was held approximately 8 centimeters (3-1/4 inches) from the surface of various components. Two observations were made:

  - The level was noted.
  - Listening through a set of headphones, sound from the near-field microphone and the 7.5-meter microphone were compared.

  The magnitude of the sound level, relative to other sources, generally indicated whether a source was a major contributor. Listening provided additional confirmation, in that the near-field of a dominant subsource usually sounded very similar to the far field. Listening was not effective for exhaust systems because all parts tended to sound the same. When evaluating the measured levels, subjective consideration must be given to size. In one case, an alternator with near-field levels 20 dB higher than the engine block was only 3 dB louder in the far-field because of the difference in surface area. Near-field levels were most effective as a screening tool to rule out the potential importance of small components.

- Exhaust outlet measurements. A microphone placed 0.5 meter (20 inches) from the exhaust outlet was found to give an excellent measure of this component. If exhaust outlet noise is significant, this position is dominated by outlet noise, and is also in the acoustic far field. Exhaust outlet data extrapolated to 7.5 meters by 6 dB per doubling usually agreed well with that obtained by differential measurements with and without the duct and muffler (see Figure 11) installed.

The above techniques were found to be very effective in planning the ordering of the configuration matrix for source evaluation. Except for the exhaust outlet measurement, it must be emphasized that only qualitative results were obtained. In cases where there was some doubt as to what the order of attack should be, the sequence was reversed while restoring the vehicle to its original configuration.

Figures 19 through 33 present the component noise source levels for the five vehicles. For each vehicle, three figures are presented: component levels at no load, partial throttle, and full throttle. The total curve on each corresponds to those in Figures 14 through 18. The component noise source characteristics of each of the five vehicles, together with approaches for noise control, are discussed below.

5.2.1 Ford F-150

At no load and partial throttle, the engine cooling fan was the biggest subsource. This was a direct drive fan. The alternator was also found to be a major noise source, comparable to the engine at no load. The major noise source on the alternator appeared to be its cooling fan, which was a stamped centrifugal impeller.

At higher throttle settings, the exhaust system and engine contributions increased, becoming the two major sources at full throttle. The bulk of exhaust noise was from the tailpipe, but the shell radiated a significant amount of sound at low RPM and full throttle.

This vehicle clearly exhibits a shift in source components with operating mode. Note that noise control measures designed at full throttle, where engine and exhaust dominate, or at no load, where fan and alternator dominate, would not be effective at other throttle settings. All four of these are of comparable magnitude at the partial throttle condition.

The first stage of noise reduction for this vehicle is replacement or modification of noisy accessories. Substitution of a clutch fan virtually eliminated fan noise; this is discussed in Section 6.1. Some kind of modification would be required to the alternator. No alternator modifications were attempted, but the impeller was obviously not designed with consideration for low noise. Simply slowing the alternator pulley ratio would eliminate it as a significant source.

The exhaust system had a single muffler, and there was substantial room for larger and/or additional components. Providing more volume in the muffler system, as discussed

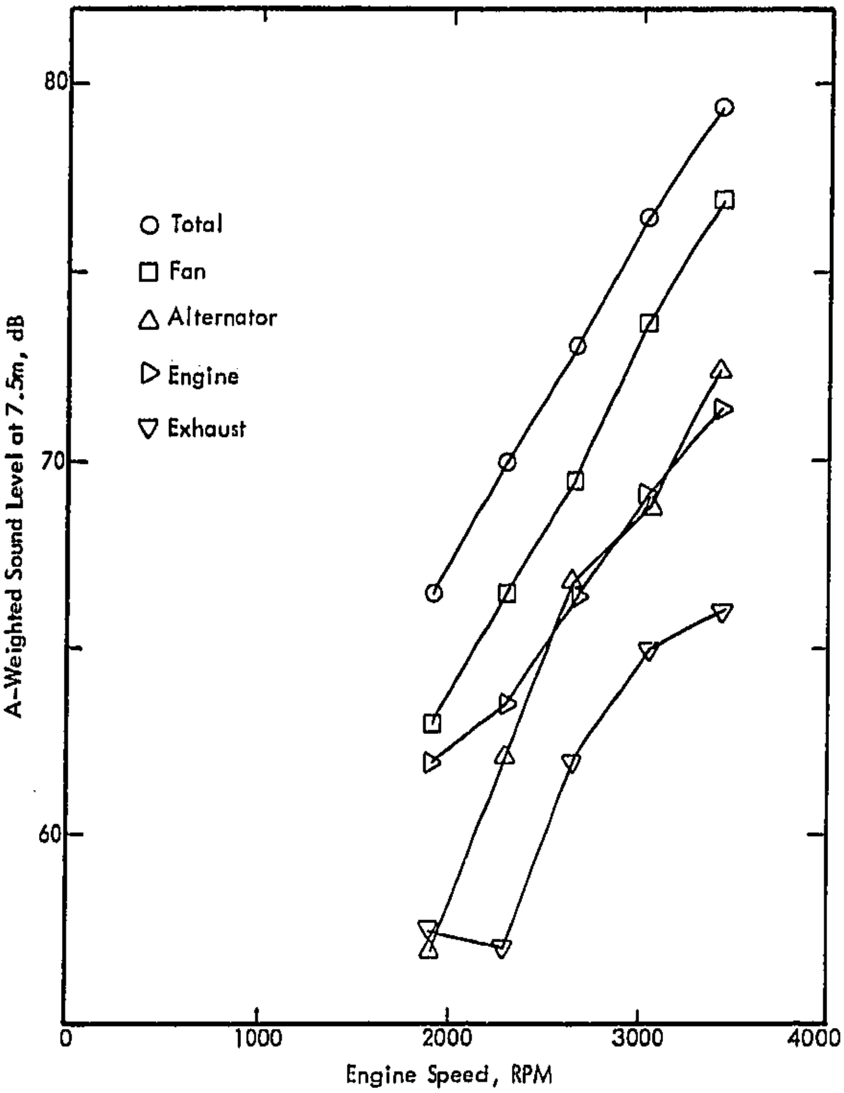

Figure 19. Noise Source Components at No Load, Ford F-150.

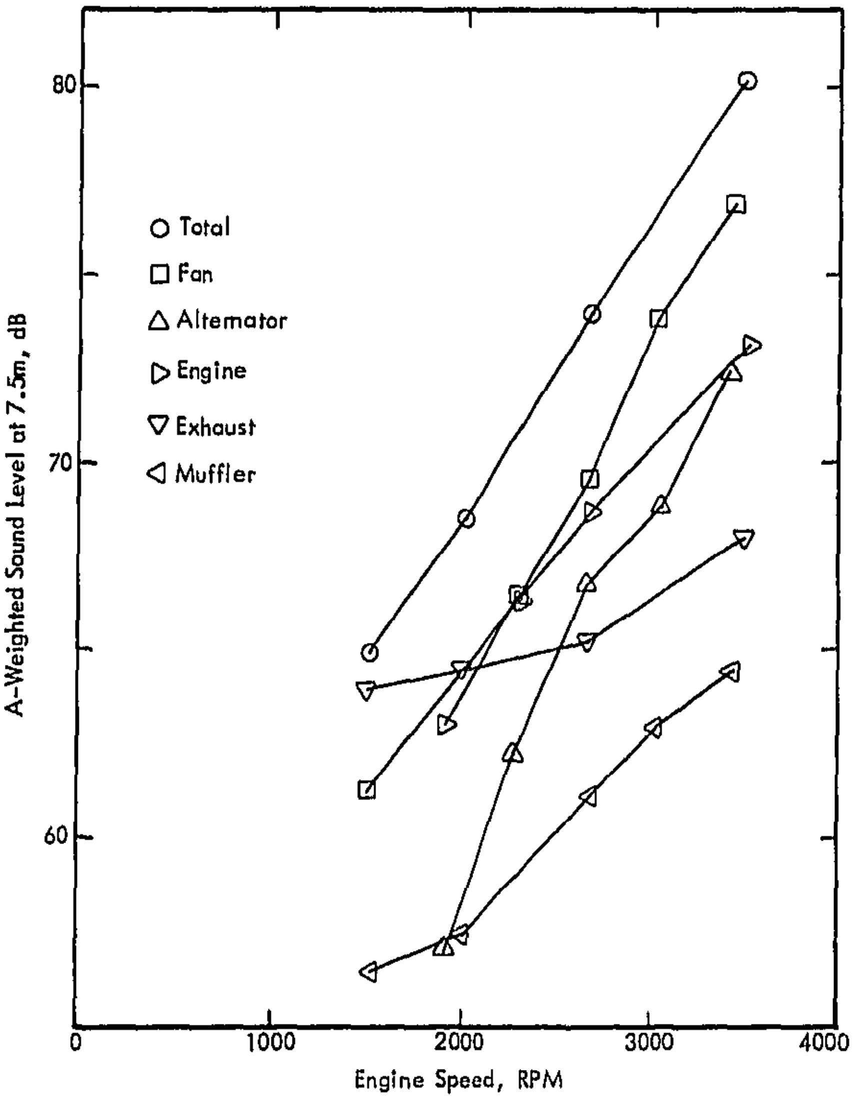

Figure 20. Noise Source Components at Partial Throttle Ford F-150.

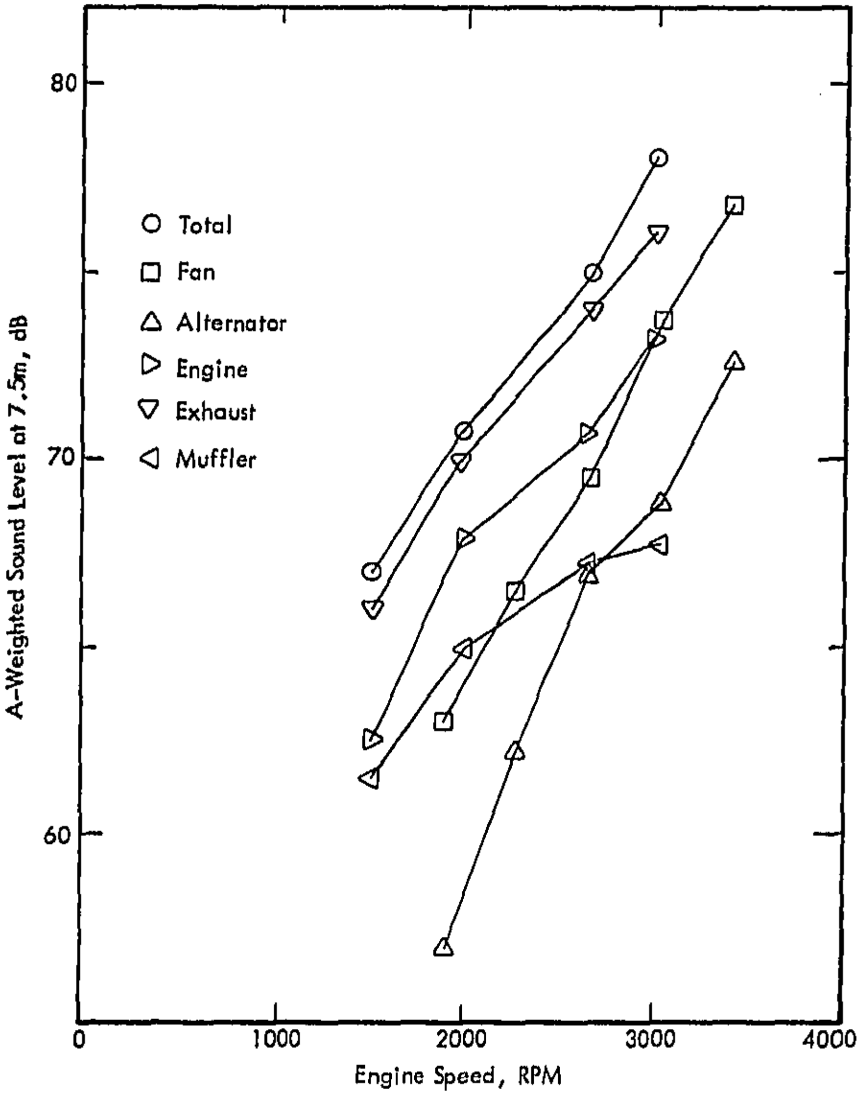

Figure 21. Noise Source Components at Full Throttle, Ford F-150.

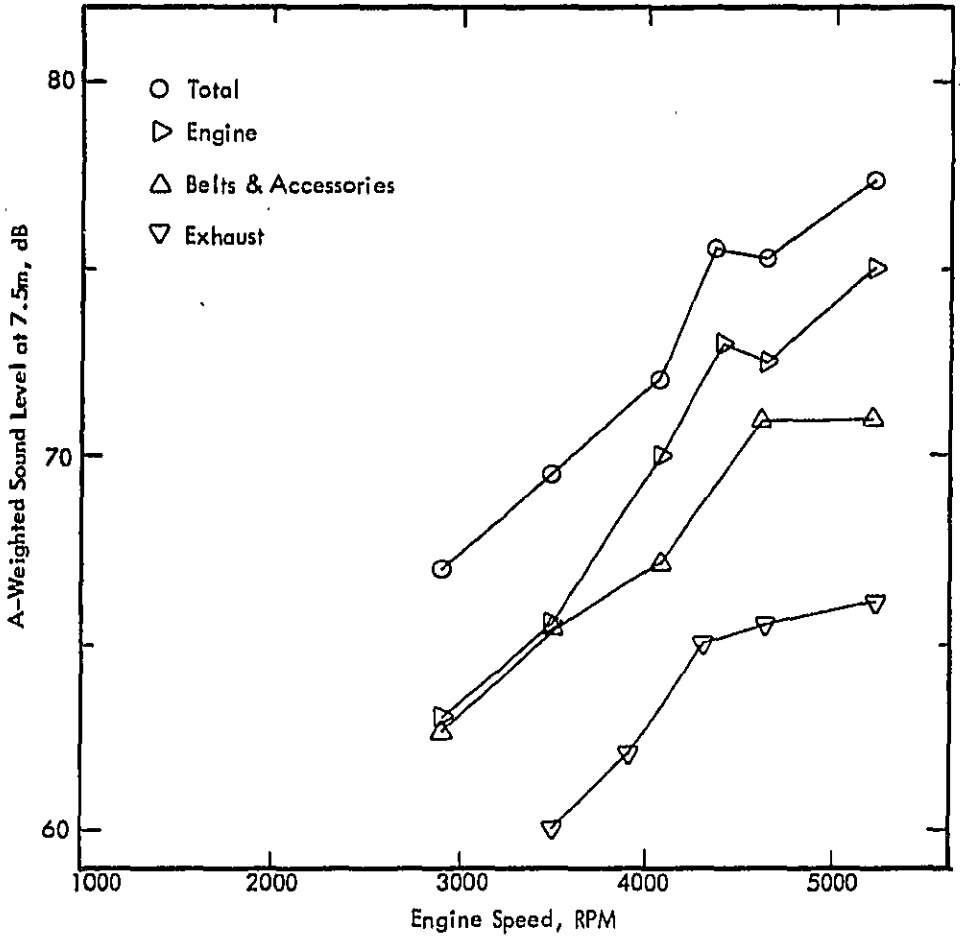

Figure 22. Noise Source Components at No Load, Toyota Corolla.

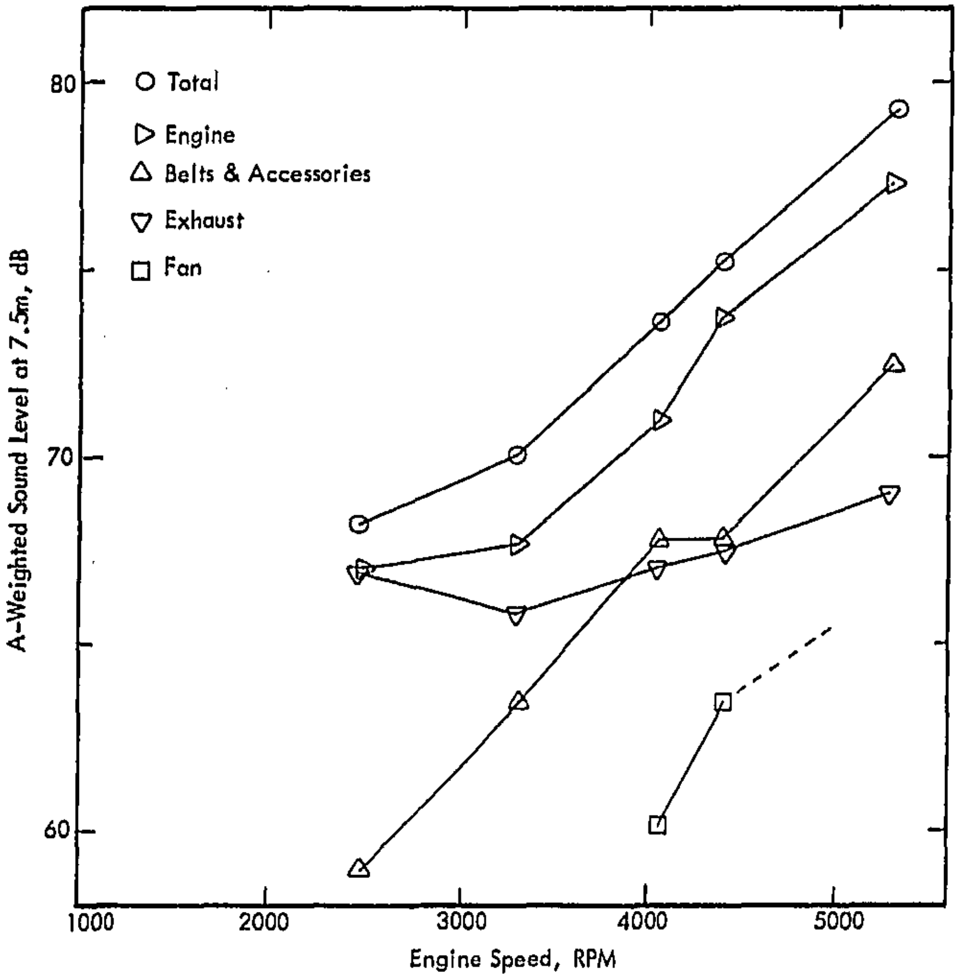

Figure 23. Noise Sources at Partial Throttle, Toyota Corolla.

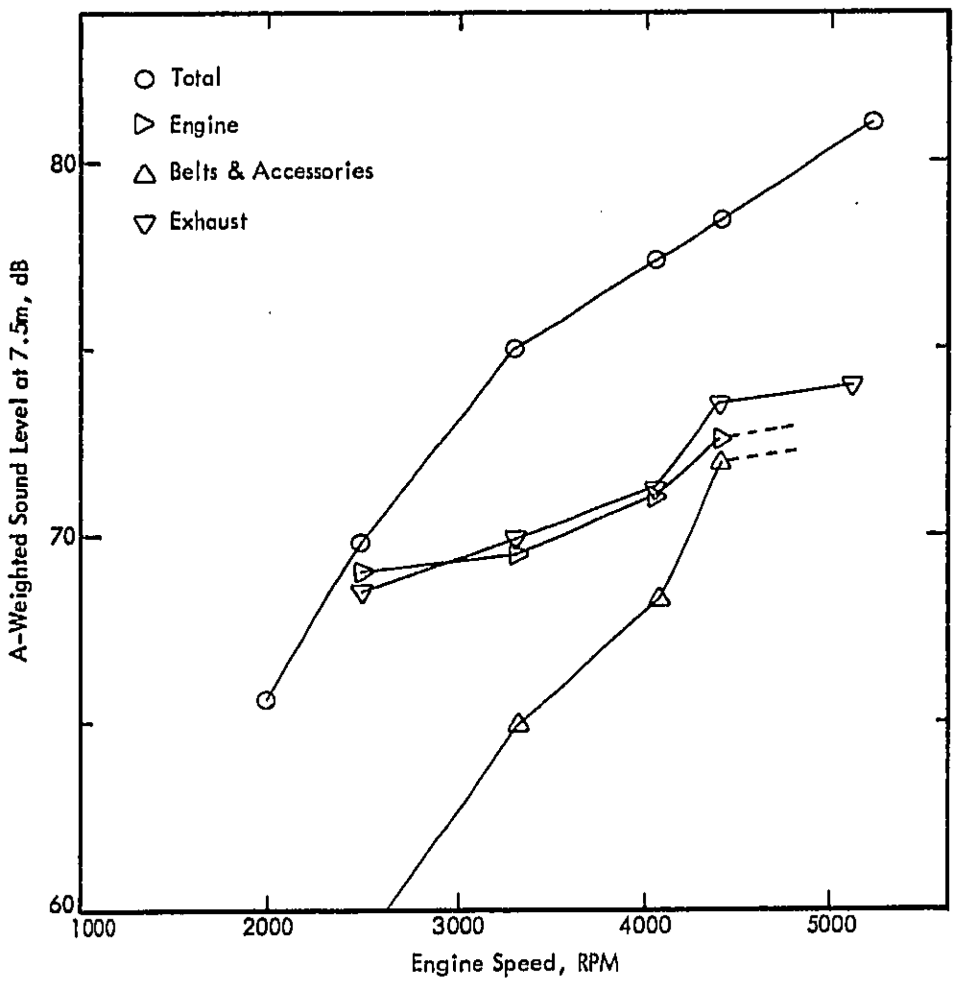

Figure 24. Noise Sources at Full Throttle, Toyota Corolla.

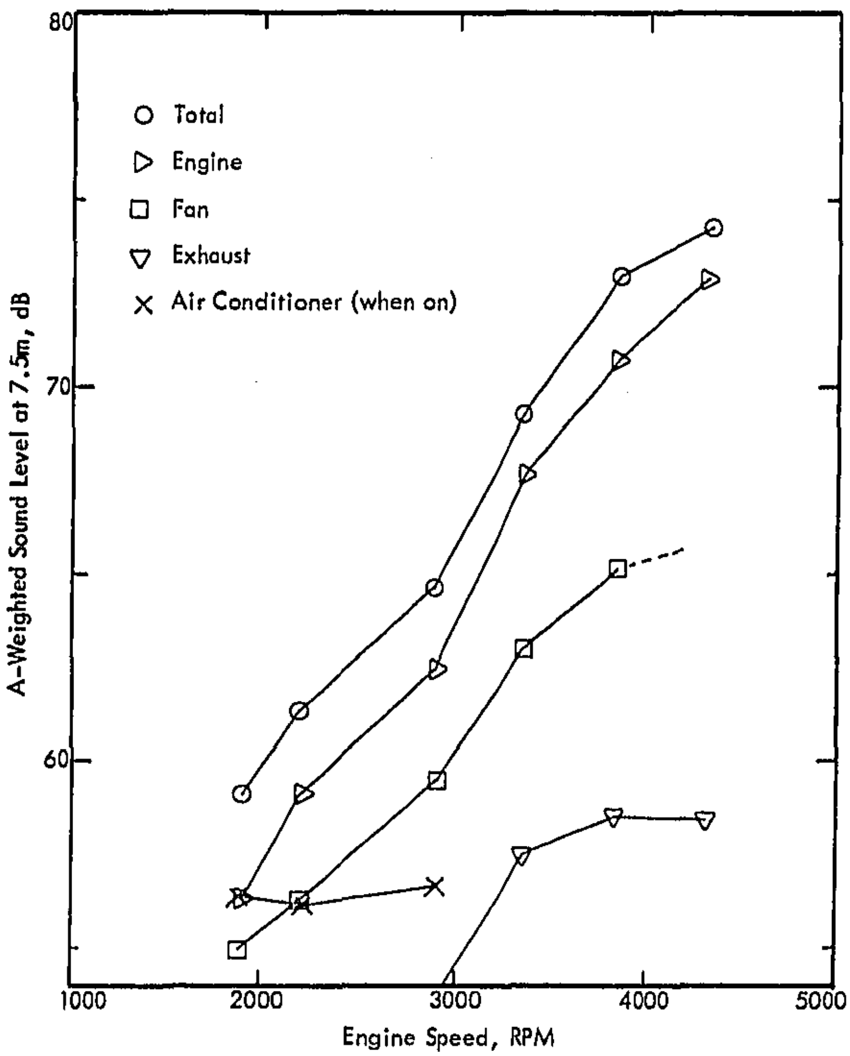

Figure 25. Noise Source Components at No Load, Chevrolet Chevette.

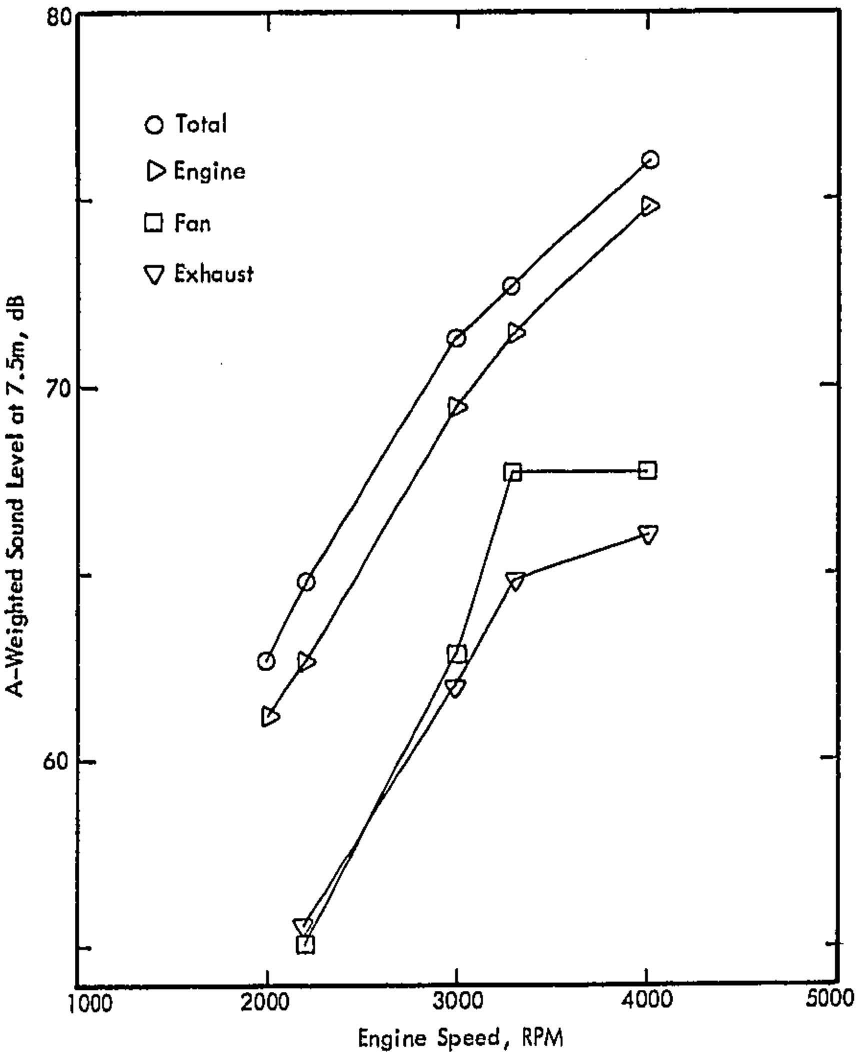

Figure 26. Noise Source Components at Partial Throttle, Chevrolet Chevette.

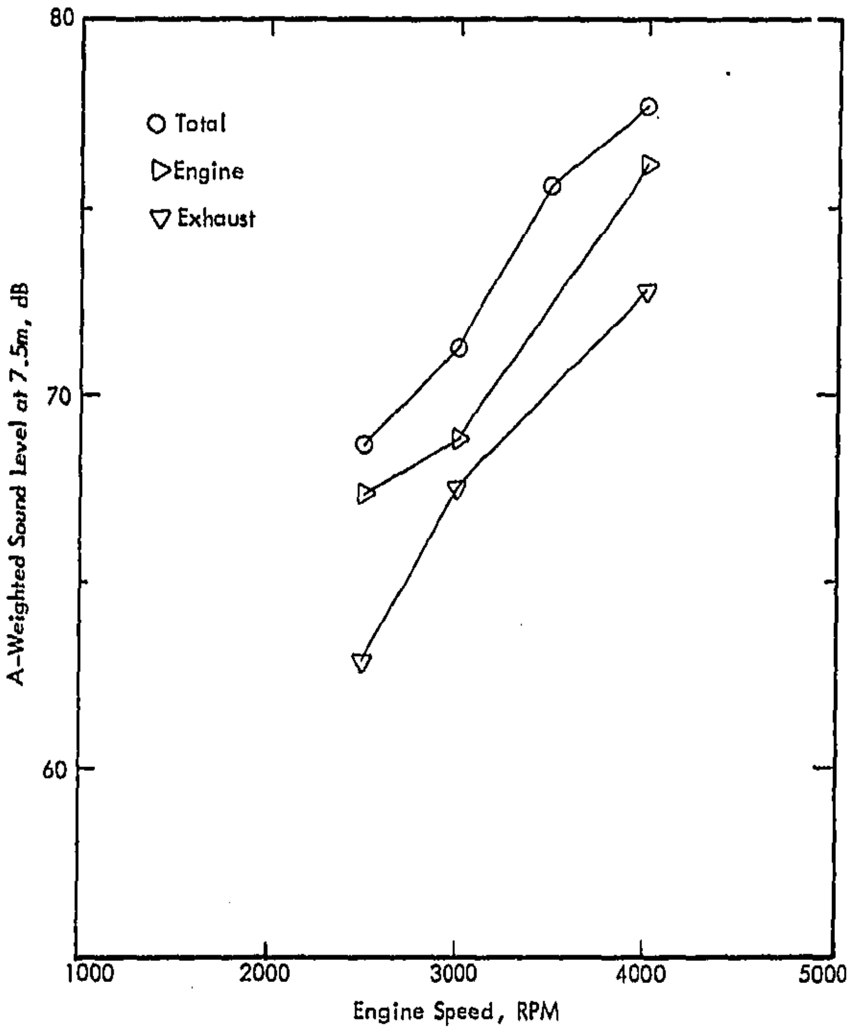

Figure 27. Noise Source Components at Full Throttle, Chevrolet Chevette.

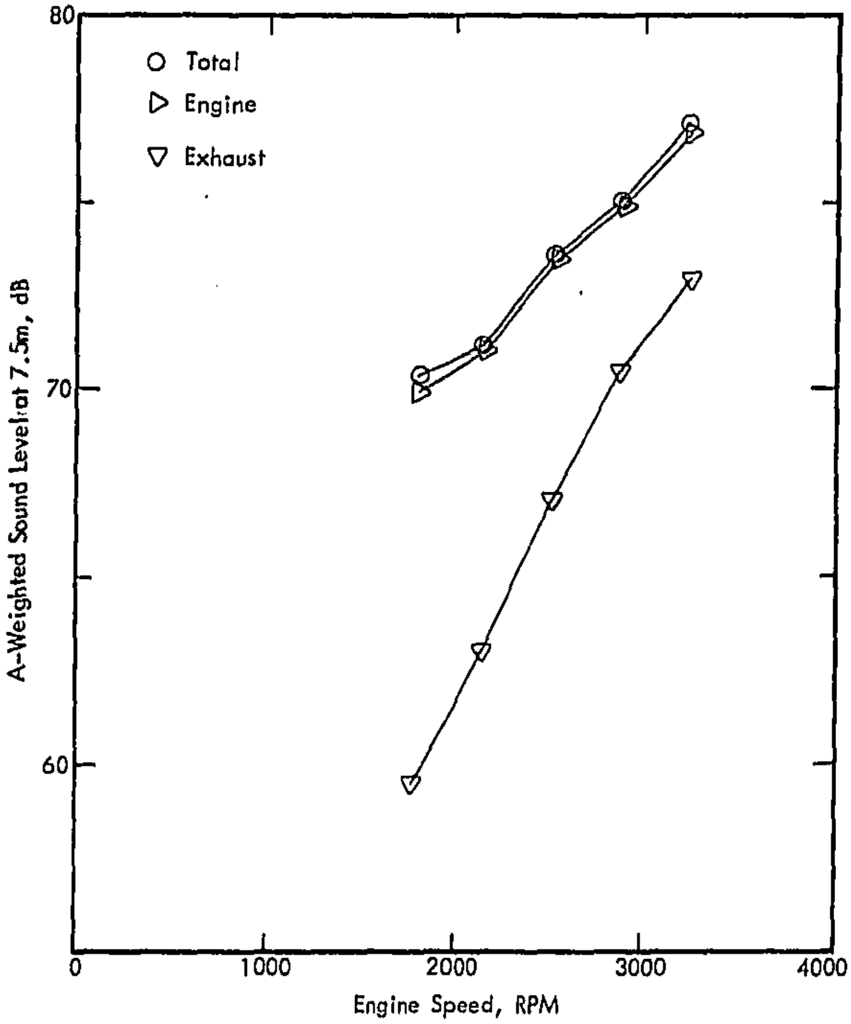

Figure 28. Noise Source Characteristics at No Load, Oldsmobile Diesel.

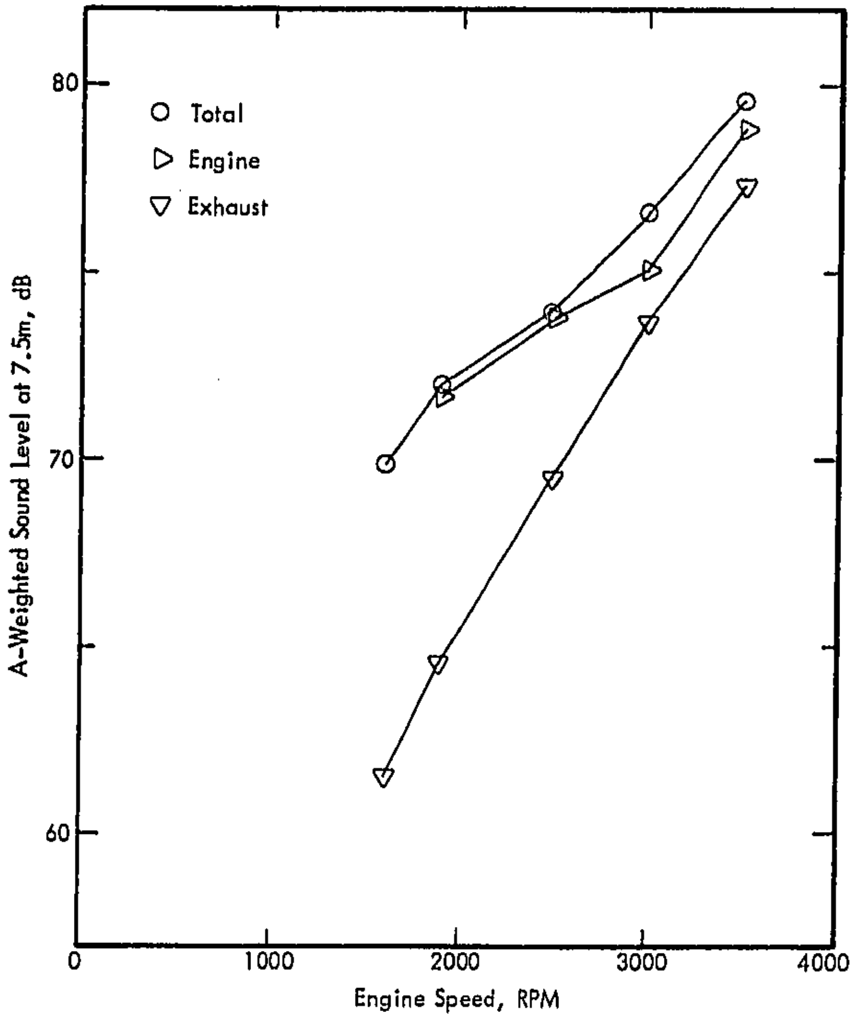

Figure 29. Noise Source Characteristics at Partial Throttle, Oldsmobile Diesel.

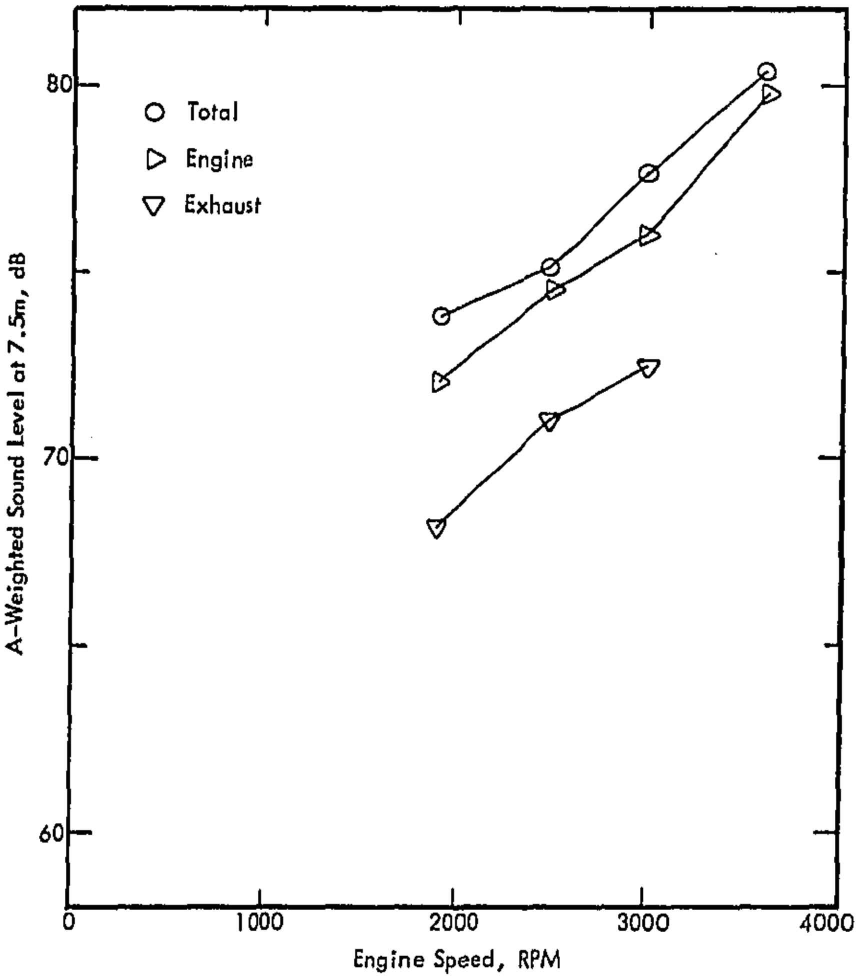

Figure 30. Noise Source Characteristics at Full Throttle, Oldsmobile Diesel.

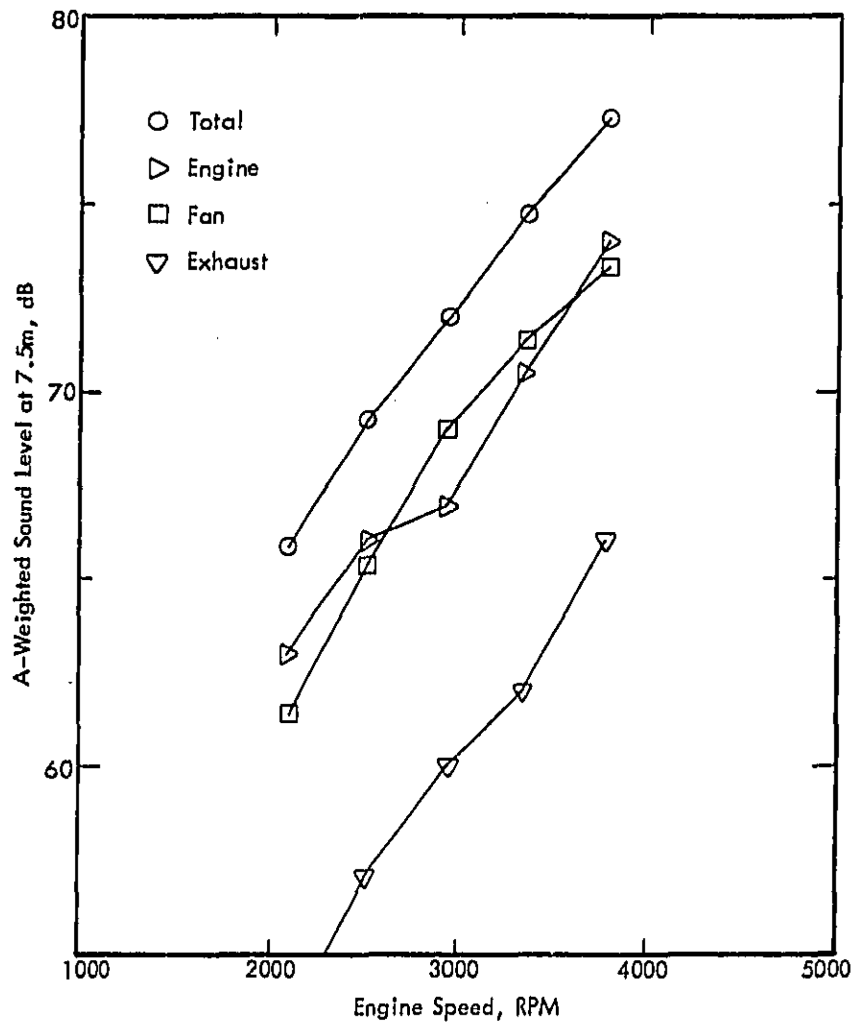

Figure 31. Noise Source Contributions at No Load, Ford Pinto.

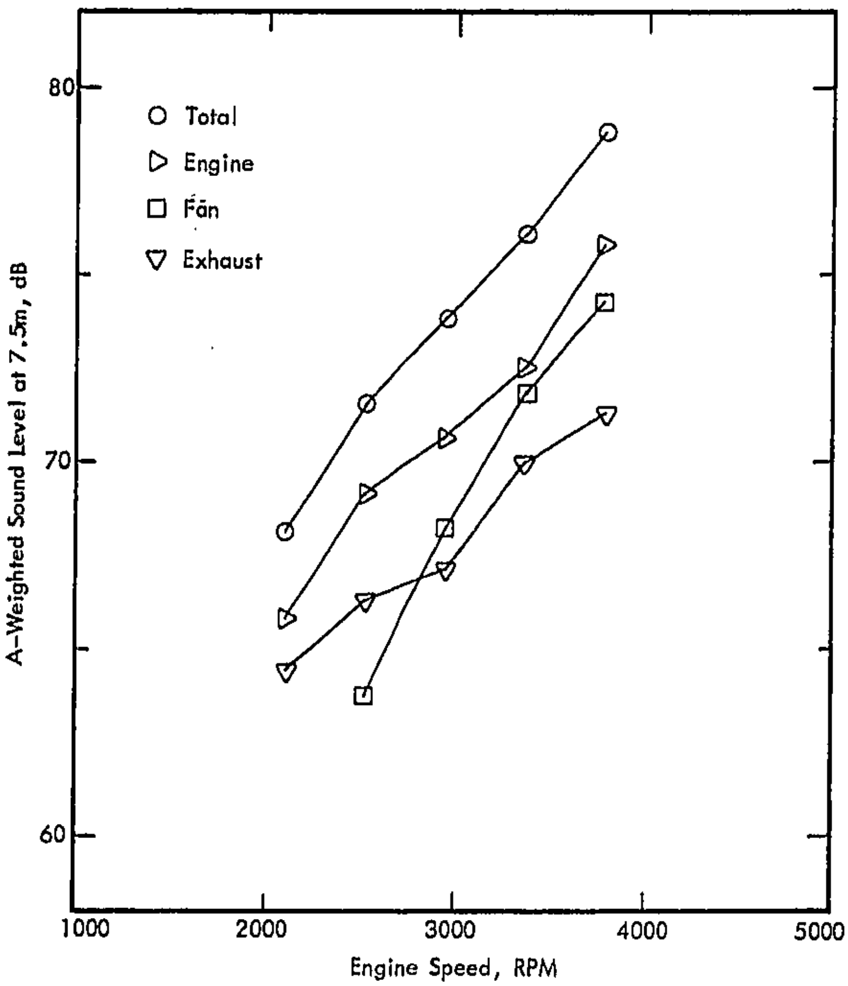

Figure 32. Noise Source Contributions at Partial Throttle, Ford Pinto.

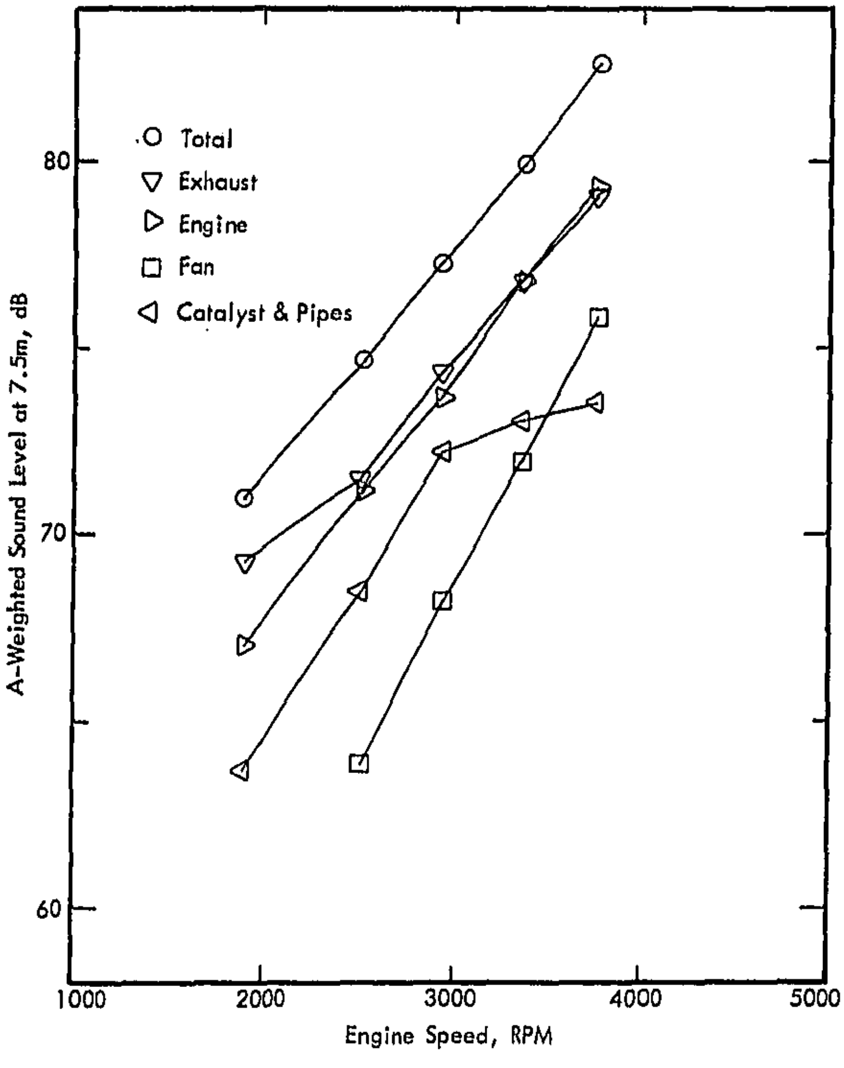

Figure 33. Noise Source Contributions at Full Throttle, Ford Pinto.

in Section 6.2, would pose no secondary difficulties. This would reduce the dominant exhaust noise at full throttle.

A second stage of noise reduction would be to reduce engine noise. Assuming the engine itself cannot be modified, an enclosure of some type would be required. This is discussed in Section 6.3. After this stage, consideration would have to be given to radiation from the muffler shell if the original design were still used in the modified exhaust system.

5.2.2 Toyota Corolla

The dominant noise source on this vehicle was the engine. Accessories and their drive belts were identified to be major sources as well, by comparing levels with the drive belts disconnected. This may not be an accurate portrayal, however. Mechanical resonances played a large role in the noise characteristics of this vehicle – note the peak at around 4400 RPM – and it is equally possible that removing the belts altered the vibration response as a whole. Qualitative and near-field observations did not identify a particular source, as they did for the Ford F-150.

The cooling fan – which had a viscous clutch – was found not to contribute measurably to noise.

Noise reduction to this vehicle, in the absence of fundamental reduction to engine/accessory noise, would require an engine compartment enclosure. Depending on the extent of reduction here, exhaust noise may be a significant secondary source. Engine enclosures and exhaust modifications were tested for this vehicle, and are discussed in Section 6.

5.2.3 Chevrolet Chevette

The engine was the dominant noise source at no load and partial throttle, with exhaust providing a significant contribution at full throttle. Fan noise is also a significant secondary source at lower throttle settings. It is difficult to quantify it precisely because of the variable nature of the clutch drive. At full throttle, engine noise was too high to permit measurement of fan noise.

Any noise reduction of this vehicle would require an engine enclosure. This would quiet both the engine and the fan. Depending on the degree of noise reduction required at full throttle, additional silencing of the exhaust would be the next step. This could be difficult, as there appeared to be little room for larger components. At full throttle, however, exhaust noise is only about one-third of the total; it is less at lower throttle settings.

5.2.4 Oldsmobile Diesel

The dominant noise source on this vehicle at all operating conditions was the engine. The exhaust noise component levels shown in Figures 28 and 29 for no load and partial throttle were obtained from near-field exhaust measurements, and may be somewhat higher than actual; the exhaust was on the right side and the 7.5-meter microphone was on the left.

None of the accessories (including the high-pressure fuel injection pump) were found to contribute measurably to the noise. A qualitative examination, as described earlier, revealed no obvious subsources other than the engine itself. Near-field noise measurements around various engine components and accessories all fell within a 4 dB range. The air inlet – often a major source on Diesels – was examined and found to be well treated, including a small silencer on the inlet duct to the filter plenum.

As a final diagnosis for sources, acceleration measurements were made on the valve covers, engine block, oil pan, and intake manifold. A-weighted vibration levels were generally within 2 to 3 dB of each other for these areas. In the absence of obvious vibration differences, no definite conclusion can be reached without actually isolating components. A subjective assessment was made by listening to the accelerometer signal and comparing it to the sound at 7.5 meters. The block vibrations sounded exactly like the 7.5-meter acoustic signal; the other components much less so, especially with regard to the characteristic Diesel knocking sound.

Recognizing the qualitative nature of this examination, it has been concluded that the basic noise source of the Diesel engine is block-related vibration. The manufacturer has obviously paid careful attention to minimizing secondary or external noise sources such as resonating sheet metal covers and inlet noise. Further noise reduction on this vehicle, in the absence of fundamental changes to the engine, would require an engine compartment enclosure.

5.2.5 Ford Pinto V6

The engine was the major noise source over the range of operating conditions. At low throttle, the fan was a comparable noise source. The vehicle had a direct drive fan. Exhaust noise became a significant source at full throttle, with radiation from the pipes and catalytic converter an important secondary source. The increases in level with throttle for the engine and exhaust system are consistent with that observed for the other vehicles.

Noise reduction at low power would require quieting the fan, most easily accomplished with a clutch fan. At high throttle, the exhaust system would need improvement. At all throttle settings, the engine contributes about half the noise, so some type of enclosure would be required to achieve a total noise reduction of more than 3 dB. Quantitative noise control modifications are discussed in Section 6.

5.3 Noise at EPA Test Condition

As seen from the above data discussion, vehicle noise and its subsources vary considerably with operating mode. The variation of the relative importance of the sources is highly significant, since noise control on one source does not give uniform reduction over all modes.

In order to simplify the noise reduction analysis of these five vehicles, it is desirable to limit the discussion to a single operating mode. The appropriate mode to use is that corresponding to the test procedure developed for EPA in Reference 1. In addition to considerations of consistency with References 1 and 2, this corresponds to a median in the operating ranges discussed above. Examining Figures 19 through 33, sources which dominate at full or partial throttle tend to be at least secondary sources at this partial throttle condition. This condition therefore has the benefit of providing a reasonable compromise with regard to being "typical" for designing noise reduction which will be effective over a wider range.

Table 7 summarizes the overall and component source levels at 7.5 meters for this mode for the five test vehicles.

Table 7. Total Noise and Component Contributions at EPA Test Conditions. L_A at 7.5 Meters, Left Side of Vehicle, dB

	Ford F-150		Toyota Corolla		Chevette		Oldsmobile		Ford Pinto	
RPM	2660		4060		3000		2500		2520	
Moving Test	75.3		73.6		72.2		74.8		73.2	
Dynamometer	74.0		73.6		71.2		74.0		71.5	
Components	Level	%	Level	%	Level	%	Level	%	Level	%
Engine	68.6	29	70.9	54	69.4	66	73.8	95	69.1	58
Fan	69.5	35	60.1	4	62.9	15	--		63.8	17
Alternator	66.7	19	--		--		--		--	
Belts & Access.	--		67.8	26	--		--		--	
Exhaust	65.2	13	67.0	22	62.0	12	69.5*	35	66.3	30
Muffler	61.2	5	--		--		--		--	
Pipes, Catalyst	--		--		--		--		--	
TOTAL	74.0		73.9		70.9		75.2		71.7	

* Exhaust on right side. In moving test, right side was 0.7 dB louder than left.

6.0 NOISE REDUCTION

Following identification of component sources and their levels, a series of modifications was conducted on each vehicle to determine the effectiveness of various noise control measures. Non-acoustical properties, e.g., structural requirements, operating constraints, etc , were not experimentally investigated. Noise reduction techniques investigated included component substitution, absorptive treatment, and engine enclosures. To a large degree, the noise reduction study was an extension of the wrapping and removal process for source identification. Engine enclosures, for example, can be viewed as part of wrapping. Not all reduction techniques were applied to all five vehicles, especially if trends had been established from other vehicles.

The findings of this study for various quieting techniques are discussed in Section 6.1. Quiet configurations for the five test vehicles are presented in Section 6.2.

6.1 Quieting Modifications

6.1.1 Cooling Fan

Two of the test vehicles — the Ford F-150 and the Ford Pinto — had direct drive fans which were significant noise sources. The other three vehicles were equipped with theromstatic clutch fan units which either did not measurably contribute noise or were substantially quieter (by at least 3 dB) than the engine alone.

Fan noise is very highly dependent on speed, varying as 50 $\log_{10}$ RPM.[3] A 25 percent reduction in fan speed, all else being equal, results in a reduction of over 6 dB. This would also reduce cooling air flow by 25 percent, however, so that fan noise reduction is not usually as simple as reducing the pulley ratio.

The fan is normally required only at low speeds, where there is no ram air. The speeds at which noise is of interest are much higher than this. A direct drive fan set up to deliver adequate cooling air at idle operates much faster than is required at driving speeds. Thermostatically controlled fans, which limit fan speeds to those actually needed, can be very effective in reducing noise at higher speeds. Accordingly, the stock fans on these two vehicles were replaced with thermostatic clutch units. Figure 34 shows the fan and clutch used on the Ford F-150, together with the original unit. The clutch fan has much greater blade area to compensate for its lower speed due to some slip under all conditions.

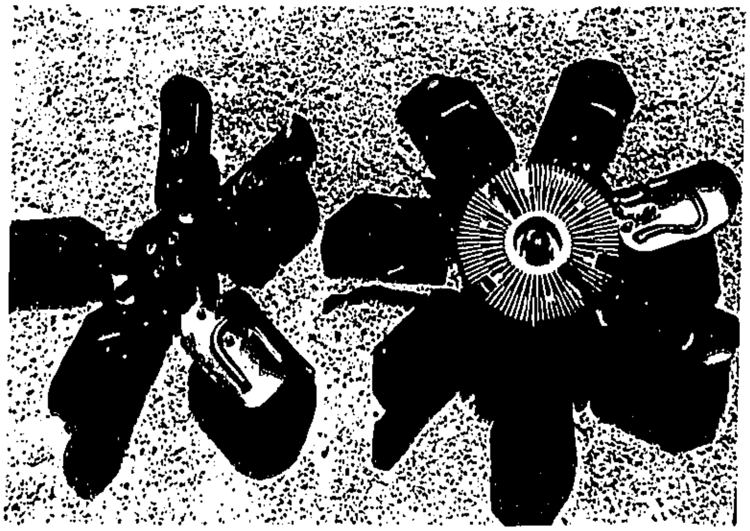

Figure 34. Original Equipment and Viscous Clutch Drive Fans, Ford F-150

With the clutch fans installed, fan noise was not measurable on these two vehicles.

When a clutch fan is installed as original equipment, the cooling system may have other differences in comparison with a vehicle with a direct fan. The most pertinent for noise is that the pulley ratio may be different. Drive speeds can be up to 50 percent higher. To assess the effect of a higher pulley ratio, the following tests were performed on the Ford F-150:

- Speed of the clutch fan was measured at various engine speeds. This is shown in Figure 35, together with the direct drive fan speed calculated from the pulley ratio.
- The clutch unit was locked and the noise from this fan measured with no slip. Figure 36 shows this in comparison with the original fan.

The measured clutch fan noise levels, clutch locked, were adjusted by the RPM curve shown in Figure 35 to obtain operating noise levels. This is shown in Figure 36 for comparison with the original fan. Also shown is the clutch fan noise level if the pulley ratio were 50 percent higher. Except for very low speeds — well below the speed for the EPA test mode — noise from the clutch fan is much lower than for the original stock fan.

Based on these findings and the source results for the vehicles with clutch fans, it is concluded that fan noise may be eliminated, except at low RPM, by the use of a thermostatic clutch fan drive.

6.1.2 Exhaust Systems

Over most of the operating range of the five test vehicles, exhaust noise was a secondary source. At high RPM and high throttle, it approached engine noise on three of the vehicles. On the Ford F-150, it was the dominant source at full throttle; on the Chevette it was never a primary source.

Exhaust system noise control has been the subject of a large amount of research and engineering. Modern exhaust systems are designed to match the characteristics of the engine. The findings described in Section 5 show that this work has generally been

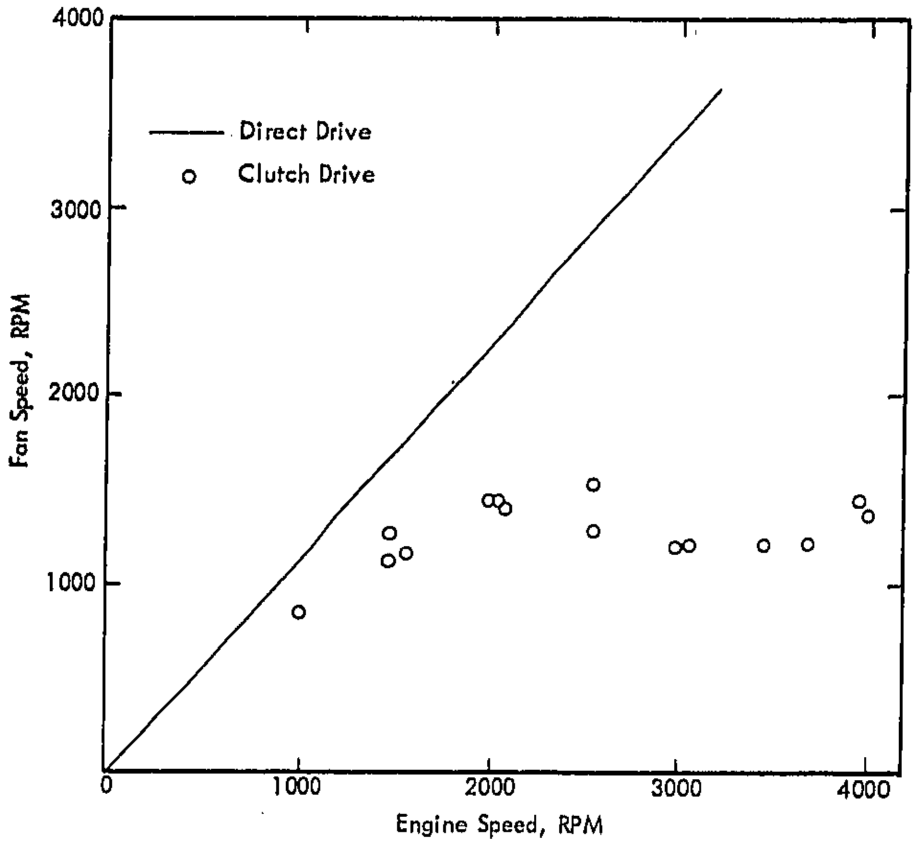

Figure 35. Speed of Clutch Fan and Direct Drive Fan, Ford F-150.

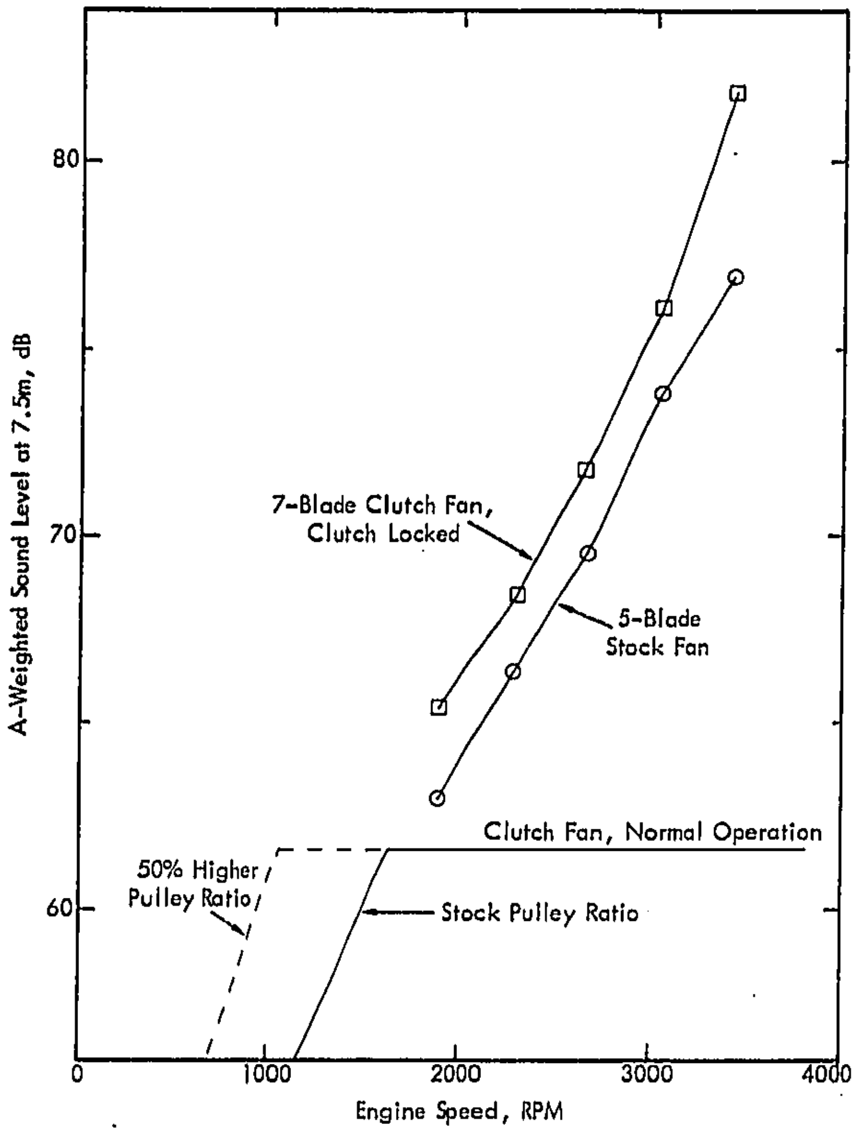

Figure 36. Noise Levels of Stock Fan and Clutch Fan, Ford F-150.

quite successful. Within the context of reducing total vehicle noise at the part-throttle EPA test condition, only moderate additional reductions of 3 to 5 dB would be required.* With reactive mufflers, this can be achieved by increasing the volume by about 50 percent.[4] Available space becomes the major constraint on feasibility.

In view of the existing body of exhaust silencing work, a detailed study of exhaust systems was beyond the scope of the present study. A demonstration pertinent to this study was that some modification could be made which met the above goal, and could be installed within available space without degrading performance. Accordingly, two modified exhaust systems were prepared:

- On the Toyota, the original rear muffler was replaced with the system shown in Figure 37. This consisted of a straight-through attenuator which replaced a section of pipe, and a new reactive muffler with larger volume than original.
- On the Oldsmobile Diesel, a second reactive muffler (nominally similar to the single original one) was added in series, replacing a section of pipe.

Neither of these modified exhaust systems received any formal design; they were based solely on available space and components.

Figures 38 and 39 compare the stock and modified exhaust systems. The Toyota's exhaust was quietened by 4 to 6 dB, and the Oldsmobile's by 3 to 5 dB, depending on speed and load.

6.1.3 Engine Enclosures

The point is very quickly reached where the dominant noise source is the engine itself. Referring to Table 7, the engine accounted for more than half the noise for four out of the five vehicles tested. The remaining vehicle, the Ford F-150, would also fall into this category after installation of a thermostatic clutch fan.

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*This is based on the practical limits of noise reduction achievable with flow-through engine enclosures. See Section 6.1.3 and 6.2.

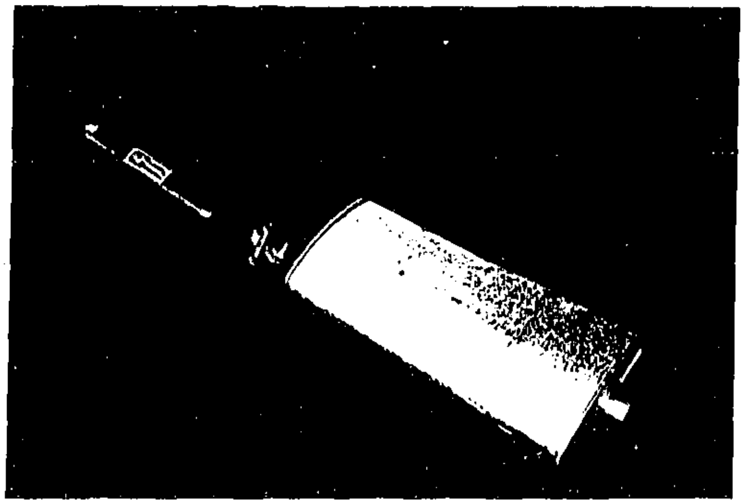

Figure 37. Modified Exhaust System, Toyota Corolla

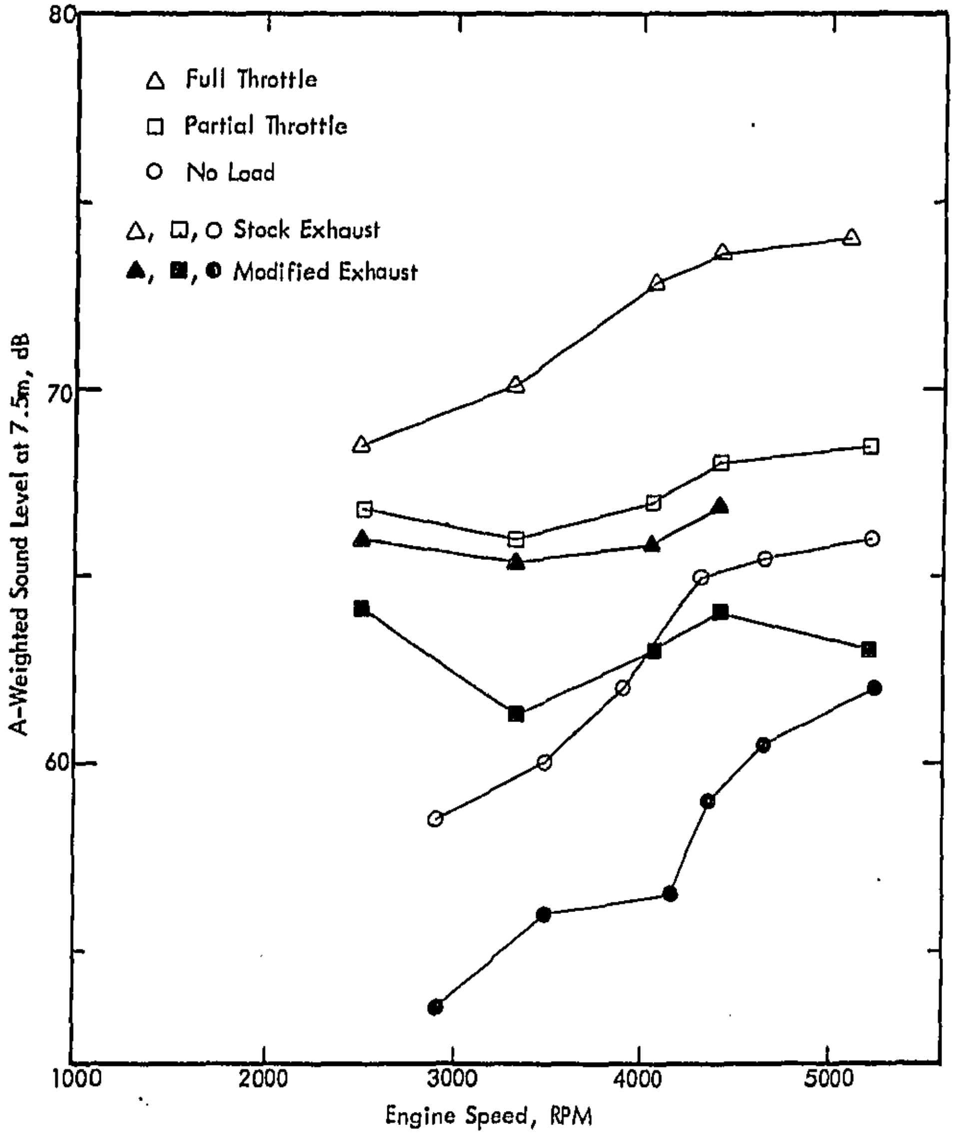

Figure 38. Comparison Between Stock and Modified Exhaust Systems, Toyota Corolla.

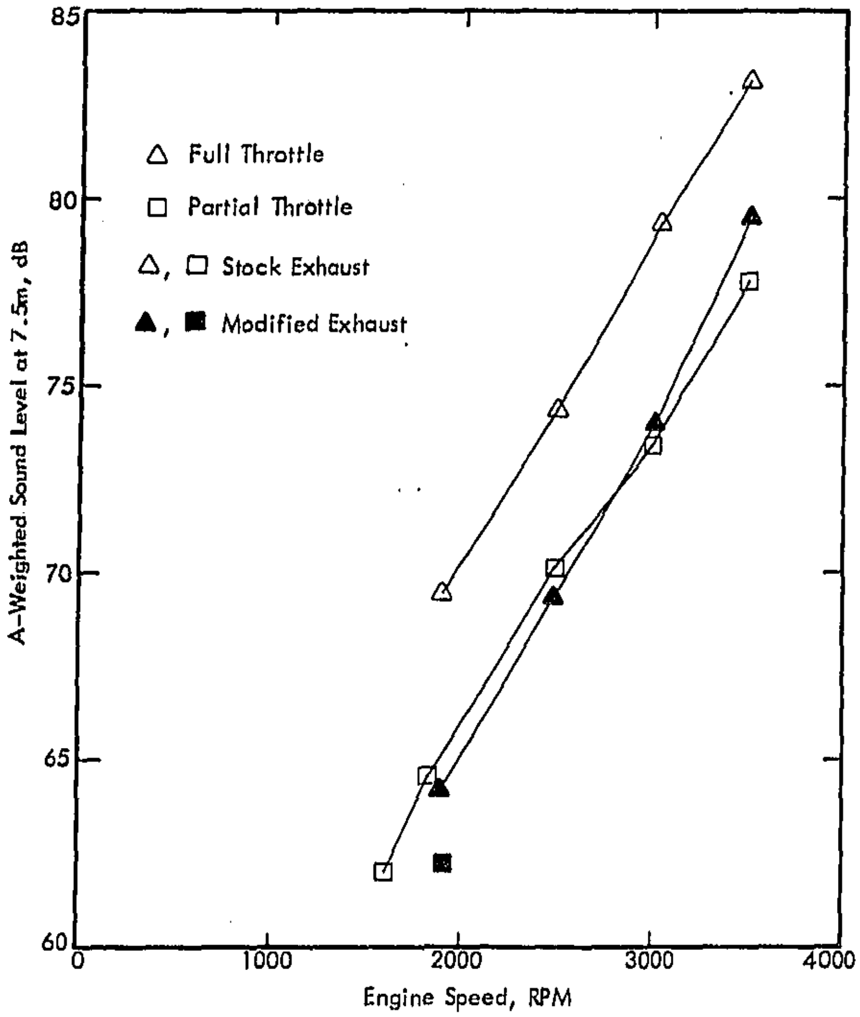

Figure 39. Comparison Between Stock and Modified Exhaust Systems, Oldsmobile Diesel.

Ideally, noise generated by the engine itself should be reduced. This is a very complex problem, however, and it is not expected that systematic solutions will be available in the near term. Accordingly, control of engine noise was obtained by the use of enclosures.

The engine compartment on a motor vehicle is essentially a box with three of the sides — front, rear and bottom — left open. The front is partially blocked by the radiator, the rear by the firewall, and the bottom is open. Of the three closed sides, the left and right sides often have some openings to allow clearance for front suspension and steering components.

Engine cooling is a major consideration in engine compartment configuration. Total encapsulation would require radical changes to vehicle design. Accordingly, this study considered only flow-through enclosures, where the front and rear were allowed to remain open for cooling air to pass through. Limited tests were conducted to examine the noise control implications of this constraint.

Partial enclosures, consisting essentially of belly pans, were constructed on four of the test vehicles. The belly pans were attached to the bottom of the radiator and to the frame rails, and extended rearward to around the position of the transmission bell housing. A space of several inches (this varied, according to the space available) generally existed between the engine oil pan and the belly pan. One to two inches of Fiberglass or Kaowool was placed in this area, and along the inner fender walls and firewall. The belly pans themselves were constructed of .032 inch lead, weighing about two pounds per square foot. This thickness of lead was used because of its availability; half the thickness would have provided adequate transmission loss. Gaps in the inner fender walls were filled with lead.

Table 8 summarizes the noise reductions obtained for the belly pans. Sideline noise reduction was in the range of 3 to 4 dB. Removal and substitution of components showed that this noise reduction applied to all underhood sources, including the cooling fan.

Because of the open nature of engine compartments, there is a question as to whether this type of modification should be viewed as an enclosure or as a shield. Viewed as an enclosure, absorptive treatment would be important. Viewed as a shield, the size and position of the enclosure with respect to the acoustic path is important.

Table 8. Noise Reduction Obtained with Partial Enclosures

Vehicle	Noise Reduction, dB*
Ford F-150	3.0
Toyota Corolla	2.6
Oldsmobile Diesel	4.3
Ford Pinto	3.4

*At EPA Test Condition.

Table 9 shows the findings for the Oldsmobile Diesel for five different absorption/belly pan combinations. The "complete" case is the belly pan as described above. The "front" case is the front half of this alone. (This case was suggested by the findings of General Motors that a substantial part of engine noise is radiated by the crank pulley.[5]) The "extended" case corresponds to the belly pan including the area under the bell housing. The open area at the rear of the extended pan was comparable to that for the "complete" case.

It is seen in Table 9 that both the length of the belly pan and the presence of absorption are important. The placement of the absorption was found to be of importance as well; benefit was obtained only if it was on the sides or bottom of the enclosed compartment. Placing large amounts of absorption above the engine, as shown in Figure 40, provided no measurable sideline noise reduction on any of the test vehicles. The large inner boundary associated with the engine prevents the engine compartment from being viewed as a single reverberant space; reverberant buildup occurs locally and must be treated with appropriately placed absorption.

The sound transmission path from the engine compartment was investigated by draping various parts of the vehicle with leaded vinyl weighing about one pound per square foot and backed with 1/4-inch open cell foam. Drapes were placed over three areas:

Table 9. Noise Reduction for Various Enclosures on Oldsmobile Diesel

	No Absorption	Absorption*
Front Enclosure	0.1 dB	-
Complete Enclosure	1.5 dB	4.3 dB
Extended Enclosure	3.0 dB	5.5 dB

*Absorptive material placed on inner fenders and firewall. Both cases had absorption lining the belly pan.

Table 10. Noise Reduction Obtained by Blocking Various Transmission Paths, Oldsmobile Diesel

	No Belly Pan	Belly Pan*
No Drapes	0 dB	4.3 dB
Drape Below Fender	4.3 dB	6.6 dB
Drape Mid-Section	-	5.6 dB
Fender Plus Mid-Section	8.3 dB	9.1 dB
Fender, Mid-Section, Plus Front	6.7 dB	-

*Complete belly pan with absorption.

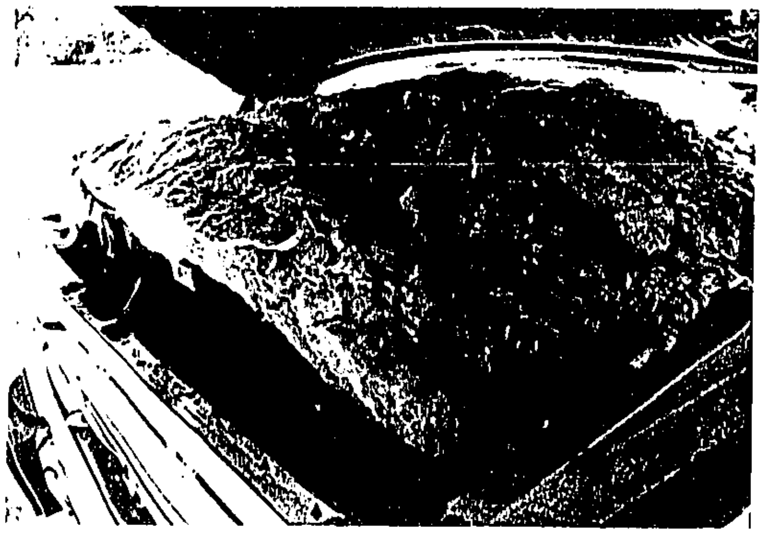

Figure 40. Typical Placement of Absorptive Material (15 cm Fiberglass) Under Hood, Resulting in No Sideline Noise Reduction

1. Front fender, extending to the ground.

2. Mid-section, filling the space from the rocker panel to the ground and from the front fender edge to the rear of the front door.

3. Front of car, covering the grille and extending to the ground.

Table 10 shows the results for the Oldsmobile. Draping the fender gave noise reduction similar to that obtained with the belly pan. The improvement with the belly pan indicates that the construction of the belly pan was somewhat lacking with respect to detailed fit and sealing. The very large noise reduction with both fender and mid-section drapes indicates that a major transmission path is to the rear. This is consistent with the data in Table 9 for the extended belly pan.

Covering the front of the vehicle was of no benefit. The measurements on the Oldsmobile showed an increase, which can occur due to re-direction of sound. Lead-wrapping the front of the Toyota and draping the front of the Ford Pinto resulted in no change to sideline noise. Because of potential cooling problems, tests with the front of the vehicle covered were of necessity brief, with a corresponding decrease in precision. However, the measurements on these three vehicles certainly suggest that little or no sideline noise reduction could be expected from a front shield.

Side shields are important if there are openings in the inner fender walls. Filling such openings on the Ford F-150 gave a 1 dB reduction to underhood sources Belly pans for the other vehicles included side shields; this is an important potential noise leak.

Draping vinyl over the hood and fenders (without blocking the space below the vehicle) provided no measurable noise reduction. The normal construction of vehicle bodies has sufficient transmission loss so that under-hood sound is transmitted under the vehicle, and not through the sheet metal.

Based on the data discussed here, the following conclusions have been reached:

- A basic enclosure, as described earlier, will reduce noise from under-hood sources by about 3 to 4 dB. Absorptive material must be included.

- If a well-sealed enclosure is constructed under-hood sources can be reduced by about 5 dB. Some rearward extension of the basic enclosure will be required.
- Further improvements would require extending the enclosure rearward.

6.1.4 Engine Accessories

Two other noise sources were identified in the test vehicles: drive belts and accessories, and the alternator. In general, belt and accessory noise was difficult to isolate from engine noise, and quite possibly cannot be treated separately (especially when vibration of the accessory on its mount is involved), so that this is properly treated as part of engine noise. On one vehicle, the alternator was a major noise source. Althrough no testing was performed to evaluate quieting of this particular unit, there are three possible reduction techniques which could be readily implemented:

- Reduce the alternator speed by changing the pulley ratio. Based on data measured for the Ford F-150 alternator, a 25 percent reduction in speed would reduce its noise by about 6 dB. It would then be significantly quieter than the engine. This approach may adversely affect battery charging at idle, however.
- A slip clutch could be used in the alternator drive. This would permit adequate alternator speed at idle, but would slow it at high speeds. Alternator speed was significant only at a speed well above that needed for full electrical output.
- Substitution of a quieter alternator. There was no apparent reason for the alternator on the Ford F-150 to be this noisy. Alternators on the other vehicles were quieter at similar engine RPM.

6.2 Quiet Vehicle Configurations

The component noise reduction analysis discussed in Section 6.1 provides the following conclusions for nominal noise reduction of various components:

- Fan noise is essentially eliminated by use of a clutch fan.
- A basic partial enclosure, as described earlier, reduces under-hood sources by 3 dB.
- A well sealed partial enclosure, extended under the bell housing, reduces under-hood noise by 5 dB.
- Exhaust outlet noise can be reduced by 3 to 6 dB through the use of larger and/or additional mufflers.

Calculation of the effect of various combinations of these modifications has been performed for the five test vehicles, using the component levels summarized in Table 7 for the EPA test condition. Reduced levels are presented in Tables 11 through 15. The modifications, and the sequences of application shown in the tables, follow the description in Section 5.2. The levels shown correspond to the totals of the source components in Table 7. Note that there are slight variations between these totals and the stock values directly measured; the calculations as presented provide consistent projections of noise reduction. The calculated noise reductions presented in Tables 11 through 15 are consistent with measured quiet configurations, differing only to the extent that actual component modifications did not match the nominal values used here. For example, enclosures on the Oldsmobile reduced under-hood noise by 4.3 and 5.5 dB, not the nominal 3 and 5 dB. For exhaust system noise reduction, a nominal 5 dB has been used except for the Ford F-150 and the Oldsmobile Diesel. 6 dB is used for the Ford because exhaust noise was a significant noise source, and 3 dB was used for the Oldsmobile because that was the smallest noise reduction directly measured for the modified exhaust.

Table 11. Reduced Noise Levels and Configurations, Ford F-150

Clutch Fan	•	•	•	•	•	•	•
Slow Alternator		•	•	•	•	•	•
Reduce Exhaust 6 dB			•		•		•
Basic Enclosure				•	•		
Sealed and Extended Encl.						•	•
Level at 7.5 m, dB	72.3	71.2	70.3	69.2	68.1	68.7	66.9
Reduction, dB re: Stock	1.7	2.8	3.8	4.8	5.9	5.3	7.1

Level for stock configuration: 74.0 dB @ 7.5 m.

Table 12. Reduced Noise Levels and Configurations, Toyota Corolla

Basic Enclosure		•	•		
Sealed and Extended Encl.				•	•
Reduced Exhaust 5 dB	•		•		•
Level at 7.5 m, dB	73.2	71.7	70.5	70.5	68.9
Reduction, dB, re: Stock	0.7	2.2	3.4	3.4	5.0

Level for stock configuration: 73.9 dB @ 7.5m

Table 13. Reduced Noise Levels and Configurations, Chevrolet Chevette

Basic Enclosure		•	•		
Sealed and Extended Encl.				•	•
Reduce Exhaust 5 dB*	•		•		•
Level at 7.5m, dB	70.4	68.4	67.7	67.0	65.9
Reduction, dB, re: Stock	0.5	2.5	3.2	3.9	5.0

Level for stock configuration: 70.9 dB @ 7.5 m

*May not be feasible due to space restrictions.

Table 14. Reduced Noise Levels and Configurations, Oldsmobile Diesel

Basic Enclosure		•	•		
Sealed and Extended Encl.				•	•
Reduce Exhaust 3 dB	•		•		•
Level at 7.5 m, dB	74.5	73.2	72.2	72.2	70.8
Reduction, dB, re: Stock	0.7	2.0	3.0	3.0	4.4

Level for stock configuration: 75.2 dB @ 7.5 m

Table 15. Reduced Noise Levels and Configurations, Ford Pinto

Clutch Fan	•					•
Basic Enclosure		•	•			
Sealed and Extended Enclosure				•	•	•
Reduced Exhaust 5 dB			•		•	•
Level at 7.5 m, dB	70.9	69.8	68.2	68.8	66.7	65.9
Reduction, dB, re: Stock	0.8	1.9	3.5	2.9	5.0	5.8

Level for Stock configuration: 71.7 dB @ 7.5m

7.0 CONCLUSIONS

A series of noise measurements has been conducted on five light vehicles, utilizing an inertial dynamometer to simulate acceleration load. Measurements were made to define the noise characteristics of the vehicles and their components over a range of operating conditions. Various noise reduction techniques were investigated to the extent of demonstrating their acoustical effectiveness. Major conclusions of this study are summarized below.

7.1 Overall Vehicle Characteristics

- Noise from light vehicles is a function of engine RPM and throttle setting alone. For an accelerating vehicle, this functional dependence is independent of transmission gear.
- Noise is very strongly dependent on RPM. For the four spark ignition engine vehicles tested, the noise level increased by 12.1 to 15.6 dB when engine speed was increased from half to full rated RPM. The noise level of the Diesel engine vehicle changed by 8.8 dB over this range.
- The difference in noise level between no load and full power was from 1.6 to 5.5 dB for the vehicles tested. The dependence of noise on throttle setting, while significant, is secondary compared to the RPM dependence. The Diesel exhibited the lowest throttle dependence of the vehicles tested.
- The noise characteristics of a particular vehicle depended strongly on the nature of the component sources, whose rank ordering was found to change with RPM and throttle for some vehicles.

7.2 Component Source Characteristics

- All source components exhibited strong dependence on engine RPM, comparable in range to that noted above for vehicle noise. Throttle dependence varied from no dependence for belt driven accessories through very strong depencence (up to 15 dB from no load to full throttle) for exhaust noise.

- In general, rank ordering of the sources varied between vehicles and between throttle settings for the same vehicles.

- It was found that major noise sources at no load and full throttle were always either dominant or important secondary sources at the part throttle test point defined in Reference 1. This test condition thus provides an excellent compromise for identifying sources which are important over a wide range of conditions.

- The major noise source in most cases was the engine. Exhaust noise was only important at full throttle. Cooling fans can be important if directly driven; the fan was the major source on one vehicle.

- The alternator was found to be a major source on one vehicle, due to its cooling fan.

- No significant noise was found to radiate from the exhaust pipes or muffler bodies. Such sources, where measurable, would become significant only after other sources (including exhaust outlet noise) were substantially reduced.

7.3 Noise Reduction Modifications

- In all cases, under-hood sources dominated at the part throttle condition. Exhaust system modifications would be useful only after these sources were quietened.

- Cooling fan noise, where significant, can be essentially eliminated by the use of a thermostatic clutch fan drive.

- Under-hood sources can be reduced by 3 dB by means of a flow-through enclosure. This enclosure consists of a belly pan under the engine, extending from the radiator to the bell housing. Holes in the fender walls must be shielded. The enclosure may be constructed of material similar to body sheet metal; a mass of one to two pounds per square foot is adequate. The enclosure must be lined with absorptive material to be effective.

- Under-hood sources can be reduced by 5 dB by means of a well sealed and extended partial enclosure. All joints and all holes in the fender wall must be sealed. The enclosure must be extended rearward to include the bell housing.
- With an enclosure in place, engine noise is transmitted out through the rear. Further improvements would require extending the enclosure rearward. The opening through the radiator and grille was not found to be a significant transmission path for sideline noise.
- Following enclosure of the engine, exhaust noise can be an important source. This can be reduced by 3 to 6 dB by increasing the size and/or number of mufflers in the system. Available space is the limiting factor on smaller cars.

REFERENCES

1. Sharp, B.H., and Donovan, P.R., "Light Vehicle Noise: Volume I — Development of a Test Procedure to Measure the Noise Emissions of Light Vehicles Operating in Urban Areas", Wyle Research Report WR 78-2, prepared for the U.S. Environmental Protection Agency, November 1978.

2. Sharp, B.H., Donovan, P.R., and Kohli, V.K., "Light Vehicle Noise: Volume II — Implementation and Evaluation of a Test Procedure to Measure the Noise Emissions of Light Vehicles Operating in Urban Areas", Wyle Research Report WR 78-18, prepared for the U.S. Environmental Protection Agency, November 1978

3. Graham, J.B., "Fans and Blower". Chapter 27 in Handbook of Noise Control, Edited by C.M. Harris, McGraw-Hill, New York, 1979.

4. Rowley, P., and Priadka, N., "State of the Art of Present-Day Intake and Exhaust Systems". Paper Presented at Noise-Con 73, Washington, D.C., October 1973.

5. Private Communication with General Motors, December 14, 1977.

APPENDIX A

TURBOCHARGED DIESEL PASSENGER AUTOMOBILE

An experimental turbocharged diesel powered Volkswagen Rabbit, on loan to the Department of Transportation for fuel economy and emission testing, became available for noise measurement. Although not a production vehicle or prototype, it is an engine configuration which may become more prevalent in future years for fuel economy reasons. The vehicle was essentially a turbocharged version of the production Diesel Rabbit described in Reference 2 and therefore would provide a direct comparison of the effects of turbocharging on noise. Table A1 summarizes the specifications of this vehicle.

Facility and Instrumentation

Testing was conducted at EPA's Noise Enforcement Facility in Sandusky, Ohio. Moving tests were conducted, utilizing the following instrumentation:

- Three acoustic channels were used utilizing B&K 4163 1/2-inch condenser microphones, 2619 preamplifiers, 2804 microphone power supplies, and 2607 measurement amplifiers.
- For the moving tests, microphones were placed at 15m to the left and right of the measurement point and one at 7.5m to the side.
- A-weighted levels were recorded on a Gulton 4-channel chart recorder. Signals from the two 15m microphones were recorded on a Kudelski Nagra IV — SJ magnetic tape recorder.
- Engine speed was measured using a magnetic pick-up and frequency-to-voltage converter as described in Section 3.2.
- Vehicle speed was measured using the fifth wheel system described in Reference A1.
- Acceleration was obtained by differentiating the speed signal.
- Vehicle data were recorded on a Gulton 8-channel chart recorder.
- To mark the position of the vehicle, a pressure tape switch was placed on the vehicle path. The switch was connected to a bistable multivibrator.

Table A1. Specifications of Turbocharged Diesel Volkswagen Rabbit

Size:	SC
Curb Weight:	2072 pounds
Engine:	L4
CID:	90
BHP:	70 @ 4800 RPM
Torque:	90 @ 3000 RPM
Transmission:	4M
Tires:	Semperit 155 SR-13

The output signal from this was a square wave, initiated by the passage of the front wheels and terminated by the rear wheels. This signal was recorded on the acoustic data chart recorder. A radio link (using a pair of Citizen's Band transceivers) was used to simultaneously transmit this signal to the on-vehicle chart recorder, thereby synchronizing the two recorders.

Figure A1 is a photograph of the vehicle during a test. Figure A2 shows the vehicle interior with chart recorder, throttle stop (the hand throttle fixture described in the body of the report) and meter displays of speed, RPM and acceleration.

Tests Conducted

The following tests were performed:

- The EPA part-throttle acceleration test[A1]
- SAE J986a and SAE J1030[A2]
- Cruise and coast up to 89 km/h (55 mph)
- Stationary steady RPM
- Stationary Idle-Max-Idle (IMI) at full and EPA throttle settings.

For the stationary tests, the microphones were placed 7.5m to the left, 0.5m from the exhaust outlet, and 1.5m in front.

Measurement Results

Table A2 summarizes the results of the moving tests. Shown are A-weighted sound levels and the speed and engine RPM corresponding to those levels. Also shown for comparison are the corresponding values for the spark ignition and the normally aspirated Diesel versions of the same vehicle, from Reference A3.

The turbocharged vehicle produced the lowest noise levels. The difference between the turbocharged and normally aspirated Diesels is slight, however, and can be partially accounted for by the difference in test RPM; this RPM is defined as 70 percent of rated speed.

Figure A1. Vehicle and Test Site.

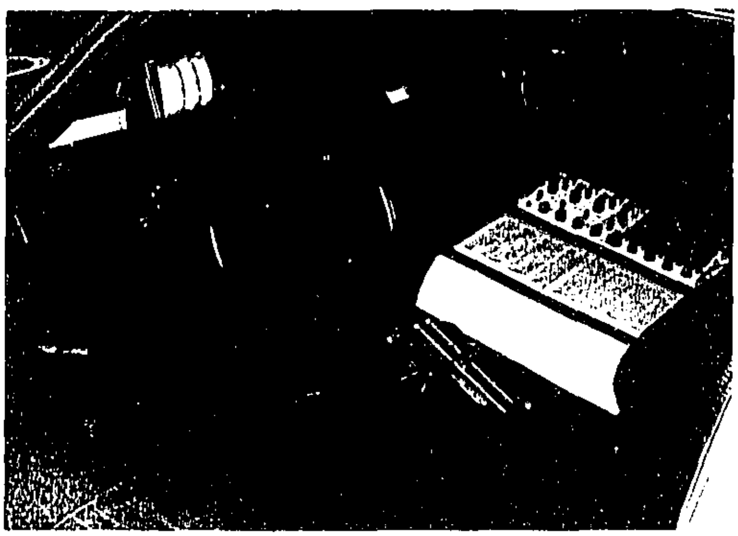

Figure A2. Instrumentation in Test Vehicle.

Table A2. Acceleration Noise Levels for Turbocharged Diesel, Compared with NA Diesel and Spark Ignition Models

Test	Vehicle	Speed km/h (mph)	RPM	L_A, dB,@ 15m
EPA Urban Acceleration	Turbocharged Diesel	27(17)	3360	68.0
	N.A. Diesel*		3499	69.2
	Spark Ignition*	31(19)	3960	69.3
SAE J986a	Turbocharged Diesel	48(30)**	3000	69.3
	N.A. Diesel*	48(30)	-	74.3
	Spark Ignition*	48(30)	-	72.7
SAE J1030	Turbocharged Diesel	73(45)	4400	75.7

*From Reference 2; vehicles #020 and #060.

**Approach speed

The SAE J986a level is substantially lower for the turbocharged vehicle. This test does not accurately reflect the full power noise, however. Due to the time lag associated with the turbocharger, the vehicle did not respond to throttle opening until the vehicle was well past the microphone position. The level measured under the SAE J1030 procedure, which is designed so that the vehicle is near the microphone when it is near rated RPM, is significantly higher. SAE J1030 data are not available for the other two vehicles.

Cruise and Coast Levels

Figure A3 shows the cruise and coast levels for the vehicles. Also shown for comparison are the acceleration test levels.

Noise Sources

In addition to moving tests, the following stationary tests were conducted for the purpose of evaluating noise sources:

- Steady RPM
- IMI with no load, at full and EPA test throttle settings.

These tests were conducted with microphones 7.5m to left side, 1.5m to the front, and 0.5m from the exhaust outlet. In addition, near-field, under-hood, measurements were made for qualitative source identification. No specific sources were found; under-hood noise is apparently all due to the engine.

Table A3 summarizes the component noise levels at 15m for the EPA test condition. These values are based on smoothing of the measured data (as described in Section 4.4) and are therefore about 1 dB below maximum levels.

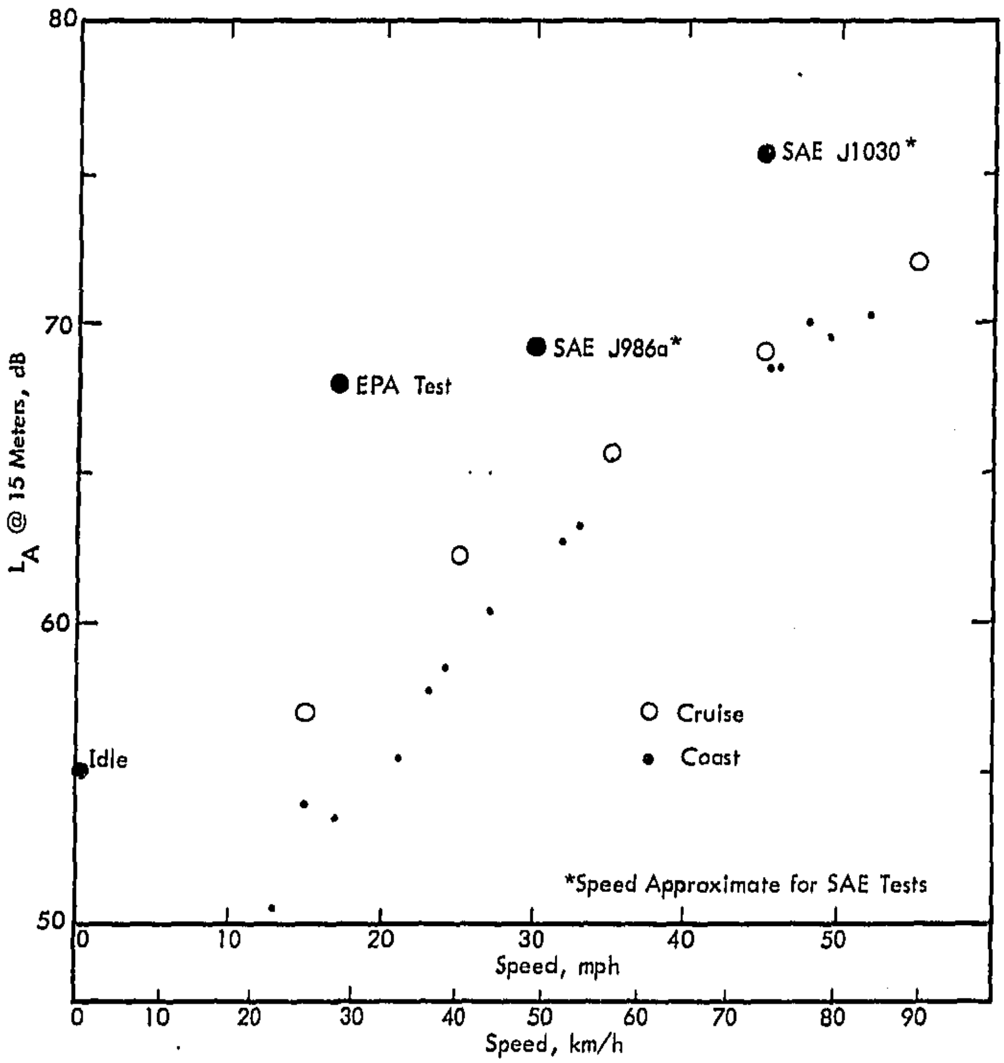

Figure A-3. Cruise and Coast Noise Levels, Turbocharged Diesel Volkswagen Rabbit.

Table A3. Noise Source Component Levels for Turbocharged Diesel Rabbit

Component	Level at 15m	Percent of Total Energy
Engine	66.5	93
Exhaust	55.0	7
Fan (when on)	48.5	-

REFERENCES

A1. Sharp, B.H., and Donovan, P.R., "Light Vehicle Noise: Volume I — Development of a Test Procedure to Measure the Noise Emissions of Light Vehicles Operating in Urban Areas", Wyle Research Report WR 78-2, prepared for the U.S. Environmental Protection Agency, November 1978.

A2. SAE Handbook 1978, Society of Automotive Engineers, Inc., Warrendale, PA, 1978.

A3. Sharp, B.H., Donovan, P.R., and Kohli, V.K., "Light Vehicle Noise: Volume II — Implementation and Evaluation of a Test Procedure to Measure the Noise Emisssions of Light Vehicles Operating in Urban Areas", Wyle Research Report WR 78-13, prepared for the U.S. Environmental Protection Agency, November 1978.

APPENDIX B

STATIONARY UNLADEN ACCELERATION TEST

For three of the vehicles tested in this study, unladen IMI noise tests were performed in addition to the loaded acceleration tests. Table B1 summarizes the results, comparing both. Shown are the levels at EPA test throttle and full throttle settings. Data shown are the averages of at least four runs.

Table B1. Loaded and Unloaded Acceleration Noise Levels, dB, at 7.5m

	EPA Throttle		Full Throttle	
	Loaded	Unloaded	Loaded	Unloaded
Oldsmobile Diesel (1900 RPM)	71.7	71.6	73.8	73.1
Ford Pinto (2520 RPM)	71.5	72.2	74.6	76.4
Turbocharged Diesel VW (3360 RPM)	73.5	73.5	-	-

Overall, the agreement is very good. Conclusions cannot be drawn from tests on only three vehicles; however, this stationary test technique appears to be well worth investigating further. If demonstrated to be practical, it would be a great simplification over the moving test.