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# NOISE ABATEMENT TRADEOFF CONSIDERATIONS

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D. Morrison

Ricardo Consulting Engineers, Ltd.
Sussex England

MAY 1981

FINAL REPORT

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16. Abstract

The intent of this study is to obtain information on small high-speed engines so that their effect on the urban environment may be assessed, and if necessary, programs devised to reduce the noise and other emissions from vehicles using these "highly desirable and efficient power plants." This work was performed by Calspan and subcontracted to Ricardo Consulting Engineers, Ltd.

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# PREFACE

Various practical measures of noise control are presented and discussed for passenger car vehicles. An analysis was performed estimating the trade-offs and their corresponding effects on fuel economy, cost and weight. The baseline vehicle considered was essentially a compact passenger car (fitted with a so-called "high speed" engine, as is commonly used in European cars) with a curb weight of 1100 kg (2400 lb). The car was referenced to the proposed 1981 Federal Light Duty Vehicle Emission standards of 1.41/3.4/1.0 g/mile (HC/CO/NOx respectively) but consideration was also given to noise control measures which may have some relevance in the context of current 1979 emission standards (2.0/15/1.5 g/mile, HC/CO/NOx). Where relevant, the proposed particulate emission standards were also considered (i.e. 0.6 g/mile, 1981 and 0.2 g/mile, 1983). Fuel economy changes were based on composite values (i.e. weighted averages of the FTP urban and highway test results). The effect of the noise control measure was with reference to a current high engine speed, full load drive-by test procedure (i.e. SAE J986b or similar European procedures - 70/157/EEC).

Both diesel and gasoline powered automobiles were considered. The diesel engines were of the indirect-injection type and conventional spark ignition and stratified charge engines constituted the gasoline engines. Other engine types, for example the rotary and two-cycle engines, were not considered because of limited data and of very small market penetration and therefore relative insignificance.

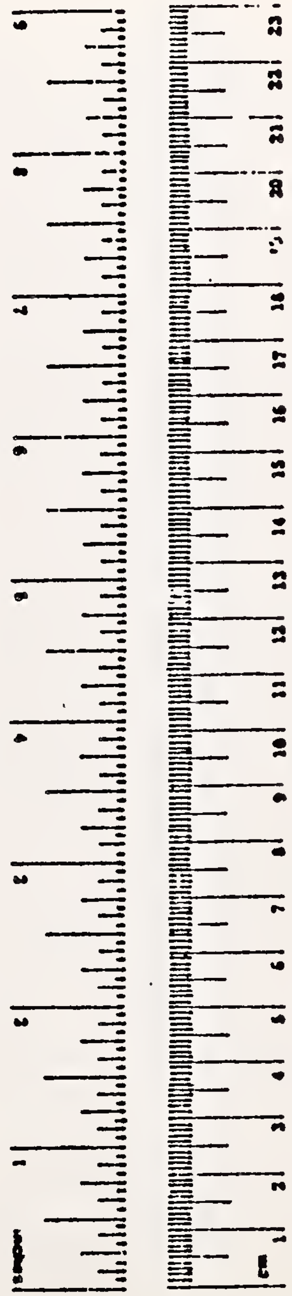
The results have been condensed into tabular form and constitute the essence of the report. These tables and their supporting notes are intended as concise summaries and may be regarded and consulted as part of this initial summary. In many cases the trade-off trends shown are best estimates or based on computer prediction models. Where possible, actual data is presented, mainly from the detailed test work carried out on a SAAB 99GL and Peugeot 504 GLD and their respective engines.

The major findings are summarized overleaf for noise reduction technologies and are broadly generalized for both diesel and gasoline cars. These noise reduction measures represent those which are considered to-date, to be most likely incorporated in future vehicles.

## Approximate Conversions to Metric Measures

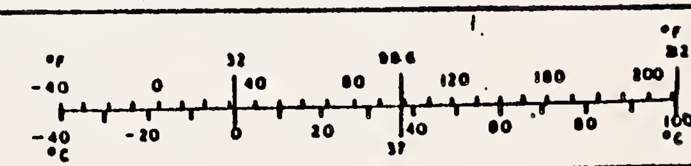
| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| in | inches | *2.5 | centimeters | cm |
| ft | feet | 30 | centimeters | cm |
| yd | yards | 0.9 | meters | m |
| mi | miles | 1.6 | kilometers | km |
| | | **AREA** | | |
| $in^2$ | square inches | 6.5 | square centimeters | $cm^2$ |
| $ft^2$ | square feet | 0.09 | square meters | $m^2$ |
| $yd^2$ | square yards | 0.8 | square meters | $m^2$ |
| $mi^2$ | square miles | 2.6 | square kilometers | $km^2$ |
| | acres | 0.4 | hectares | ha |
| | | **MASS (weight)** | | |
| oz | ounces | 28 | grams | g |
| lb | pounds | 0.45 | kilograms | kg |
| | short tons (2000 lb) | 0.9 | tonnes | t |
| | | **VOLUME** | | |
| tsp | teaspoons | 5 | milliliters | ml |
| Tbsp | tablespoons | 15 | milliliters | ml |
| fl oz | fluid ounces | 30 | milliliters | ml |
| c | cups | 0.24 | liters | l |
| pt | pints | 0.47 | liters | l |
| qt | quarts | 0.95 | liters | l |
| gal | gallons | 3.8 | liters | l |
| $ft^3$ | cubic feet | 0.03 | cubic meters | $m^3$ |
| $yd^3$ | cubic yards | 0.76 | cubic meters | $m^3$ |
| | | **TEMPERATURE (exact)** | | |
| °F | Fahrenheit temperature | 5/9 (after subtracting 32) | Celsius temperature | °C |

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



## Approximate Conversions from Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| mm | millimeters | 0.04 | inches | in |
| cm | centimeters | 0.4 | inches | in |
| m | meters | 3.3 | feet | ft |
| m | meters | 1.1 | yards | yd |
| km | kilometers | 0.6 | miles | mi |
| | | **AREA** | | |
| $cm^2$ | square centimeters | 0.16 | square inches | $in^2$ |
| $m^2$ | square meters | 1.2 | square yards | $yd^2$ |
| $km^2$ | square kilometers | 0.4 | square miles | $mi^2$ |
| ha | hectares (10,000 $m^2$) | 2.5 | acres | |
| | | **MASS (weight)** | | |
| g | grams | 0.035 | ounces | oz |
| kg | kilograms | 2.2 | pounds | lb |
| t | tonnes (1000 kg) | 1.1 | short tons | |
| | | **VOLUME** | | |
| ml | milliliters | 0.03 | fluid ounces | fl oz |
| l | liters | 2.1 | pints | pt |
| l | liters | 1.06 | quarts | qt |
| l | liters | 0.26 | gallons | gal |
| $m^3$ | cubic meters | 35 | cubic feet | $ft^3$ |
| $m^3$ | cubic meters | 1.3 | cubic yards | $yd^3$ |
| | | **TEMPERATURE (exact)** | | |
| °C | Celsius temperature | 9/5 (then add 32) | Fahrenheit temperature | °F |




METRIC CONVERSION FACTORS

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TABLE P-1.

| Noise Reduction Measures | Noise Reduction (dBA) | Vehicle Weight Change (%) | Fuel Economy Change (%) | Vehicle Cost Change (%) |
|---|---|---|---|---|
| 1. Reduced Engine Speed by 10% | 1 to 2 | 0 to +3 | +3 to +4 | 0 to +7 |
| 2. Engine Size/Configuration | 1 to 2 | 0 to +2 | -4 to +5 | 0 to +5 |
| 3. Combustion Process Changes | 0 to 1 | -1 to +1 | -4 to +5 | 0 to +7 |
| 4. Engine Structure/External Components/Shielding | up to 8 | 0 to +3 | -2 to 0 | +1 to +5 |
| 5. Intake/Exhaust Improvements | 0 to 2 | +1 to +2 | -5 to -4 | +1 to +2 |

# 1. INTRODUCTION

This report was prepared as part of the "High-Speed Engines" Project for the U.S. Department of Transportation, Ricardo being subcontracted through Calspan Corporation. The complete project covered noise, performance, economy and emission tests on two typical European passenger cars (a SAAB 99GL and a Peugeot 504 GLD) and their respective engines. Tests on 10 additional cars were also conducted.

This report is specifically concerned with the various practical options for light duty vehicle noise control. The reference vehicle type considered is the passenger car powered by a "high-speed" engine. In the context of this report, the size of the car is 1100 kg curb weight (2400 lb) and may be powered by either a spark ignition gasoline engine or an indirect injection diesel engine. The engine powers considered are 70 kW for the gasoline engine and 50 kW for the diesel engine. Such engine/vehicle combinations would be considered typical for many current European passenger cars. For the purposes of this report two emission builds have been considered as baselines. First, the current 1979 Federal standards (2.0/15/1.5 g/mile, HC/CO/NOx respectively) and second the proposed 1981 Federal standards (0.41/3.41/1.0 g/mile, HC/CO/NOx). In certain instances consideration has also been given to impending exhaust particulate legislation. Because of general improvements in emission control devices and the trend towards improving fuel economy the build differences are likely to have only very small effects on noise control. Also, in the case of the so-called "high speed" engine the engine noise is often very largely controlled by mechanical noise sources rather than combustion. This is so even in a light duty diesel engine at the relatively high speeds called for in current noise test procedures.

When considering a given noise reduction measure in this report, the corresponding attenuation achieved and other effects (fuel economy, weight, cost) are always referred to these baseline cars. i.e. the effects are not necessarily additive. The major noise source for such vehicles is normally the engine, and it is this area which the report concentrates on when considering the various noise abatement options and their impact on vehicle fuel economy. Influences on weight and cost will also be considered.

## 2. VEHICLE NOISE SOURCES

The principal passenger car noise sources (approximately in order of importance as far as current legislative test procedures are concerned) are typically, engine, exhaust, intake, fan, rolling (tire and wind noise) and transmission. A schematic model of these passenger car noise sources is shown in Figure 1. In many instances certain of these sources are negligible compared with the major source. Two examples of source breakdowns are given in Figure 2 for the two typical European passenger cars - the SAAB 99GL and the Peugeot 504 GLD. (These cars were the subject of detailed drive-by and engine noise tests[1] and will be referred to frequently during this report. A specification of these cars is given in Appendix 1). The breakdown shown is for the maximum noise level condition at 7.5 m during a "wide-open-throttle" acceleration following a zone entry speed of 50 km/h (ref 1, Figures 13 and 31). The results show that in the case of the gasoline engine car (the SAAB) on the left hand side the exhaust noise was the dominant source whereas the engine was by far the major noise source on both sides in the case of the diesel engine car (the Peugeot). In the SAAB, the exhaust noise was particularly high to the LHS of the car. To the right, the engine was the major source (as shown in Figure 2). The results serve to illustrate: a) that the engine is not always the major source, b) that the engine if not a major source is still a significant source, c) that the intake, fan and rolling noise sources are generally small by comparison with the engine. As a generalization it is considered that the breakdowns shown for each of these two cars is a fair representation of the two classes of vehicle (i.e. gasoline and diesel).

As has been stated earlier, because of the importance of the engine as a vehicle noise source, this report will concentrate mainly on engine noise abatement. It is therefore relevant to examine the breakdown of sources of the engine itself. The fan, intake and exhaust sources are considered separate from the engine noise in this context, and the engine noise is then broadly broken down into "combustion" and "mechanical" noise. The combustion noise as the name implies, is that noise originating from and completely influenced by the combustion process. Mechanical noise is that noise associated with mechanical impact (e.g. piston slap), bearing noise and the valve train. The combined effect of these two sources is to excite the engine structure, which responds to a greater or lesser degree. The magnitude of the response depends on a great many factors and by careful design the structural response can be greatly reduced for a given excitation input. This will be mentioned in more detial in Section 3. A schematic model of engine combustion and mechanical noise is shown in Figure 3.

# 3. NOISE CONTROL MEASURES

The various passenger car noise sources will be discussed in turn with respect to abatement measures and to exterior noise. Section 3.7 discusses interior noise.

## 3.1 ENGINE NOISE CONTROL

Figure 3 shows a simplified model of engine noise. The three major sources may be broadly classified as combustion, mechanical and gas flow. The various important options for engine noise reduction may be considered in relation to these three areas as follows:

### 3.1.1 Changes to the Combustion Process

On a combustion noise controlled engine, reducing the combustion excitation by timing retard can be of benefit. Different engines respond in different ways, however, (in particular from the point of view of the structure response and piston slap, for example). It is important to establish that the combustion noise is significant, therefore, before attempting measures to reduce it. On most gasoline engines of the size considered in this report, the noise associated with the true combustion excitation is normally small or negligible compared with the mechanical sources.

Retarding the ignition or injection timing is perhaps the simplest means of changing the combustion characteristics. Other possibilities are changing the compression ratio, improving the spark energy or number of ignition sources (for a gasoline engine), fumigation (for a diesel engine), turbocharging and exhaust gas recirculation (EGR). For combustion noise reduction, the objective is essentially to reduce the rate of change of pressure rise (i.e. at the start of combustion) and also to reduce the maximum cylinder pressure. All of the above measures are aimed at acheiving one or more of these effects, for example, by reducing the combustion delay period. The argument remains, however, that unless the combustion noise is a significant source then such measures are unlikely to be effective in reducing the overall radiated noise. It should also be noted that exhaust emission and fuel economy requirements place strict bounds on the combustion optimization.

a) Timing Control

Figure 4 illustrates the small changes in sound pressure level achieved with relatively large changes in spark timing for the SAAB 2 litre engine. The disproportionately greater effect on the cylinder pressure level spectrum is shown in Figures 5 and 6. From this type of data, a basic combustion/mechanical noise source breakdown is possible[2], the results of which are shown in Figures 7 and 8. These results typically illustrate that even at full load, the combustion noise excitation of a standard gasoline engine is small in comparison with the mechanical sources. Timing retard on a gasoline engine is therefore unlikely to be very effective as a noise reduction measure, especially on the already retarded low emission build engine considered as the baseline.

In the case of the light duty indirect injection diesel engine, however, at normal operating loads and speeds the effect of the combustion noise on the overall noise can be influenced significantly by injection timing. In some engines it is possible to observe a rate of change of noise level with timing as high as 6 dBA/10° (i.e. similar to that expected from naturally aspirated direct injection diesel engines) when the timing is relatively advanced.[3] On most indirect injection engines, however, the process of engine development in recent years is leading to the adoption of retarded timings in the interest of noise and emission reduction. The application of further injection retard therefore reduces the delay period to a minimum fairly quickly and engine performance begins to be impaired. This is discussed more fully in the next section and in the Tables.

The effect of injection timing on the sound pressure level is shown in Figure 9 for a typical 2 litre light duty IDI diesel engine. A combustion/mechanical noise breakdown is also shown in Figures 10 and 11. Here it may again be seen that whereas an advance in injection timing generally increases the radiated noise, retarding from standard conditions yields only marginal reductions due to the small contribution of the combustion noise to the overall radiated noise level.

b) EGR

The effect of exhaust gas recirculation on reducing combustion noise is likely to be small and in the case of a gasoline engine where the combustion noise is rarely a major noise source, is likely to be insignificant. The use of EGR is currently not envisaged as a noise reducing measure; rather, a means of reducing NOx formation. It is possible, however, that the EGR may be effective in reducing diesel idle noise, (particularly cold idle noise), by increasing cylinder temperature on engines where the combustion noise at idle is a dominant noise source.

c) Air Fuel Ratio

For a given engine load, on a gasoline engine, changing the air: fuel ratio within practical limits is unlikely to have any measurable effects on the overall radiated noise. This is again due to the fact that gasoline engine noise is not normally controlled by combustion noise as has already been shown in Figures 7 and 8. On a light duty diesel engine, the air: fuel ratio is dictated primarily by smoke and emission considerations and cannot be considered as a means of combustion noise reduction. Also, as the diesel engine operates unthrottled, the quantity of fuel injected (i.e. the fuel/air ratio) determines the engine power output for a given speed. The question of air/fuel ratio changes for a diesel engine is therefore not relevant from a noise reduction consideration.

For so-called "lean burn" homogeneous charge engines, again detailed noise data is not available but is thought that the combustion noise contribution is likely to be broadly comparable with that of conventional gasoline engines.

d) Stratified Charge

In the case of a "stratified charge" engine, detailed noise data is as yet not available but some preliminary tests on high compression ratio pre-chamber systems have suggested that the combustion noise on such engines is likely to be significant.[4] In broad terms the rates of pressure rise and $P_{max}$ values are likely to be some where between those of a light duty diesel engine and a gasoline engine, but there will be very large variation from engine to engine depending on the particular stratified charge system used. For example, on systems working essentially at "normal" gasoline engine compression ratios the combustion noise contribution may be small; $P_{max}$ should certainly be comparable. On the other hand such systems may develop higher rates of pressure rise. It is impossible to generalize in this case.

3.1.2 <u>Reduction of Engine Speed</u> (Combustion and Mechanical Noise Source Reduction)

This somewhat fundamental approach is nevertheless effective in reducing engine noise for a drive-by test where a fixed entry speed condition is specified. Reducing the rated power speed and utilizing higher gear ratios is of course also effective over the whole vehicle speed range. An alternative approach may be the use of turbocharging where the resulting increase in power allows the use of a higher final drive ratio. Typical noise/engine speed characteristics are 40-50 dBA/decade for a light duty IDI engine and 50-60 dBA/decade for a light duty gasoline engine. These characteristics are illustrated in Figure 12 and are based on actual results from a large number of engines. Such curves are useful for prediction purposes and showing trends between bore and size and speed differences. They also illustrate that as a first order approximation "conventional" multi-cylinder engine noise is a function of engine speed and bore only, at rated speed, full load conditions.

3.1.3 Bore: Stroke Ratio (Combustion and Mechanical Noise Source Reduction)

It has been shown above that for a given engine swept volume and load, the radiated noise is a primary function of engine speed and bore size (i.e. piston area), as shown in the prediction curves in Figure 12. Thus an undersquare configuration (with all other factors being equal) should be potentially less noisy than an oversquare. In designing present-day engines this factor is given serious consideration as there are other associated benefits from the reduced engine speed (for a given mean piston speed) - for example improved mechanical efficiency and hence better fuel economy.[5] The specific power output is reduced, however, and it is a question of selecting the required compromise between the principal parameters of engine size, weight, speed, bore/stroke ratio, power output noise and fuel economy.

3.1.4 Pistons (Mechanical Noise)

Noise associated with the piston, in particular piston "slap" (as the piston impacts the liner after moving across the bore around TDC) is one of the most important noise sources on an internal combustion engine. This is particularly so for diesel engines with their relatively high maximum cylinder pressures, piston slap being greatly influenced by $P_{max}$. Various designs of piston aimed at minimizing slap are possible but the one becoming widely used is that employing steel bracing bands or struts cast into the piston skirt. The object of this design is to limit the aluminum piston skirt expansion and thus permit tighter piston-bore clearances for cold or light load running in cast iron cylinder blocks. A typical design of such an expansion controlled piston is shown in Figure 13. An experimental alternative form of piston design to achieve close clearance running over the full operating range is the articulated piston, where the skirt is separate from the crown. This is also shown in Figure 13. Some actual noise reductions achieved over the speed range on a Comet V engine are shown in Figure 14. At cold idle the possible gains are even greater, as shown in Figure 15, but much depends on the clearances used with the standard engine and how significant piston slap is on a particular standard engine.

### 3.1.5 Valve Train (Mechanical Noise)

The noise associated with the valve train can be significant particularly when gears are employed. Chain drives are normally preferred to gears from a noise point of view. Further reductions can often be achieved with a toothed belt system, as used in many present day light duty passenger car engines (both diesel and gasoline). Curves showing the differences in noise levels associated with each of these three valve train systems are shown in Figure 16. The noise differences are based on actual results but are combined results from tests on two different 2 litre Comet V engines.

### 3.1.6 Changes to the Structural Response

a) Vibration Damping and Mass Loading - Vibration damping can reduce the noise from certain critical engine components (notably pressed steel items such as the oil pan and valve rocker cover). Such a palliative treatment can be very effective in cases where the resonant frequencies are high ($\sim$1kHz). A typical approach is to use sandwich sheet material (e.g. Sound Deadened Steel - SDS), with the damping element in the center. Mass loading can also be of value for the same type of pressed steel component where suitable pads may be bonded to the component in critical regions (usually unsupported, non-corrugated areas). Mass loading is sometimes used for the attenuation of low frequency internal vehicle noise caused by floor panel vibration. In this case suitable pads are bonded to the floor panel usually in the center of unsupported regions.

b) Isolation - As an alternative to damping, certain engines covers may be isolated from the crankcase by means of a suitably flexible material. Isolation is particularly effective (as with damping) at high frequencies and is therefore employed on such components as the rocker cover, front timing cover, and intake manifold. Isolation may also be possible for the oil pan but the physical strength and oil sealing of the isolation medium present significant engineering problems, where long, reliable operational lifetimes are required.

c) Structural Changes - Structural changes to the engine, as far as noise control is concerned, are aimed at either impeding vibration transmissions from an internal source (e.g. piston slap, combustion) or changing the vibration characteristics of the external surfaces so that less noise is radiated in the audible frequency range. In many cases, primary bending modes of the entire engine are significant and the use of longitudinal strengthening, bedplates or bearing beams can be effective in reducing these vibration modes. Examples of a bedplate and a bearing beam are shown in Figure 17. In an engine where, for example, the first free-free bending mode is at a low frequency and the vibration levels at this frequency are not giving rise to high radiated noise, then there is nothing to be gained in utilizing a heavy bed plate or beam. This can also be the case when considering modifications of a standard engine. The bedplate alone, without additional modifications to the engine, can be ineffective as a noise reduction measure. The bedplate does, however, allow further structural modifications to the engine to exploit this extra stiffness, and it is here that gains can be achieved.

Conventional light duty engine crankcases often suffer from excessive "panel" vibration, where the unsupported exterior panels between the main bulkheads are free to vibrate, excited by high frequency vibration from combustion and/or piston slap. Typical panel vibration occurs in the frequency range 1-4kHz and can therefore be a significant influence on the overall noise level. Figure 18 shows panel vibration on a six cylinder engine, at different 1/3rd octave frequencies. Panel vibration may be reduced by resorting to ribbing (either internal or external). Strategically located ribs stiffen the panels and also break up the panel into smaller areas. The rib must be of at least 2-3 times the casting thickness at this point to be of any measurable effect. Typical source noise reduction obtained from a ribbed cylinder block is shown in Figure 19.

Later generations of light duty engine could incorporate radically different structure design from current conventional types. Greater effort will be spent in designing the main engine structure as light as possible but at the same time having adequate stiffness and strength in critical areas for low noise. Such a structure system is shown in Figure 20 (the Ricardo Low-Noise Engine) where the strength and stiffness is concentrated in the wide horizontal "plates" in the aluminum cylinder block and in the upper and lower cast iron crankshaft main bearing carriers. A light, stiff aluminum bearing beam is also incorporated (see Figure 17). The outer surfaces of the engine are non-load bearing and may therefore be made from either a light, ribbed casting (very high resonant frequency) or laminated "damped" steel sheet (very low resonant frequency).

### 3.1.7 Shielding

Close fitting shields may be a practical palliative where structural changes are not feasible or possible. A close fitting shield is one which envelops part of the engine, normally the crankcase walls, and is mounted very close to the component being shielded (∿10-20 mm). Two of the most important considerations when designing sheilds are the mounting and the edge sealing. The mounting usually used is a simple rubber vibration isolator, but can be solid. Edge sealing may be by using rubber moulded strip (as used for window sealing on automobiles). Figure 21 shows the typical degree of attenuation expected with different types of close fitting shields. For some components, notably the exhaust manifold, shielding is the only practical noise abatement measure. (Isolation of this component may be possible in certain cases but the reliability and effectiveness is questionable in the long term).

### 3.1.8 Enclosures

Enclosures may be an integral part of the vehicle body and serve to either completely or partially enclose the engine. The enclosure walls are normally a considerable distance from the engine and act as barriers to the entire engine noise (as opposed to close fitting shields which are more specifically aimed at one particular area on the engine). One of the most effective and practical enclosure designs is one which takes the form of a tunnel, open at the top and bottom of the engine. An enclosure or partial enclosure can be designed such that access to the engine is not unduly restricted and in a passenger car application such areas as the hood and wheel arches can actually form part of the enclosure system. Total enclosures can be very effective in noise reduction but can give rise to potential cooling problems and accessibility to the engine unless great care is exercized.

## 3.2 EXHAUST NOISE CONTROL (GAS FLOW NOISE SOURCE REDUCTION)

The exhaust noise contribution to the overall exterior noise during a pass-by test varies greatly from car to car. In the case of the SAAB 99 the exhaust noise was subject to resonances and therefore at certain critical engine speeds the exhaust noise was considerably augmented. This is shown for a stationary test in Figure 22. The result of the same test for the Peugeot is given in Figure 23, showing the complete absence of any significant resonance peaks.

Compared with reducing engine radiated noise, attenuating exhaust noise to acceptable levels is relatively straightforward and in many current passenger cars the exhaust noise contribution is negligible. In order to avoid engine efficiency losses by causing high exhaust back pressures, the muffler system must be designed to offer minimum restriction to the exhaust flow, compatible with noise attenuation. This is often most effectively done by using two or more mufflers of sufficiently large volume to absorb the exhaust pressure energy. This is at the expense of increased vehicle weight and cost, however. More scientific approaches to muffler design will be used in the future, where optimized design may be achieved through computer modelling.

## 3.3 INTAKE NOISE CONTROL (GAS FLOW NOISE SOURCE REDUCTION)

The same basic arguments apply as for exhaust noise control. The intake silencer must not restrict the air flow and must be of sufficiently large volume to effect satisfactory noise attenuation at the low frequencies primarily associated with intake noise (i.e. at 2 and 4 times the engine speed for a 4 cylinder, 4 cycle engine). Intake noise control is more important on an unthrottled diesel engine than a gasoline engine at part load, but equally important for both categories during a full load pass-by test.

Improvements in intake muffler systems should well be able to keep abreast of noise reduction in the major vehicle noise source - the engine.

## 3.4 FAN NOISE CONTROL

A conventional radiator fan driven directly by the engine can be a significant noise source on a passenger car. The simplest and most effective control measure is to employ a thermostatically controlled electrically driven remote mounted fan (or fans). Most passenger car radiators are over-cooled by conventional fan systems and so noise reduction is directly achieved with an electrically driven fan system as this is rarely actually operative during normal driving. Also, the noise radiated by an electrically driven fan when operative is very low; typically 50-60 dBA at 1m from the radiator grill - insignificant at a 7.5 m or 15 m drive-by measuring distance.

## 3.5 ROLLING NOISE CONTROL

At typical current noise legislative vehicle speeds ($\sim$50km/h) the noise associated with tires and air resistance from the body work is very small (see Figure 2). An empirically derived equation for passenger car rolling noise, as a function of curb weight and vehicle speed is[6,7]

$$\text{Noise level @ } 7.5\text{m} = 32 \log V + 11.2 \log V - 20 \quad \text{dBA} \pm 1\text{dBA}$$

Where V = vehicle speed in km/h

W = curb weight in kg

Thus for the type of car considered (i.e. 1100 kg curb weight), at 50 km/h the rolling noise would be of the order 68 dBA at 7.5 m, compared with an overall noise level of 78-82 dBA, and has therefore only a very small influence on the overall drive-by noise.

It is unlikely that the performance of tires will be compromised for low noise for this class of vehicle. As far as noise associated with air resistance is concerned, this is already very small and in the future likely to be even less with the increasing trend towards better aerodynamic shape in the interests of fuel economy.

## 3.6 TRANSMISSION NOISE CONTROL

Transmission noise may be considered as that caused by the transmission itself, (i.e. gear noise, final drive noise) and that caused as a result of vibration excitation from the engine. Once again, exterior noise levels associated with passenger car transmission are normally very small compared with the engine at pass-by test conditions, and so no major measures for transmission noise control are envisaged to be required. One instance where transmission noise can be significant, however is during idling. On certain front wheel drive passenger cars with integral engine/gearbox assemblies, the transfer gears can cause considerable "rattle" when hot. This can be alleviated by resorting to a duplex chain drive, as now used on the SAAB 99.

For diesel engines, idle noise can be a problem and in some cases the gearbox excitations resulting from the cyclic torque fluctuations at idle can result in the gearbox radiating as much measured noise as the engine. In such cases, a means of isolating the transmission is required, possibly by declutching or using a specially modified clutch plate. The problem does not arise in automatic transmissions, where a fluid coupling is employed. Strengthening the gearbox casing by using ribs is another possible approach to reduce the surface noise radiation.

## 3.7 INTERIOR NOISE

The majority of the above comments and noise control measures for exterior noise are equally applicable to interior noise. Passenger car interior noise is very often largely controlled by low frequency noise and typical interior noise frequency spectra are shown in Figures 24-26. (Note that these spectra are in fact A-weighted thus illustrating the importance of the low frequency levels). Measures to reduce interior noise must therefore take this into account and must be aimed at the difficult task of reducing noise typically at frequencies below 200 Hz. The particular areas for considerations in reducing interior noise are:

a) Engine Noise - Engine noise can often be effectively reduced inside the car by using a high transmission loss material over the engine compartment bulkhead. Sealing of holes and vents in the bulkhead is critical. Engine compartment treatment by lining with absorbant material (e.g. foam, felt or fiberglass) is also a possible approach but is limited in practicability and normally restricted to lining the hood only. The mounting of the engine is important, as vibration transmitted through the mounts can excite resonances in various parts of the body structure, especially unsupported large panels, and the whole body at low frequencies. ("body-boom"),

b) Tire and Suspension Noise - Control of this essentially low frequency noise is normally limited to suspension location design, stiffness, and adequate interior damping and shielding (using high transmission loss material). Floor panel resonance excited by suspension "thump" may be reduced by using suitable damping pads.

c) Wind Noise - Wind noise may be controlled primarily by good sealing around doors and windows with avoidance of raised lips. The body shape is usually secondary to good sealing but a low-drag design is obviously advantageous from this aspect.

# 4. IMPACT OF NOISE CONTROL MEASURES ON FUEL ECONOMY, WEIGHT & COST

It must be appreciated that in many instances it will be impossible to present actual data on trade-offs between noise, fuel economy, cost and weight as such data is not readily available. A cost trade-off is especially difficult, as accurate estimations of future production costs of what are currently prototype noise reduction measures, are outside the scope of this report. It is proposed to deal with the various measures as discussed in Section 3 and where possible, present as much specific data as possible but where such detailed data does not exist, to estimate trends on the basis of engineering experience in this field, together with data from computer predictions.

Where noise reduction measures are discussed it will be assumed that the noise reduction and related parameters referred to are considered in comparison with present day vehicles, i.e. to 1979 Federal emission standards. Where any difference arises with respect to 1981 Federal limits this is qualified later.

The various vehicle noise sources will be considered in turn together with the appropriate control measures as set out in Section 3.

## 4.1 ENGINE NOISE

Engine noise control measures may affect fuel economy in four different ways:

- reduction of engine speed
- change of engine configuration (i.e. bore/stroke ratio and number of cylinders)
- changes to the combustion process
- changes to the vehicle weight.

Some approaches may have no significant effect on fuel economy, such as redesigned pistons.

Considering these four broad areas of engine noise control, the specific measures may be grouped as follows. Note that in some cases there is an interaction between measures. For example turbocharging may beneficially affect combustion noise, adversely affect mechanical noise and also permit lower engine speeds (through higher gearing).

a) Reduction of Engine Speed

- by increasing the overall gear ratio, thus sacrificing performance
- by increasing selected gear ratios to satisfy current regulatory test procedures
- by increasing the overall gear ratio and turbocharging thus restoring performance
- by increasing engine size and reducing rated speed, without change in performance

A computer prediction of the effect of final drive ratio (i.e. engine speed/load) on the Federal Test Procedure fuel consumption of 1100 kg gasoline car is shown in Figure 27.

b) Change of Engine Configuration

- reduce bore and increase stroke to maintain engine swept volume (no change in engine speed)[5]. (Test results from a single cylinder research engine showing the effect on performance and fuel consumption are given in Figure 28).
- reduce bore and stroke and increase number of cylinders (same swept volume and speed).
- reduce engine size and turbocharge to restore performance.

c) Measures Which Affect the Combustion Process

- injection/ignition timing
- compression ratio (see Figure 29 for effect on fuel economy)

- spark energy (gasoline engine)
- no. of spark sources (gasoline engine)
- turbocharging
- fumigation (diesel)
- EGR
- Pilot injection (diesel)
- Air/Fuel ratio

d) Measures Which Only Increase the Vehicle Weight

- vibration damping and mass loading
- close fitting shields
- structural changes
- tunnel enclosures
- total enclosures

A computer prediction of the effect of vehicle weight on fuel economy is given in Figure 30 for a diesel and a gasoline passenger car.

Measures for noise control which have no significant effect on fuel economy or weight are:

- piston design changes (expansion controlled)
- isolation of engine components

An attempt to quantify the noise reduction likely to be achieved and the corresponding effect on fuel economy, weight and cost for each of the above measures has been made in Tables 1, 2 and 3. Table 1 considers the case of a diesel engine passenger car and Table 2 a gasoline engine car. (Table 3 considers measures common to both vehicles). The various noise reduction systems are listed individually and the noise reduction associated with that system estimated. The baseline car considered is 1100 curb weight, powered by a 4 cylinder, 2 litre (122 $in^3$) swept volume engine of 70 kW for the gasoline engine and 50 kW for the diesel. For all the noise control measures discussed, the engines are naturally aspirated and in 1979 U.S. Federal emission build, i.e. 1.5 g/mile HC, 15 g/mile CO, 2.0 g/mile NOx. To meet these limits, the gasoline engine is likely to have an oxidation catalyst, EGR, ignition retard, a 'low' compression ratio ($\sim$8.5) and fuel injection. The diesel engine will be a standard Comet V, optimized for performance and economy (i.e. with the injection timing approxiamtely 5° cr advanced beyond that normally recommended specifically for a low noise build engine). Such a performance optimized Comet V is capable of comfortably meeting the 2.0 g/mile NOx limit (see results for the Peugeot 504 GLD in Appendix 1). Both vehicles are assumed to have four ratio manual transmissions with an overall top gear ratio of around 17 km/h per 10 rev/s engine speed. When referred to the proposed 1981 Federal emission standard (i.e.: 0.41/3.4/1.0 g/mile HC/CO/NOx), and the proposed 1981 particulate standard (0.6 g/mile), most of the control measures considered are still valid, as are their expected effects on fuel economy, cost and weight. The only areas likely to be affected are those where the combustion process is changed. For example, a light duty diesel in a passenger car to 1979 Federal limits will be able to operate close to performance/economy optimized injection timings but will need retard to meet the 1981 Nox limit. Thus for this latter build of engine, timing retard is not an effective noise control measure (Ref. Figure 9), whereas it can be for present day engines.

Where percent changes, noise reductions and mile/US gal fuel economy changes are quoted these are in all cases related to the respective 1979 baseline car. For the noise reduction figure, this is based on a high engine speed, low gear, maximum acceleration drive-by test (e.g. 70/157/EEC, SAE J986b). The fuel economy figures are based on the Federal Test Procedure (Composite) and expressed in mile/US gal. It must be appreciated that because the two regulatory procedures are very different in their engine operational requirements (the drive-by test being a full load, high speed condition, the FTP being essentially part load, low speed with only occasional high speed sections and with no high speed full load conditions), it is possible that some measures to reduce the drive-by noise may have no effect on the FTP composite fuel economy and vice versa. Where such cases occur, these will be discussed with reference to the table supporting notes.

A typical baseline composite fuel economy of 23 mile/US gal has been assumed for the gasoline car and 30 mile/US gas for the diesel car. (Figures obtained from 1978 Gas Mileage Guide and are averages of selected passenger cars in the 2000 - 2400 lb weight class). The cost change is a best estimate assuming the modification employed is mass produced and ignoring any initial major development costs (for example, although changing to a 5 ratio from a 4 ratio gearbox would incur considerable re-tooling and development costs, once in production would mean, only a very small cost difference).

The values and changes quoted are supported by experimental data where possible but where such data cannot be presented (either through non-existence or confidentiality) then a best estimate has been made based on engineering experience with such engines and vehicles. Such trade-off data is impossible to quantify precisely as there is so much variation from engine to engine and vehicle to vehicle. An attempt has been made to realistically generalize where possible. In some cases a parameter change may actually, in isolation, cause a noise increase. These are clearly shown, and have been left in the tables for comparison purposes.

The tables are basically self explanatory but a number of notes have been added to clarify certain areas. The tables essentially form the main framework of this report in conjunction with supporting notes.

## 4.2 EXHAUST NOISE

See Table 3.

## 4.3 INTAKE NOISE

See Table 3.

## 5. REFERENCES

1. Ricardo Report. Calspan High Speed Engine Study – Summary Report Covering Test Work on the SAAB 99 and Peugeot 504D Vehicles DP.77/1158.

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10* W.M. Scott. Noise of Small Indirect Injection Diesel Engines. SAE 730242.

11* Internal Ricardo Report. Effect of Exhaust Gas Recirculation on Emissions from Opel 2100D Diesel Car. DP.78/339.

13* Internal Ricardo Report. Development of the Turbocharged Opel 2100D for Passenger Car Application. DP.20818.

* Not specifically referred to in text.

# APPENDIX A

## BRIEF SPECIFICATION OF SAAB AND PEUGEOT VEHICLES

| | SAAB 99GL | PEUGEOT 504 GLD |
|---|---|---|
| Engine type | Gasoline, naturally aspirated | Indirect injection diesel Comet V, naturally aspirated |
| Engine capacity | 1.985ℓ (121 $in^3$) | 2.304ℓ (141 $in^3$) |
| Nominal max power* | 82 kW (110 bhp) | 52 kW (70 bhp) |
| Max power as tested with gearbox fitted | 72 kW (96 bhp) | 48 kW (64 bhp) |
| Nominal max engine speed | 92 rev/s (5500 rev/sec) | 75 rev/s (4500 rev/min) |
| Weight as tested for drive-by tests | 1260 kg (2780 lb) | 1480 kg (3260 lb) |
| Weight as tested for performance tests | 1415 kg (3120 lb) | 1560 kg (3440 lb) |
| Emission build standard | California 1976 (no catalyst) | California 1976 |
| Fuel Injection system | Bosch (Continuous Injection) | CAV (Rotodiesel) |
| Transmission | 4 forward ratios, manual | 4 forward ratios, manual |
| Overall gear ratio (km/h per 10 rev/s engine speed) | 18.3 | 17.5 |

<u>Vehicle Performance</u> (Results from Ricardo Tests)

| | | |
|---|---|---|
| 0→80 km/h | 11.2 s | 16.0 s |
| Max speed (top gear) | 170 km/h | 140 km/h |
| Mile/US Gal (Urban Cycle) | 17.3 (19/23 †) | 28.6 (28/30 †) |

BRIEF SPECIFICATON OF SAAB AND PEUGEOT VEHICLES (Con't)

Exhaust Emission Test Results

(LA4 CVS Test)

| | | |
|---|---|---|
| HC | 0.91 | 0.59 |
| NOx | 1.5 | 1.1 |
| CO | 7.1 | 1.4 |

* Manufacturer's figure

† Urban/Combined figures from 1978 Gas Mileage Guide

TABLE 1. DIESEL PASSENGER CAR NOISE REDUCTION MEASURES AND FUEL ECONOMY, WEIGHT AND COST TRADE-OFFS

(reference numbers refer to supporting notes following the tables)

| Noise Reduction Measure | Noise [1] Reduction (dBA) | Vehicle Weight Change (%) | Fuel [2] Economy Change (mile/US gal) | Cost [3] Increase (%) | Comments |
|---|---|---|---|---|---|
| <u>Reduce engine speed by 10%</u> | | | | | |
| a) by changing final drive ratio | 1 to 2 | 0 | +1 | 0 | performance loss [4] |
| b) by changing intermediate ratio gear | 1 to 2 | 0 | <+1 | 0 | some performamce loss [5] |
| c) increase final drive ratio and increase engine swept vol. | 0 | +1 to +2 | <+1 | ∿ +1 | performance restored [6] |
| d) increase final drive ratio and turbocharge | 1 to 2 | +2 to +3 | +1 | ∿ +7 | to restore performance [7] |
| <u>Engine size/configuration change</u> | | | | | |
| a) Reduce bore:stroke ratio | 1 to 2 | < +1 | -1 | 0 | same swept vol. and speed [8] |
| b) Reduce bore and stroke - increase no. of cyls. from 4 to 6 | 1 to 2 | +2 | ∿-1 | +4 | [9] |
| c) Reduce swept vol. and turbocharge to restore power | 1 to 2 | 0 to -1 | +1 to +2 | ∿ +5 | swept vol. from 2ℓ to 1.5ℓ [10] |
| <u>Combustion Process Changes</u> | | | | | |
| a) retard injection by 5° | 1 | 0 | -1 to 0 | 0 | [11] |
| b) increase compression ratio by 1 | < 1 | 0 | -1 | <+1 | production tolerance difficulties [11] |
| c) turbocharge (power increase by 35%) | increase by 1 to 2 | +2 to +3 | -2 | ∿ +7 | performance increase [12] |
| d) fumigation | <1 | <+1 | -1 | < +1 | [13] |
| e) EGR (10% at full load) | 0 to 1 | <+1 | 0 | < +1 | |
| f) pilot injection | 0 to 1 | <+1 | -1 | < +1 | [14] |

TABLE 2. GASOLINE PASSENGER CAR NOISE REDUCTION MEASURES AND FUEL ECONOMY, WEIGHT AND COST TRADE-OFFS

(reference numbers refer to supporting notes following the tables)

| Noise Reduction Measure | Noise Reduction (dBA) | Vehicle Weight Change (%) | Fuel Economy Change (mile/US gal) | Cost Increase (%) | Comments |
|---|---|---|---|---|---|
| Reduce engine speed by 10% | | | | | |
| a) by changing final drive ratio | 1 to 2 | 0 | <+1 | 0 | performance loss 4 |
| b) by changing intermediate ratio | 1 to 2 | 0 | <+1 | 0 | some performance loss 5 |
| c) raise final drive ratio and increase engine swept vol. | 0 | +1 | ~0 | <+1 | performance restored 6 |
| d) raise final drive ratio and turbocharge | <1 | +2 to +3 | <+1 | ~+7 | to restore performance 7 |
| Engine size/configuration change | | | | | |
| a) Reduce bore:stroke ratio | 1 to 2 | <+1 | <+1 | 0 | const. speed & swept vol. 8 |
| b) Reduce bore and stroke - increase no. of cyls. | 1 to 2 | +2 | -1 | +2 | increase cyls. from 4 to 6 9 |
| c) Reduce swept vol. and turbocharge | 1 to 2 | 0 to -1 | +2 | ~+5 | from 2ℓ to 1.5ℓ 10 |
| Combustion Process changes | | | | | |
| a) retard ignition at high speed | 0 to 1 | 0 | 0 | 0 | 15 |
| b) increase comp. ratio by 1 to 9.5 | (<1 increase) | 0 | +1 to +2 | 0 | performance increase - noise increased in WOT drive-by |
| c) increase comp. ratio to 12 | (~1 increase) | 0 | +2 | <+1 | HRCC concept - increased power 16 |
| d) increase c.r. to 12, reduce swept vol. | ~1 | -1 | +2½ | <+1 | to equal original performance |
| e) increase spark energy | 0 | 0 | 0 or <+1 | <+1 | see 17 |
| f) increase no. of spark plugs to 2 | 0 | 0 | <+1 | <+1 | 18 |
| g) turbocharge | (increase by 1 to 2) | +2 to +3 | -1 to -2 | ~+7 | performance increase |
| h) EGR (10% at WOT) | <1 | <+1 | 0 | <+1 | 19 |
| i) A/F ratio (less at high speed) | <1 | 0 | 0 | 0 | 19 |

TABLE 2. GASOLINE PASSENGER CAR NOISE REDUCTION MEASURES AND FUEL ECONOMY, WEIGHT AND COST TRADE-OFFS (CONTINUED)

(reference numbers refer to supporting notes following the tables)

| Noise Reduction Measure | Noise Reduction (dBA) | Vehicle Weight Change (%) | Fuel Economy Change (mile/US gal) | Cost Increase (%) | Comments |
|---|---|---|---|---|---|
| <u>Reduce engine speed by 10%</u> | | | | | |
| a) by changing final drive ratio | 1 to 2 | 0 | <+1 | 0 | performance loss[4] |
| b) by changing intermediate ratio | 1 to 2 | 0 | <+1 | 0 | some performance loss[5] |
| c) raise final drive ratio and increase engine swept vol. | 0 | +1 | ∼0 | <+1 | performance restored 6 |
| d) raise final drive ratio and turbocharge | <1 | +2 to +3 | <+1 | ∼+7 | to restore performance 7 |
| <u>Engine size/configuration change</u> | | | | | |
| a) Reduce bore:stroke ratio | 1 to 2 | <+1 | <+1 | 0 | const. speed & swept vol. 8 |
| b) Reduce bore and stroke - increase no. of cyls. | 1 to 2 | +2 | -1 | +2 | increase cyls. from 4 to 6 9 |
| c) Reduce swept vol. and turbocharge | 1 to 2 | 0 to -1 | +2 | ∼+5 | from 2ℓ to 1.5ℓ[10] |
| <u>Combustion Process changes</u> | | | | | |
| a) retard ignition at high speed | 0 to 1 | 0 | 0 | 0 | 15 |
| b) increase comp. ratio by 1 to 9.5 | (<1 increase) | 0 | +1 to +2 | 0 | performance increase - noise increased in WOT drive-by |
| c) increase comp. ratio to 12 | (∼1 increase) | 0 | +2 | <+1 | HRCC concept - increased power[16] |
| d) increase c.r. to 12, reduce swept vol. | ∼1 | -1 | +2½ | <+1 | to equal original performance |
| e) increase spark energy | 0 | 0 | 0 or <+1 | <+1 | see [17] |
| f) increase no. of spark plugs to 2 | 0 | 0 | <+1 | <+1 | 18 |
| g) turbocharge | (increase by 1 to 2) | +2 to +3 | -1 to -2 | ∼+7 | performance increase |
| h) EGR (10% at WOT) | <1 | <+1 | 0 | <+1 | 19 |
| i) A/F ratio (less at high speed) | <1 | 0 | 0 | 0 | 19 |

TABLE 3. PASSENGER CAR NOISE REDUCTION MEASURES AND FUEL ECONOMY, WEIGHT AND COST TRADE-OFFS (COMMON TO BOTH DIESEL AND GASOLINE ENGINES)
(reference numbers refer to supporting notes following the tables)

| Noise Reduction Measure | Noise Reduction (dBA) | Vehicle Weight Change (%) | Fuel Economy Change (mile/US gal) | Cost Increase (%) | Comments |
|---|---|---|---|---|---|
| Damping/Mass Loading components | 0 to 2 | <+1 | 0 | <+1 | 20 |
| Isolation of engine components | 0 to 2 | <+1 | 0 | <+1 | 20 |
| Comprehensive shielding | 1 to 3 | +1 to +2 | ∿-½ | +1 to +2 | 21 |
| Enclosure (tunnel type) | 4 to 6 | +1 to +3 | ∿-½ | +2 to +3 | 21 |
| Major engine structure re-design | 4 to 8 | 0 | 0 | ∿+5 | 22 |
| <u>Gas Flow Noise Reduction</u> | | | | | |
| Improved Exhaust Muffler | 1-2 | +1 to +2 | -1 to -2 | +1 to +2 | 23 |
| Improved Intake Muffler | 0-1 | ∿+1 | ∿-1 | ∿+1 | 24 |
| <u>Other</u> | | | | | |
| a) re-designed piston (close clearance) | 1 to 2 | 0 | 0 | 0 to 1 | |

## TABLE REFERENCE NOTES

1. Noise reduction is on basis of "wide-open-throttle" maximum acceleration drive-by procedure (e.g. SAE J986b, 70/157/EEC).

2. Fuel economy change is on basis of Federal Test Procedure and is combined result of urban and highway tests.

3. The cost increase is a best estimate assuming the measure is implemented on a mass production basis, and is a percentage of the baseline vehicle cost (assumed at 1979 US prices to be approximately $6000).

4. Due to the type of noise test procedure, the actual noise reduction is not simply a function of the reduced engine speed, but is also dependent on the particular torque characteristics of the engine and hence acceleration of the vehicle. On a drive-by test the noise level is largely dependent on engine speed. Thus, if in increasing the gear ratio the acceleration of the vehicle is reduced then the engine speed achieved at the center of the zone would be correspondingly less than 10 percent gear ratio change alone. A greater noise reduction than that expected by a straight 10 percent speed reduction would therefore be achieved. This is one of the problems in using a 'wide-open-throttle' noise test procedure as a reference but as this is currently universally used for legislative purposes it is the only real criterion which can be used. See Figure 27 for the predicted effect of gear ratio on fuel economy.

5. Note 4 applies here. Increasing a particular intermediate gear ratio is a means of achieving lower pass-by test noise levels but does not inherently make the engine any quieter over its operating range.

6. In this case, the engine swept volume (and hence power) has been increased to restore the vehicle acceleration performance to baseline. The acceleration of the vehicle in the zone is

thus assumed equal to baseline and thus the noise trade off is a function of the reduced engine speed v increased bore size (Ref. Figure 12). For this generalized case it is reasonable to assume that one balances the other as a first order approximation.

7. Note 6 also applies but in this case the engine bore is not increased and so there will be a net reduction in the drive-by noise, largely as a result of the increased gearing and also as a result of the possibly marginally reduced combustion noise contribution. The mechanical noise may well have increased, however, through increased piston slap as a result of higher $P_{max}$ levels caused by turbocharging. For this mild degree of turbocharging the compression ratio is assumed unchanged from baseline. The net effect of turbocharging and gear ratio is unlikely to change significantly the fuel consumption from baseline (on a LA4 basis).

8. No change in engine speed is assumed. A possible bore/stroke change is from 90Ø x 79 to 82Ø x 95 (dimensions in nm). This noise reduction measure (of reducing the bore) has been considered as independent of any other changes for the sake of showing trade-offs. In reality, however, if a small bore version of the engine was chosen then the speed would possibly by reduced (to avoid increasing the mean piston speed) and the stroke increased beyond that required for a 2 litre swept volume to compensate for the power loss associated with the lower speed.

9. In reducing the bore and stroke and increasing the number of cylinders (keeping engine speed and swept volume constant) the engine weight will increase and also the fuel economy and power are likely to slightly worsen as a result of increased frictional losses and increased surface/volume ratio of the combustion chamber. In practice, the engine speed would be increased to compensate for this power loss but for the purposes of isolating individual effects, it is assumed

the speed has not been changed. The noise reduction results from the reduced bore. The number of cylinders for a given bore size has little effect on radiated noise, within reason, for a multi-cylinder engine of conventional design (Ref. 3).

10. The net noise reduction in this case arises mainly from the balance between the effects of the marginally reduced combustion noise contribution by turbocharging, and the marginally increased mechanical noise as a result of increased $P_{max}$ through turbocharging. The engine speed and vehicle performance have been unchanged.

11. At high speeds, the noise associated with light duty IDI diesel engines is almost always more mechanical in origin rather than combustion. Thus any measures which affect the combustion process on this type of engine are likely on average to have only a small effect on noise. As the hypothetical engine being considered is a performance optimized Comet, however, 5° injection retard may be advantageous in slightly reducing combustion noise but at some expense of fuel economy (Ref. 3). This measure only applies to 1979 build cars, as to meet the proposed 1981 emission limits the diesel engine will almost certainly be close to the limit of its injection retard (compatible with low NOx/low HC trade-off, acceptable economy and smoke).

12. Turbocharging will increase mechanical noise slightly and reduce the combustion noise marginally (see Note 10). The overall effect is not a noise reduction but a noise increase as a result of the higher engine speed in mid zone due to the increased performance.

13. As note 11; any effect on combustion noise is likely to be small. The performance loss associated with EGR may slightly reduce the drive-by noise level solely because of the reduced acceleration and thus engine speed in the zone.

14. As notes 11 and 13 with respect to combustion noise. Measures such as EGR, fumigation and pilot injection may have adverse

effects on particulate emissions and this would need to be very carefully considered in the light of future particulate emission regulations. EGR would therefore only be relevant in the light of current emission regulations. For 1981 proposed limits, the light duty diesel vehicle will almost certainly have EGR already. A curve showing the trade-off between NOx, particulates and FTP urban fuel economy for a diesel passenger car is shown in Figure 31. (Note: these actual test results are for a 1400 kg curb weight car).

15. The combustion noise contribution from light duty gasoline engines is normally small. (See Figures 7 and 8). Most of the noise radiated is of mechanical origin (e.g. piston slap). The effect of further timing retard (this hypothetical low emission build engine is already retarded) is difficult to generalize. If it is assumed that it is restricted to high engine speed only then there will probably be no effect on the FTP test economy and only a very small noise reduction on drive-by as a result of the reduced performance. This applies whether current or proposed '81 emission limits are considered. It is difficult to generalize about the effects of combustion chamber changes on emissions and economy, as by 1981 there will be considerable advances in emission control (and engine control) devices which will enable the '81 car to meet the more stringent emission limits but at fuel economies improved from current levels. The main point as far as noise control is concerned is that changes to the gasoline engine combustion (in '79 or '81 emission builds) are not likely to have any significant effect on overall noise and will give legislative drive-by noise reductions only by virtue of the worsened performance during the WOT acceleration.

16. The High Ratio Compact Chamber concept has not yet been evaluated sufficiently well to enable real figures to be tabled. If the system is compatible with low emissions and is durable then the fuel economy gains are likely to be significant. There is unlikely to be any major effect on

engine noise per se but under drive-by conditions the vehicle could reach higher speeds in the zone due to the improved performance and thus give rise to correspondingly higher noise levels.

17. If, by increasing the spark energy, it is possible to weaken the mixture strength further (i.e. increase the air: fuel ratio) then there may be a slight gain in fuel economy. Otherwise, simply increasing the spark energy will not improve economy.

18. Increasing the burn rate by using more than one spark source may result in an increased tolerance to EGR and hence permit less retarded spark timings. Thus there could be a small gain in fuel economy as a result of being able to operate less retarded (fuel economy being more sensitive to timing than EGR as a first order approximation).

19. Both EGR and A/F ratio effects (and also ignition timing) may be tailored for high speed and load to suit the drive-by test procedure without penalizing fuel consumption on the LA4 cycle.

20. Vibration damping of critical engine components is likely to be very marginal at reducing drive-by noise unless the pre-treated (isolated or damped) source was a very significant noise source. Any fuel economy penalty arises solely from increased weight and in reality is probably negligible.

21. The fuel economy penalty arises from increase in vehicle weight (Ref. Figure 30). The actual weight of shields or enclosure is very dependent on design and how much use is made of integral vehicle body work such as the hood and wheel arches.

22. The ultimate in re-designed low noise engines. With careful design optimization, the weight penalty could be zero. The cost of such engines is likely to be substantially higher than current conventional engines, however. The noise reductions stated assume the engine is a major source during a drive-by test. In reality, if the engine noise alone is reduced by

up to 8 dBA then other sources, intially second order, will become significant and thus an overall 8 dBA reduction is unlikely to be achieved unless these other sources are correspondingly attenuated. Considering a baseline drive-by level of 74 dBA (to SAE J986b), 69 dBA would be considered as a minimum realistic drive-by level for this type of vehicle. Thus -5 dBA overall should be able to be achieved from baseline possibly employing a combination of structural and enclosure techniques for the engine coupled with adequate exhaust and intake mufflers for the gas noise sources.

23. With the reduced engine noise, exhaust muffler design will become increasingly important. Improved muffler systems will almost certainly increase the vehicle weight (increased volume and number of individual mufflers) and cost. The weight increase will result in a small fuel economy penalty (probably less than 1 mile/US gal) but the greatest adverse effect on economy is likely to arise from the increased back pressure and the corresponding detrimental effect on engine efficiency. Turbocharged light duty engines are particularly sensitive to high exhaust back pressures.

24. As 21 except that the weight and breathing loss penalties are likely to be lower due to the fact that intake noise requires far less noise reduction (to a given level) than exhaust noise because of the very different gas excitation energies.

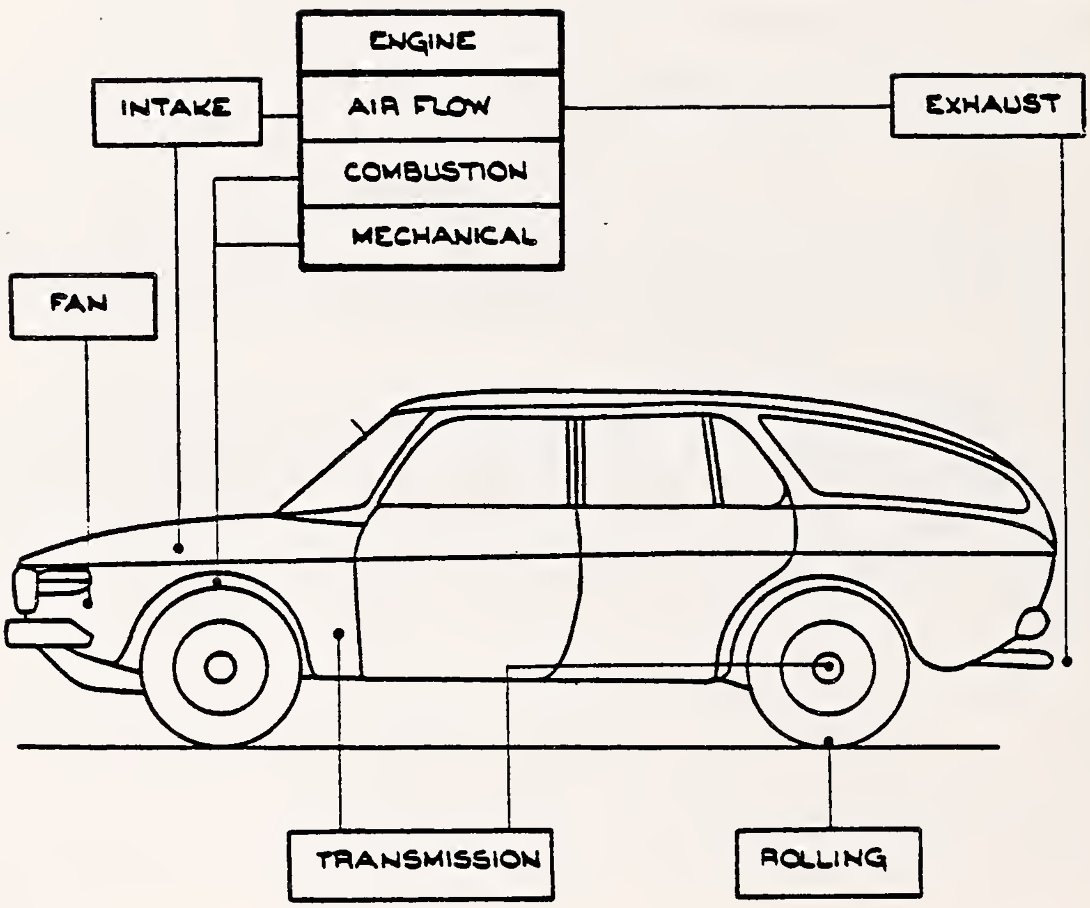


FIGURE 1. SIMPLE MODEL OF PRINCIPAL PASSENGER CAR NOISE SOURCES

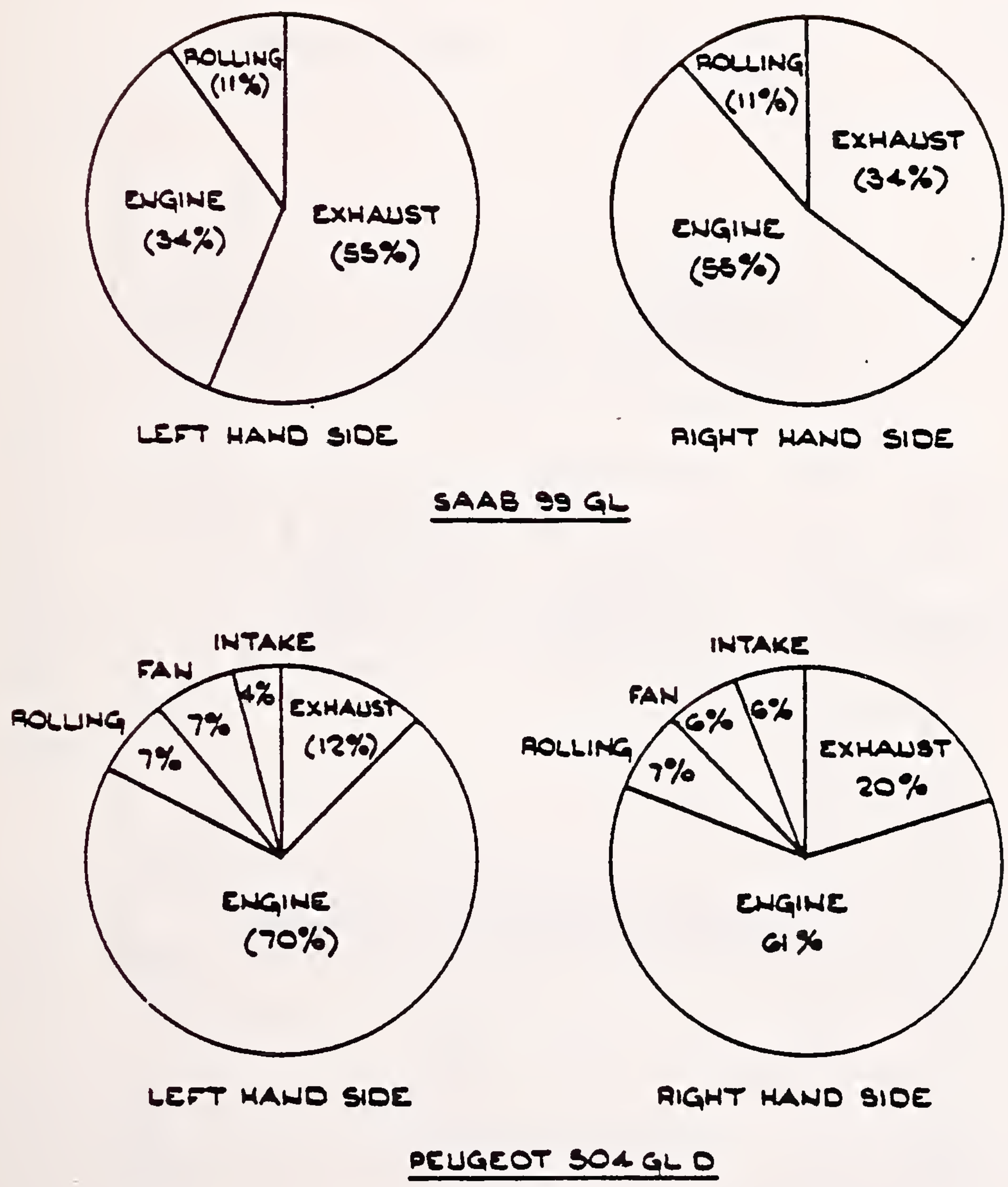


FIGURE 2. PASSENGER CAR NOISE SOURCE BREAKDOWNS FOR 2ND GEAR, 50 Km/h ENTRY SPEED, MAXIMUM ACCELERATION CONDITIONS

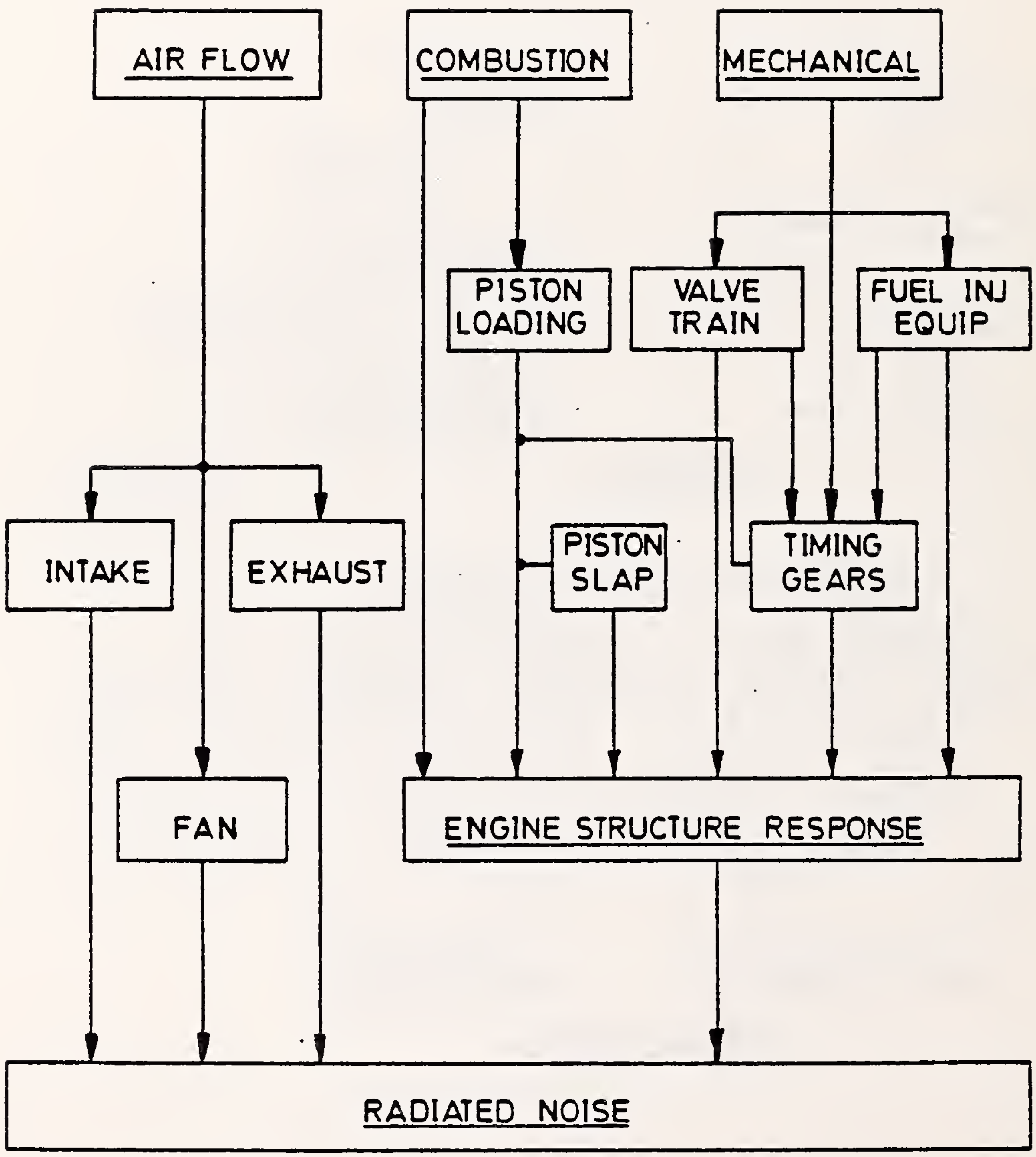


FIGURE 3. ENGINE NOISE GENERATION - A SIMPLE MODEL

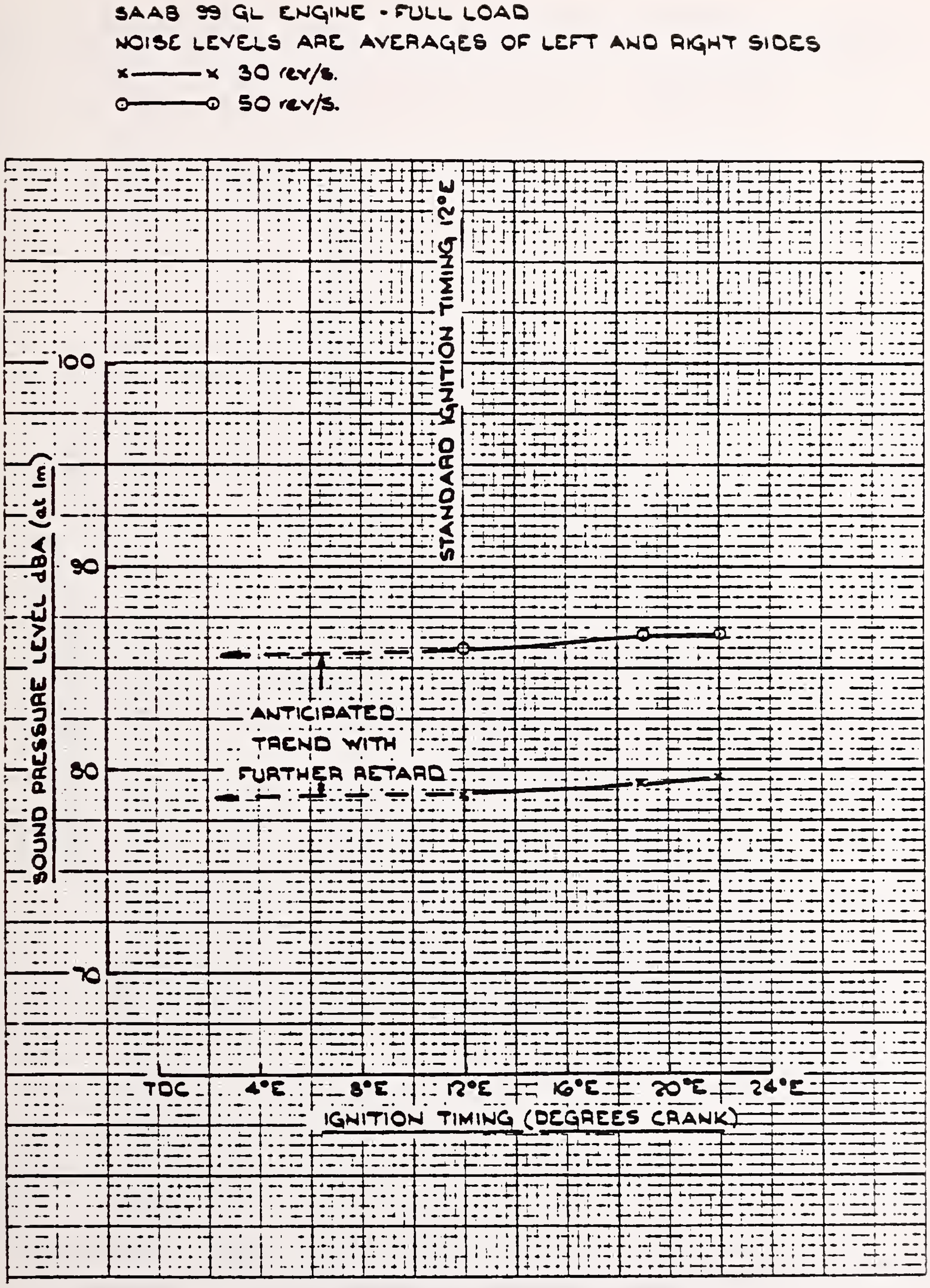


FIGURE 4. EFFECT OF IGNITION TIMING ON NOISE LEVEL

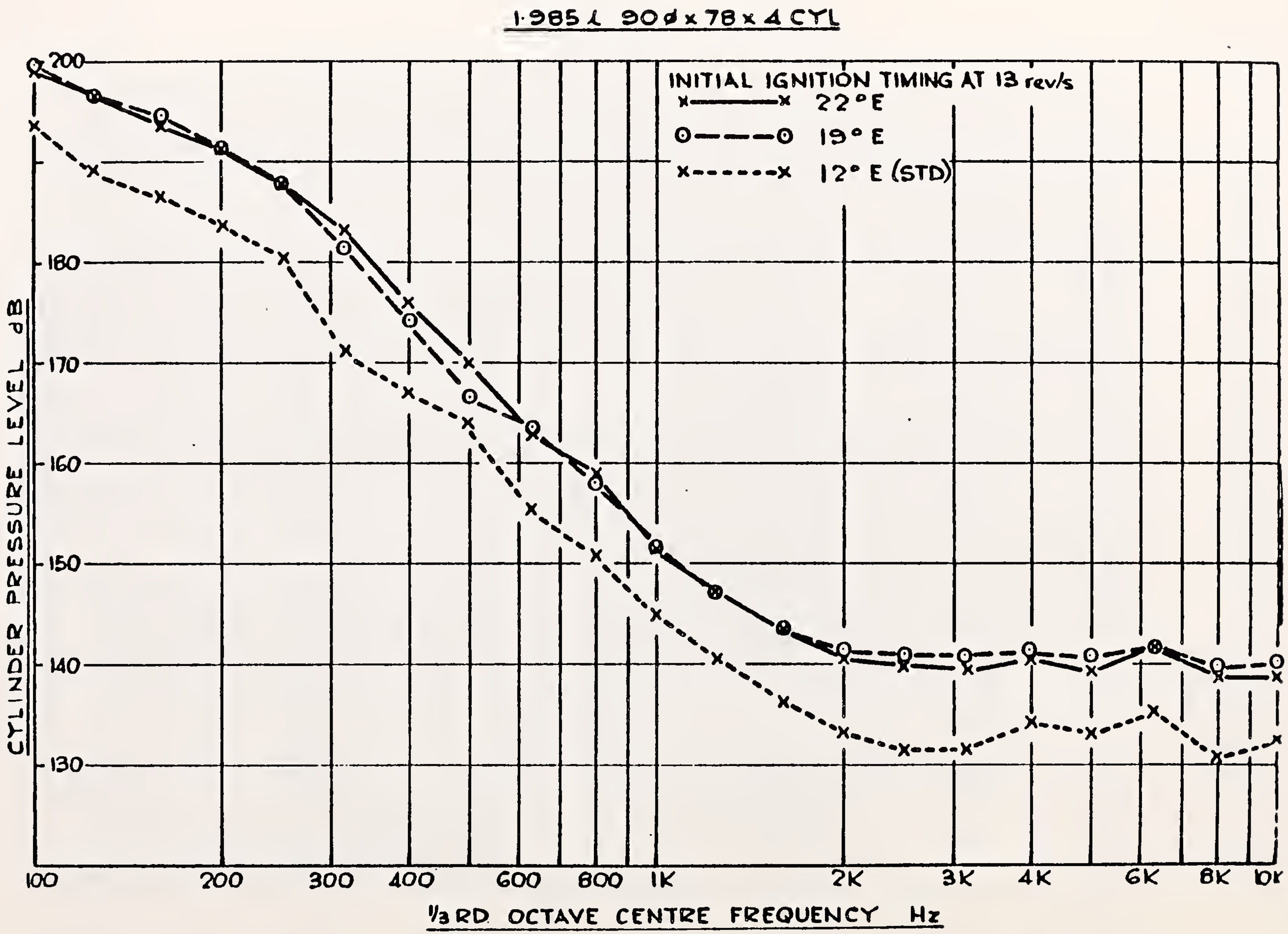


FIGURE 5. SAAB BI: EFFECT OF IGNITION TIMING ON CYLINDER PRESSURE LEVEL SPECTRUM AT 100% LOAD 30 rev/s

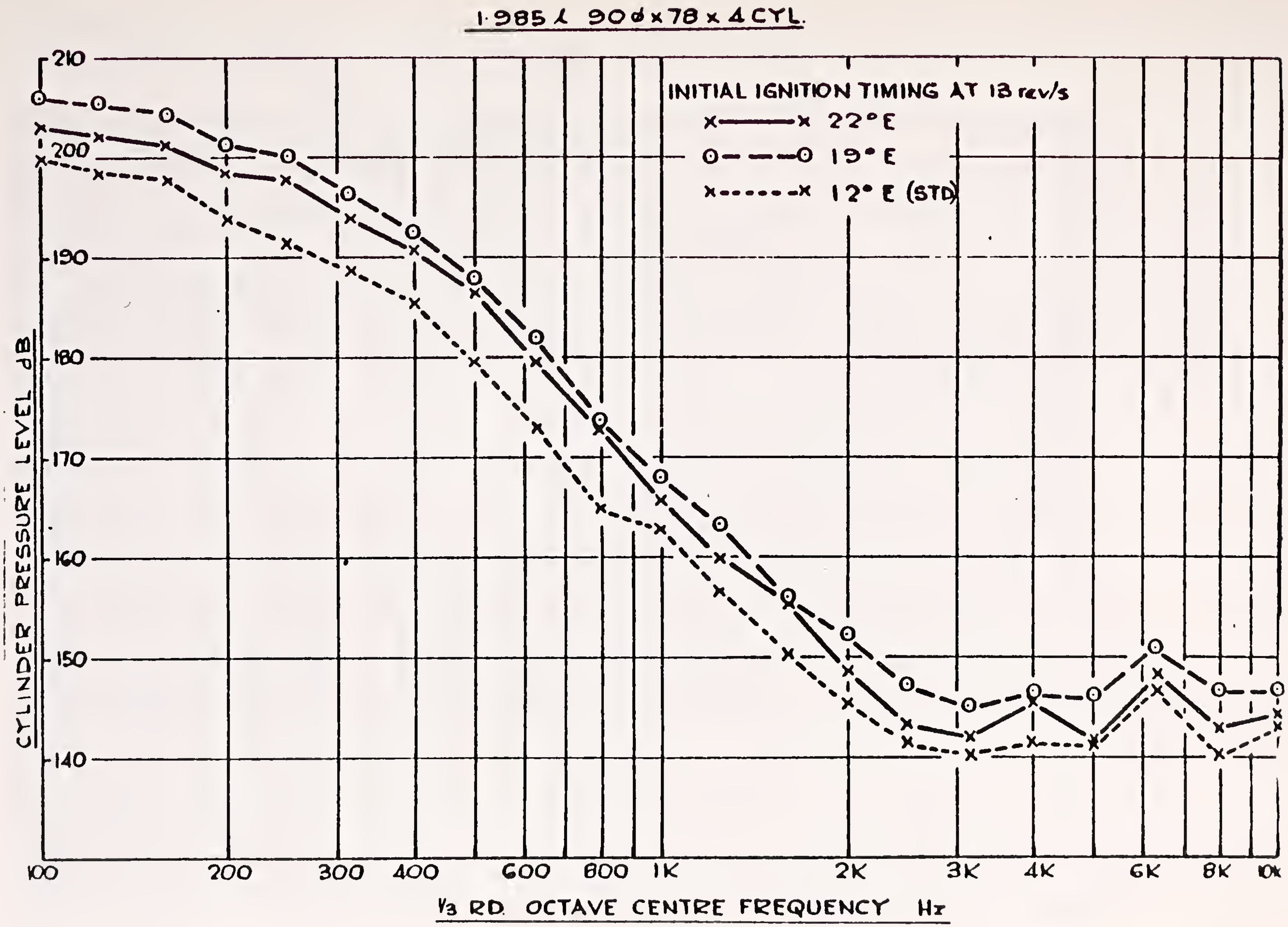


FIGURE 6. SAAB BI: EFFECT OF IGNITION TIMING ON CYLINDER PRESSURE LEVEL SPECTRUM AT 100% LOAD 50 rev/s

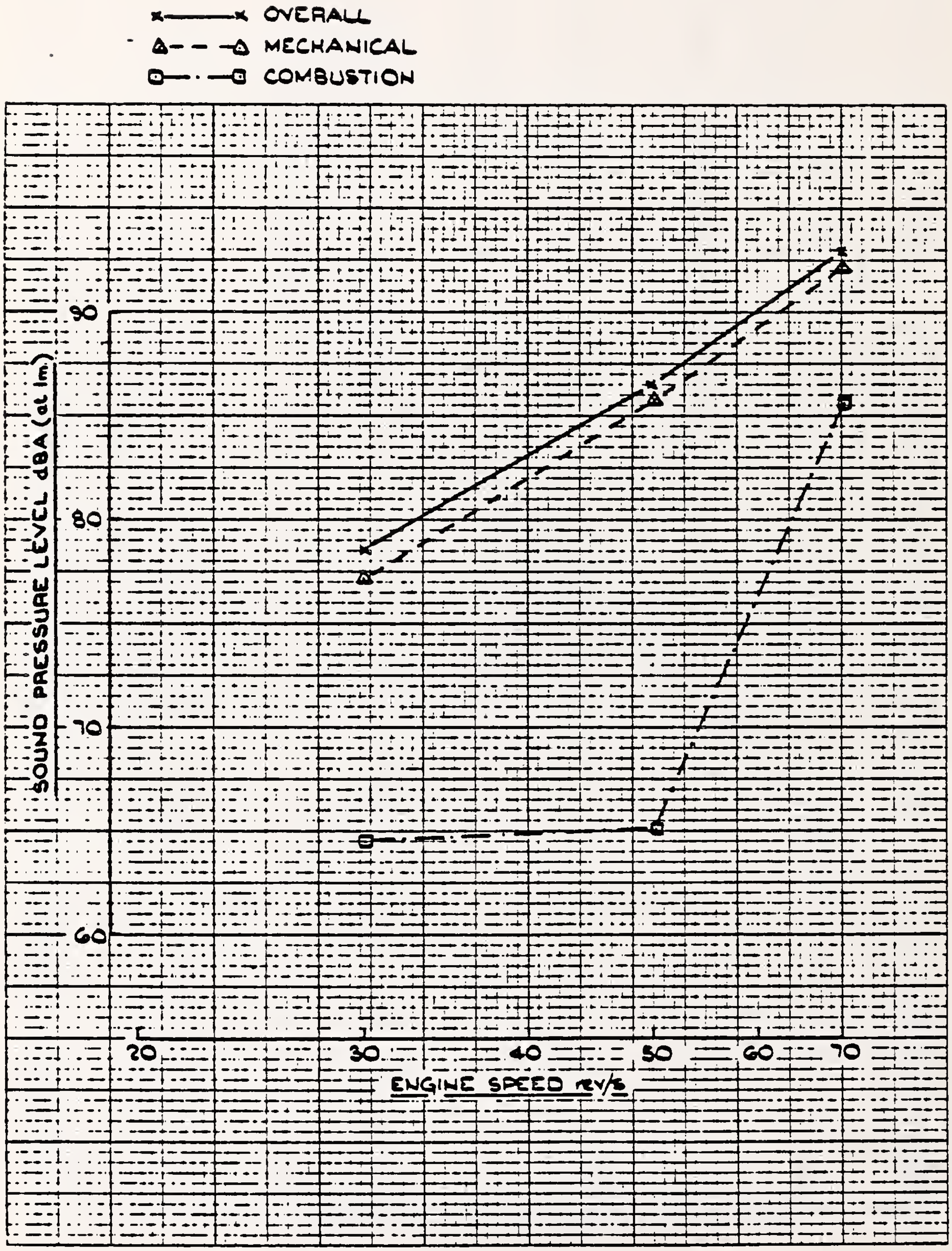


FIGURE 7. COMBUSTION/MECHANICAL NOISE BREAKDOWN FOR 2L GASOLINE ENGINE (SAAB BI) FULL LOAD (LHS)

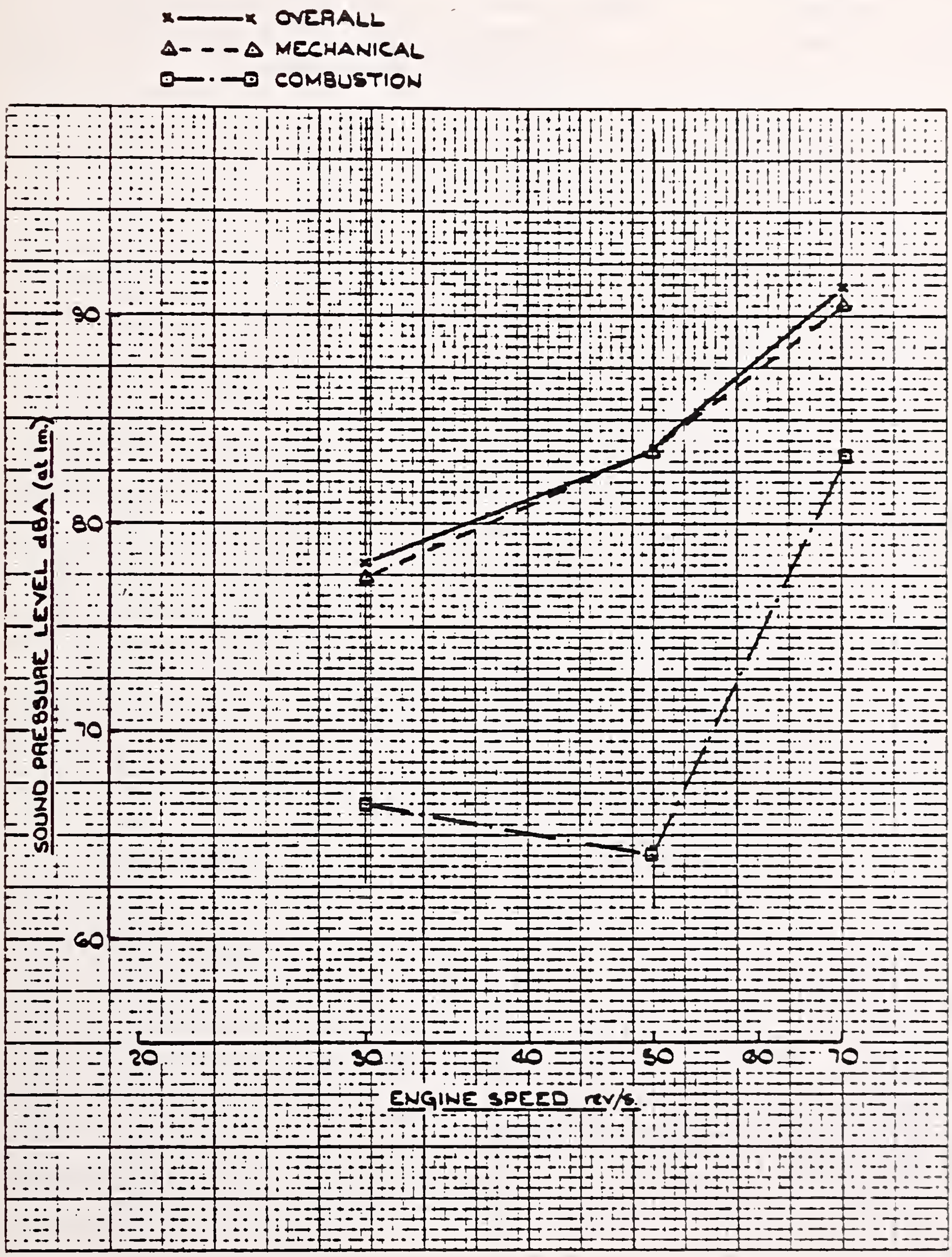


FIGURE 8. COMBUSTION/MECHANICAL NOISE BREAKDOWN FOR 2L GASOLINE ENGINE (SAAB BI) FULL LOAD (RHS)

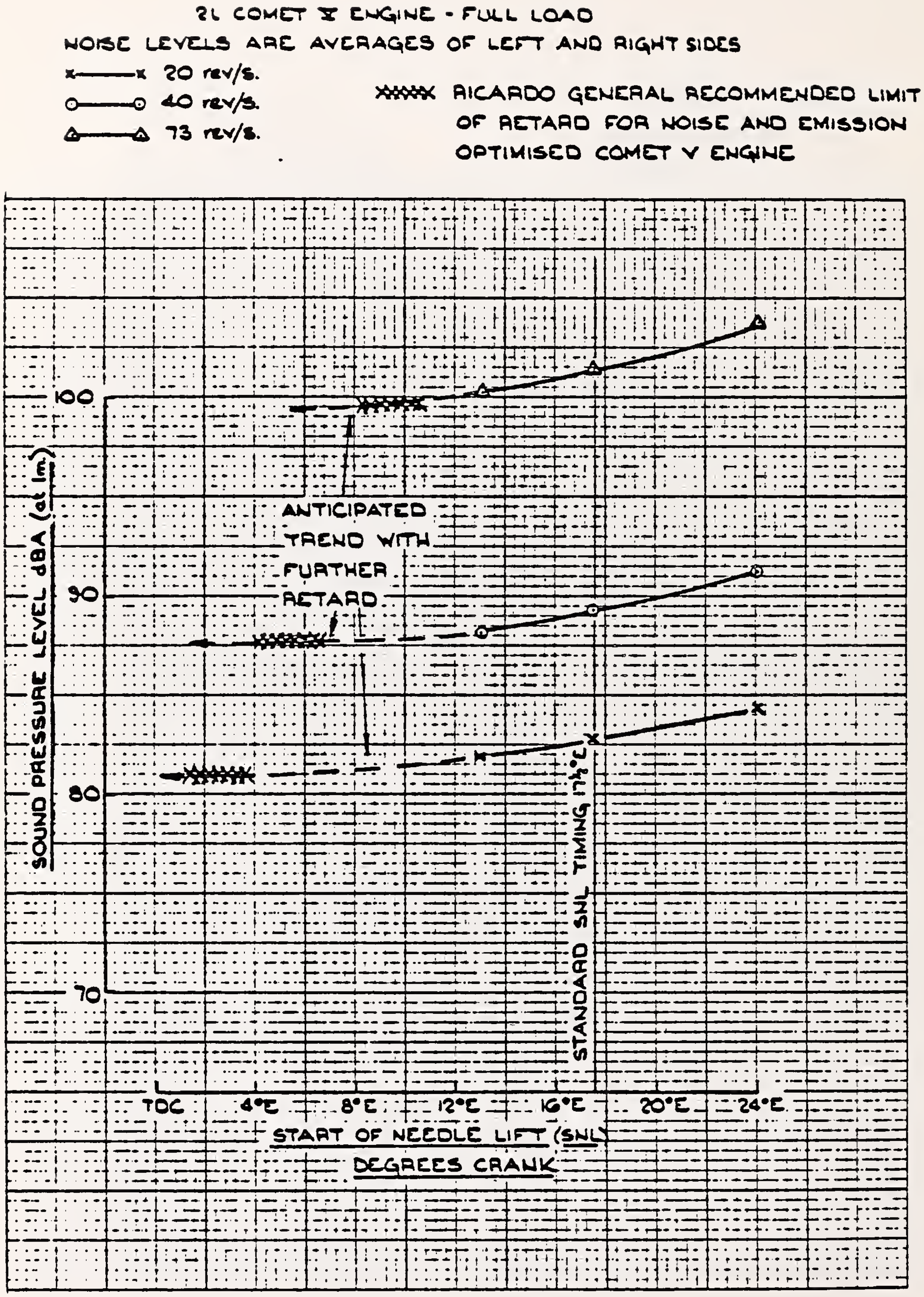


FIGURE 9. EFFECT OF INJECTION TIMING ON NOISE LEVEL

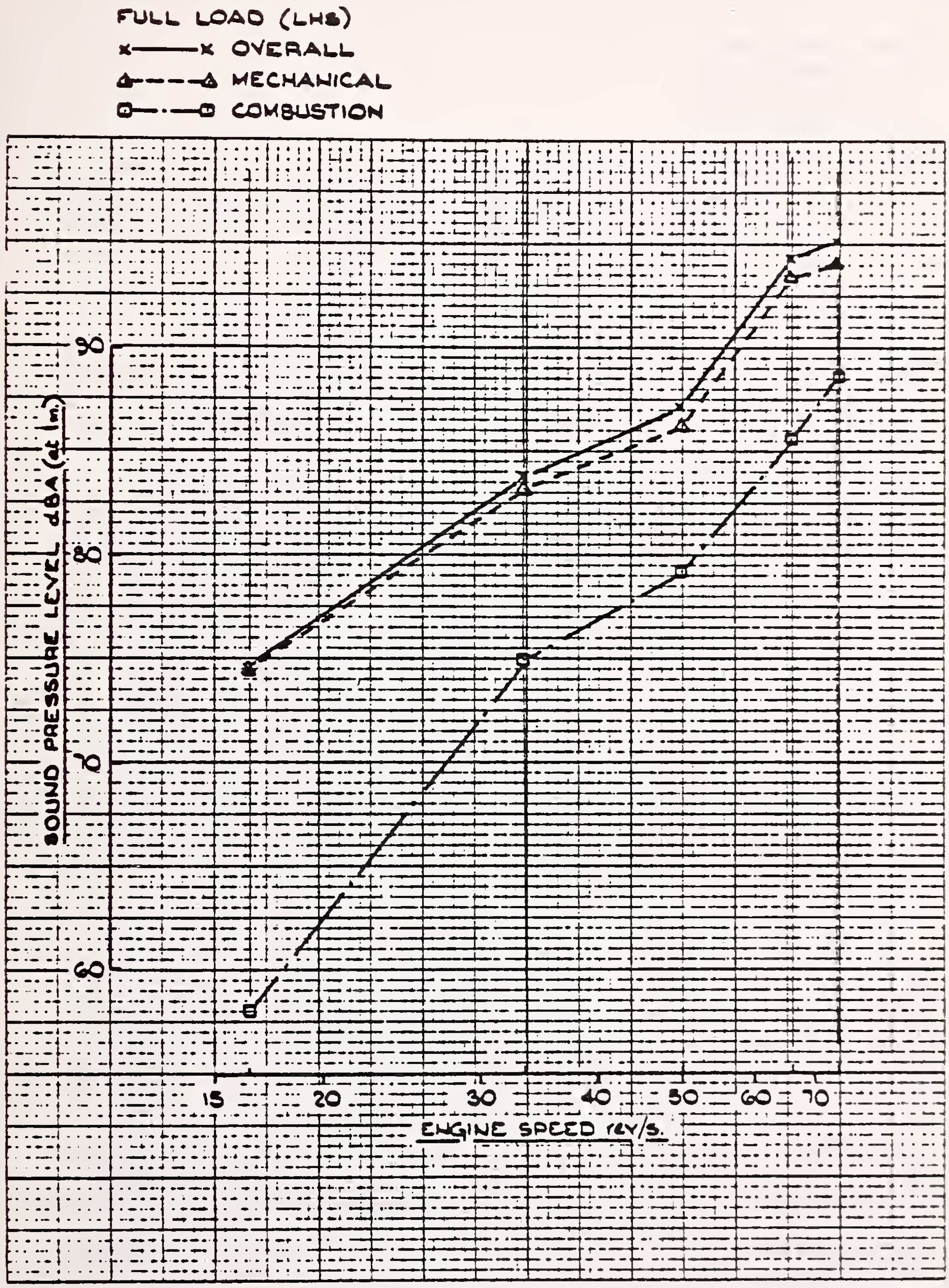


FIGURE 10. COMBUSTION/MECHANICAL NOISE BREAKDOWN FOR 2L COMET V DIESEL ENGINE

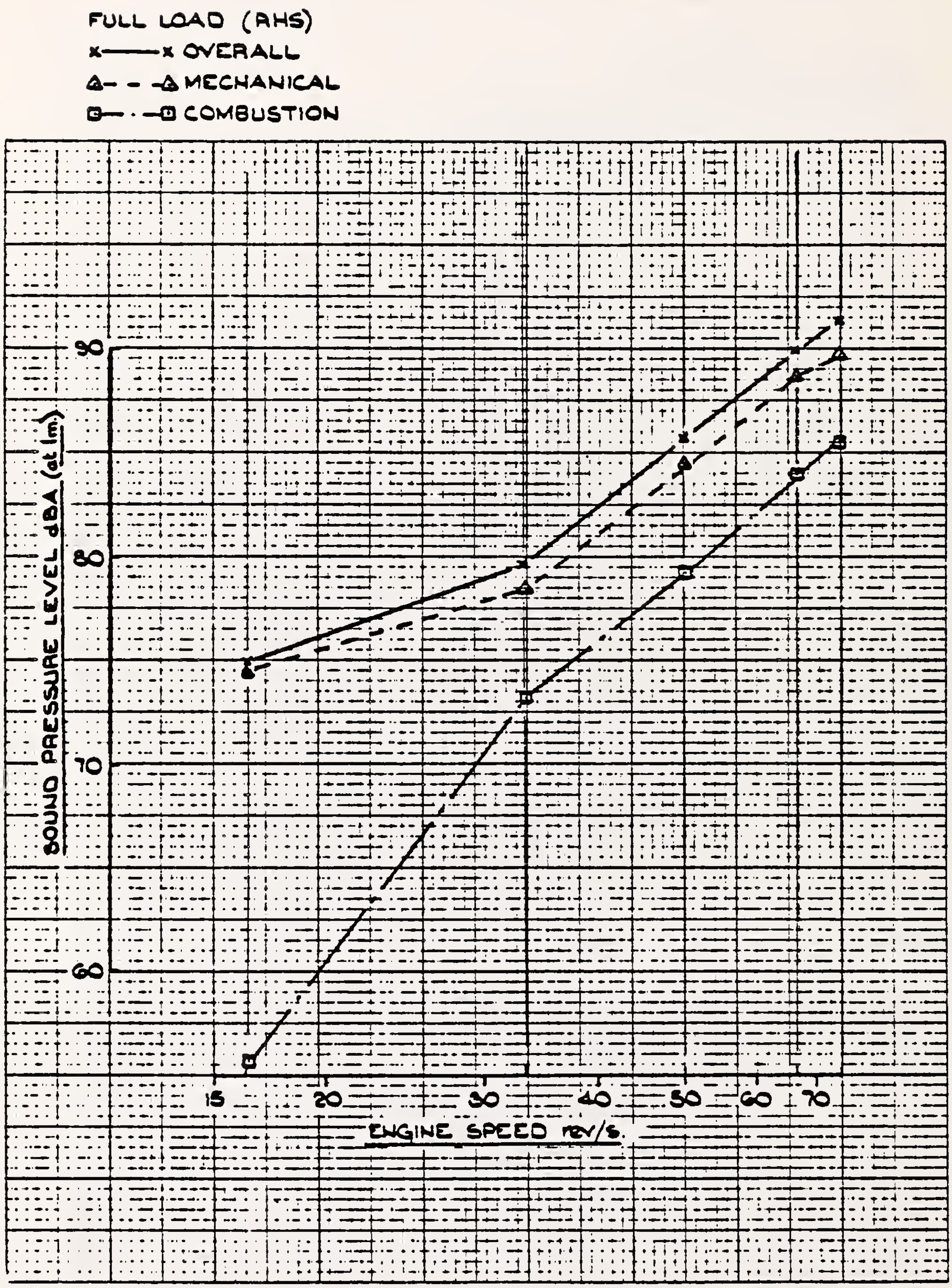


FIGURE 11. COMBUSTION/MECHANICAL NOISE BREAKDOWN FOR 2L COMET V DIESEL ENGINE

(AT RATED SPEED, FULL LOAD CONDITIONS)

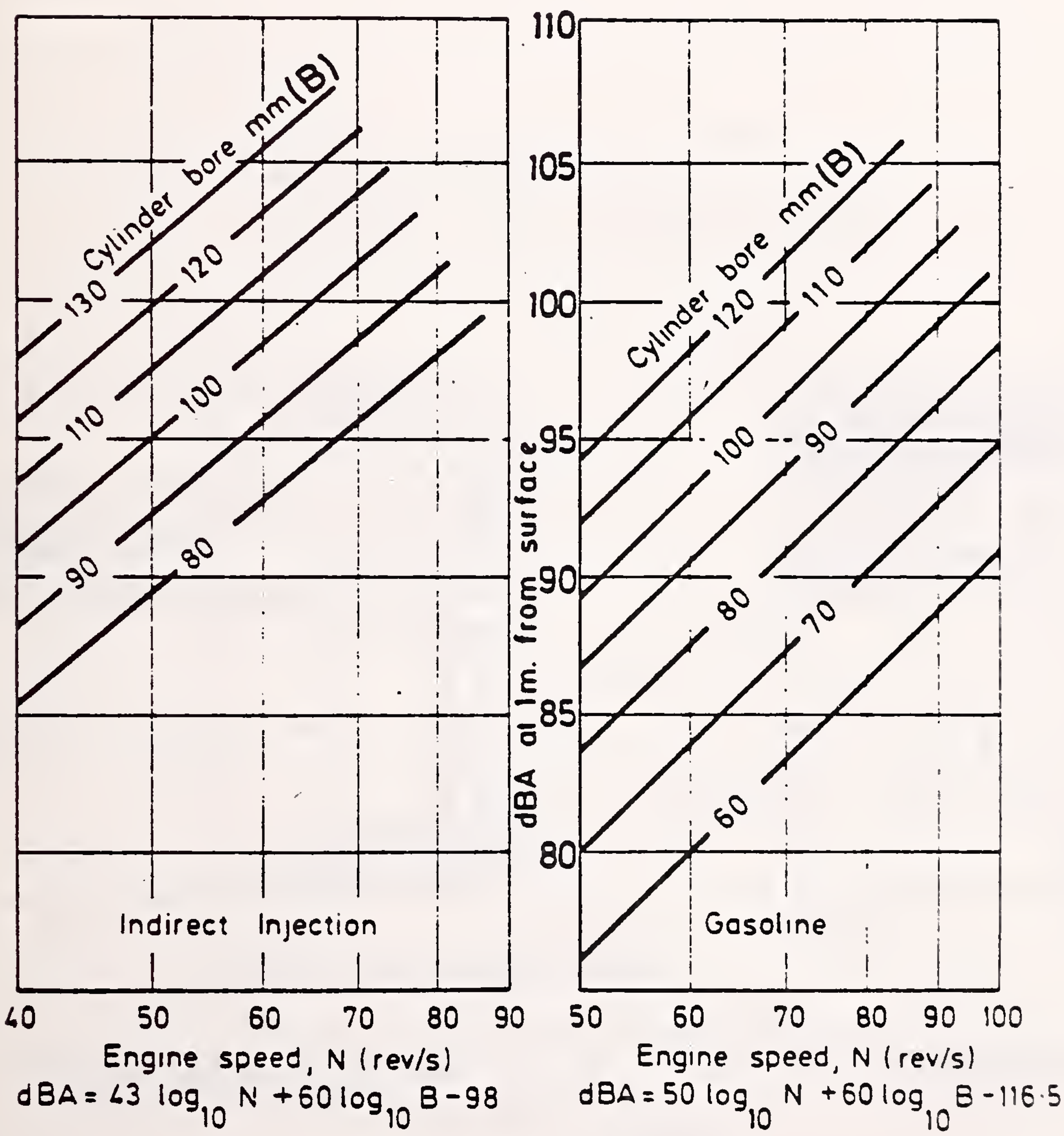


FIGURE 12. ENGINE NOISE PREDICTION CURVES - IDI DIESEL AND GASOLINE

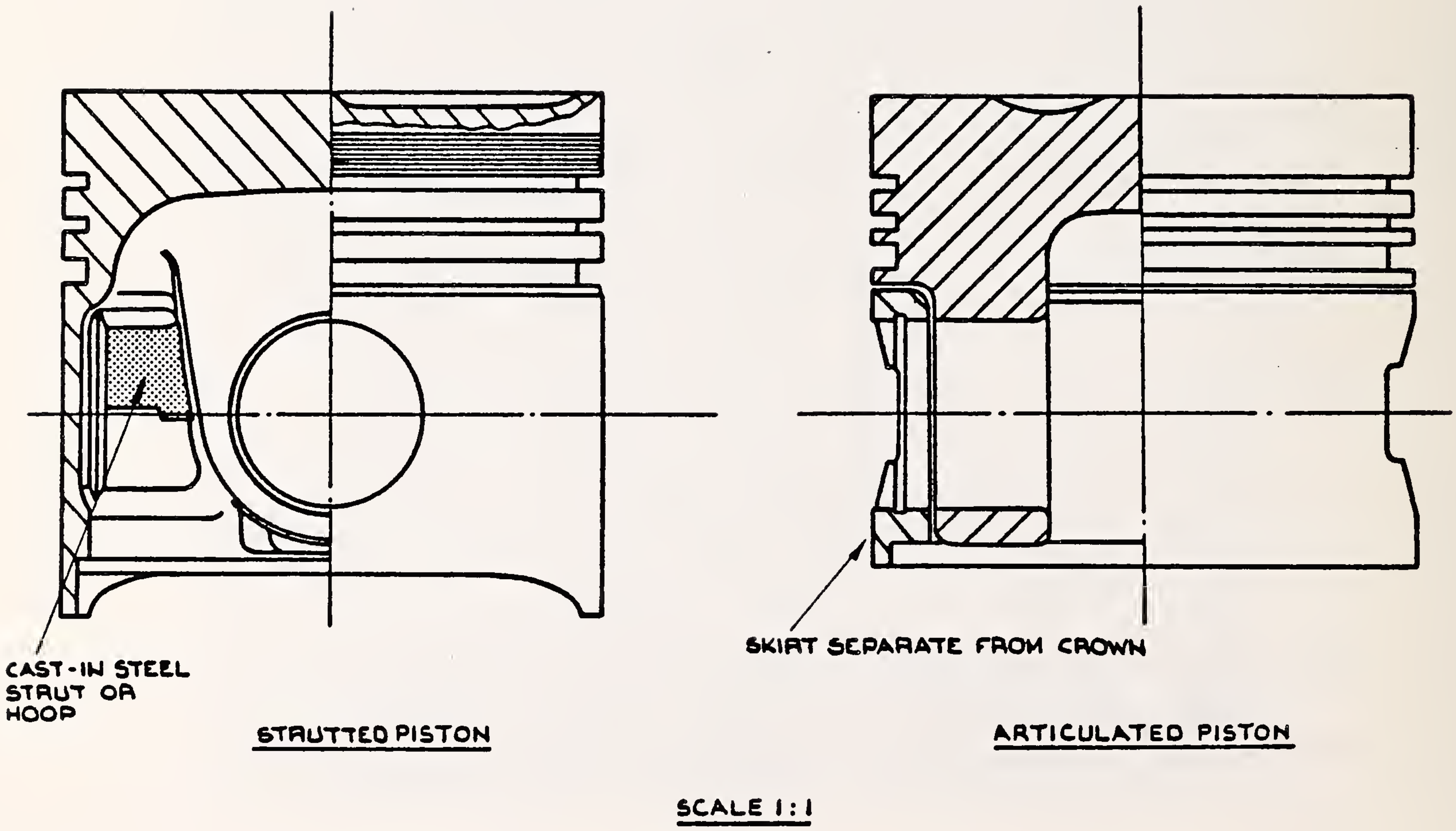


FIGURE 13. EXPANSION CONTROLLED PISTONS FOR LIGHT DUTY COMET V ENGINES

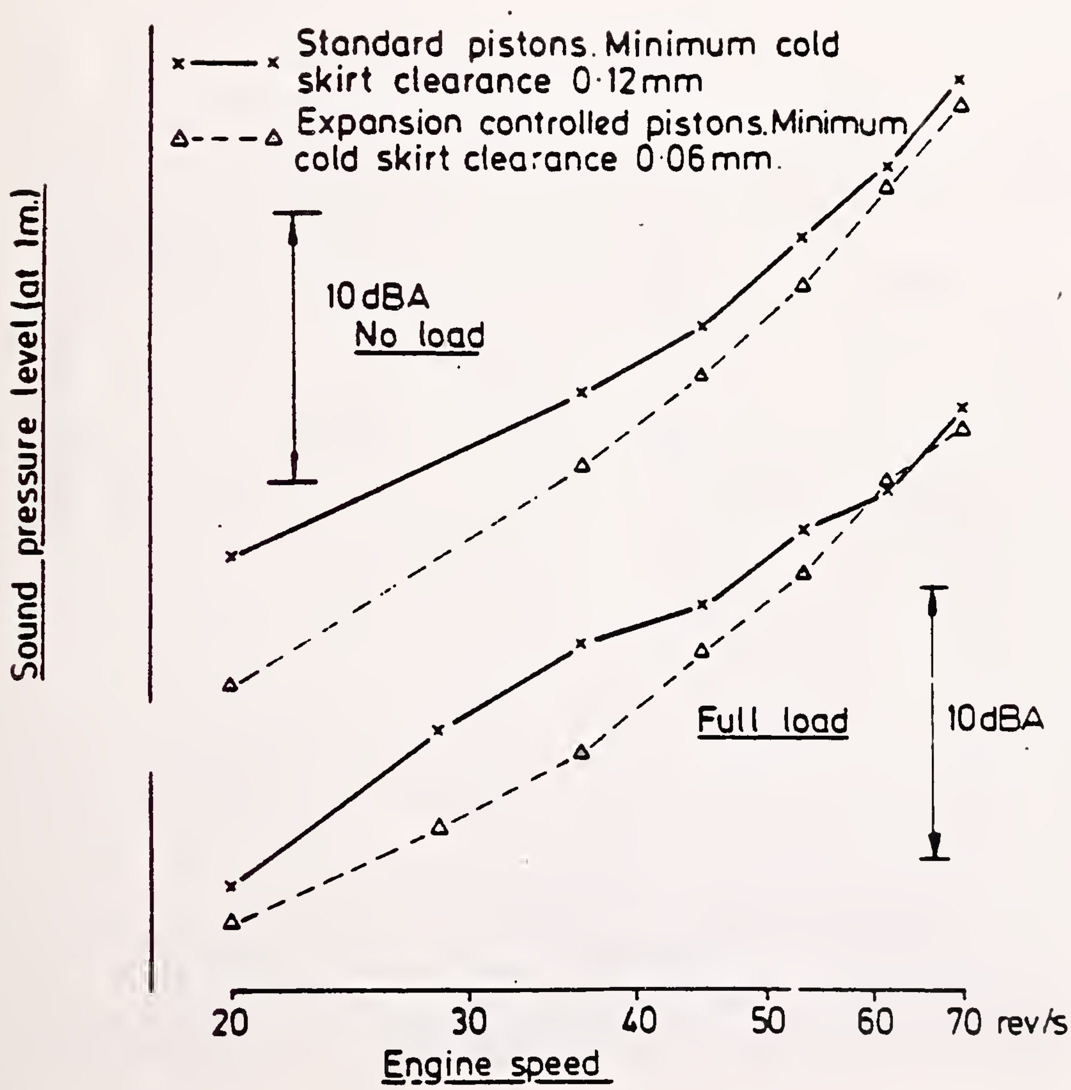


FIGURE 14. EFFECT OF EXPANSION CONTROLLED PISTONS ON THE OVERALL NOISE LEVELS FOR A 2L COMET V ENGINE

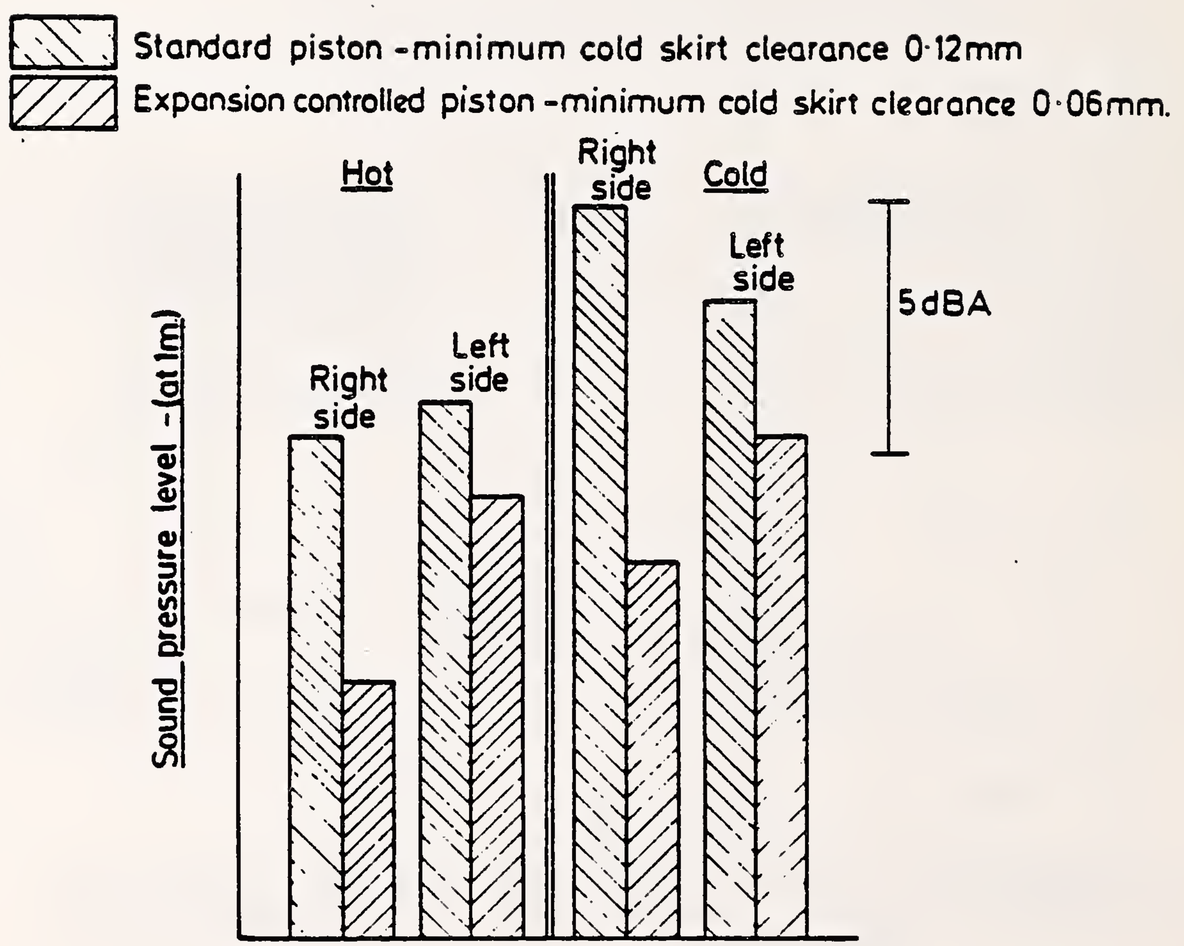


FIGURE 15. EFFECT OF EXPANSION CONTROLLED PISTONS ON THE OVERALL NOISE LEVELS AT IDLING (650 rev/min) FOR 2L COMET V ENGINE

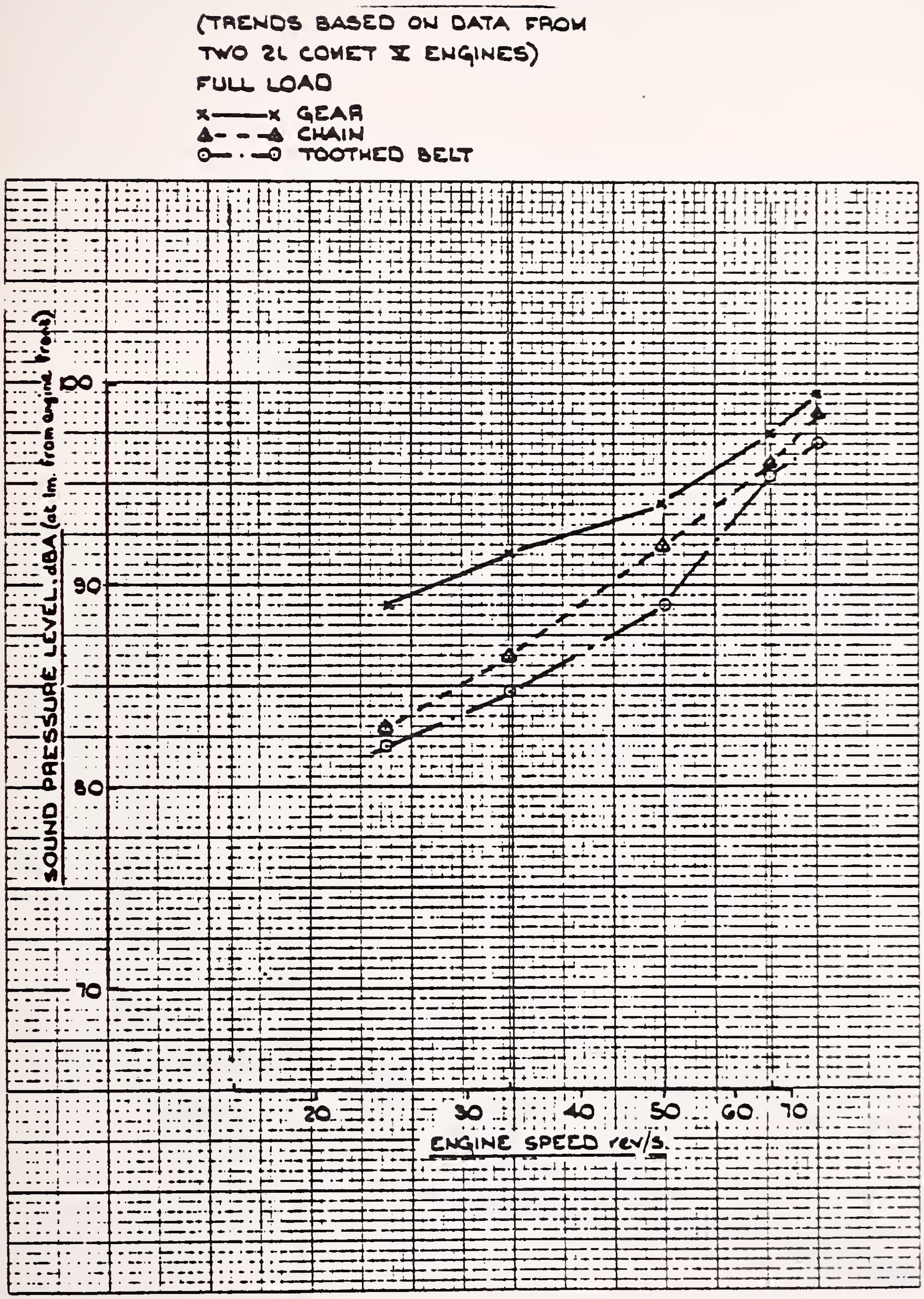


FIGURE 16. TYPICAL NOISE REDUCTION TRENDS FOR DIFFERENT TIMING DRIVE SYSTEMS

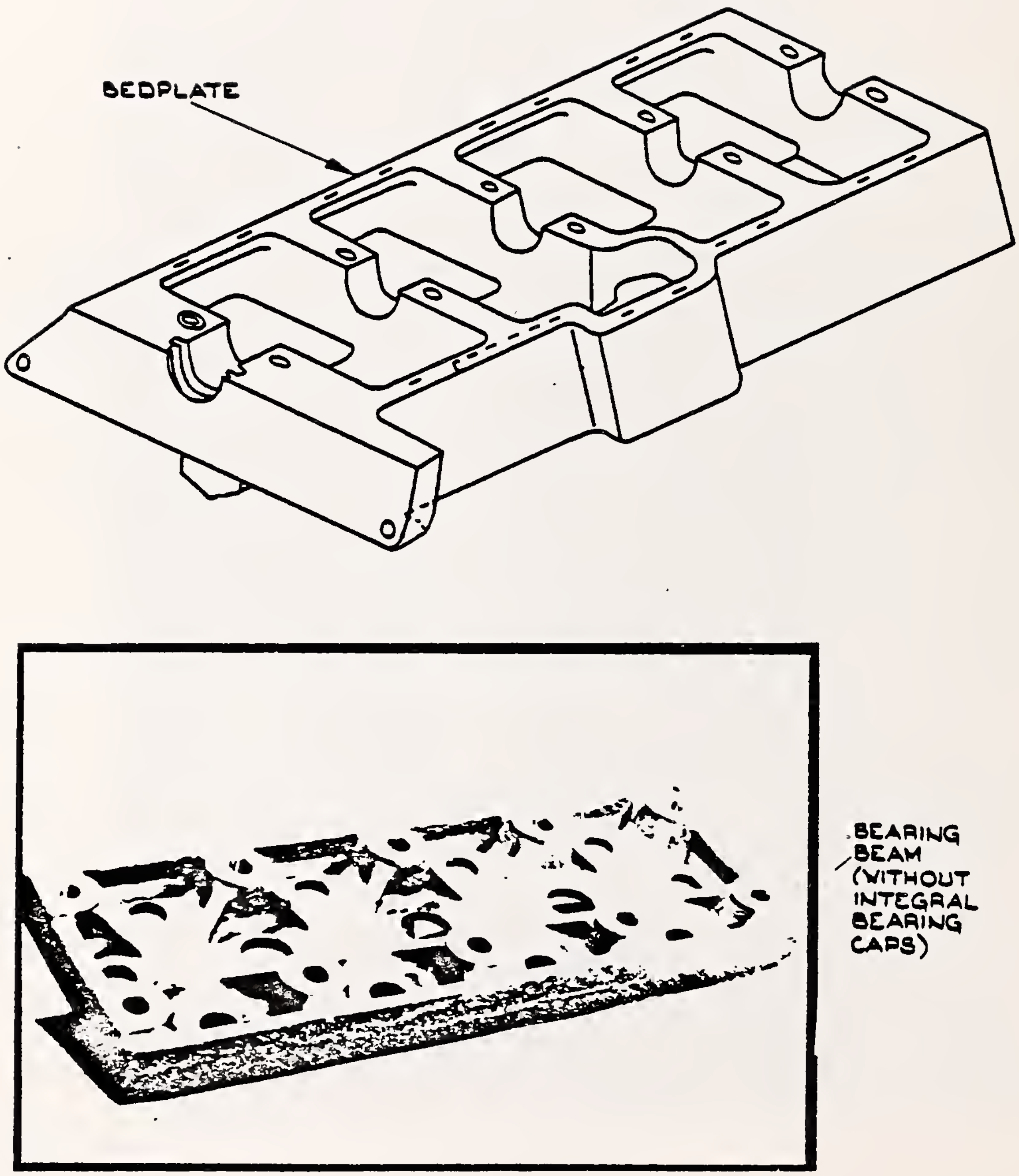


FIGURE 17. TYPICAL LIGHT DUTY ENGINE BEDPLATE AND BEARING BEAM

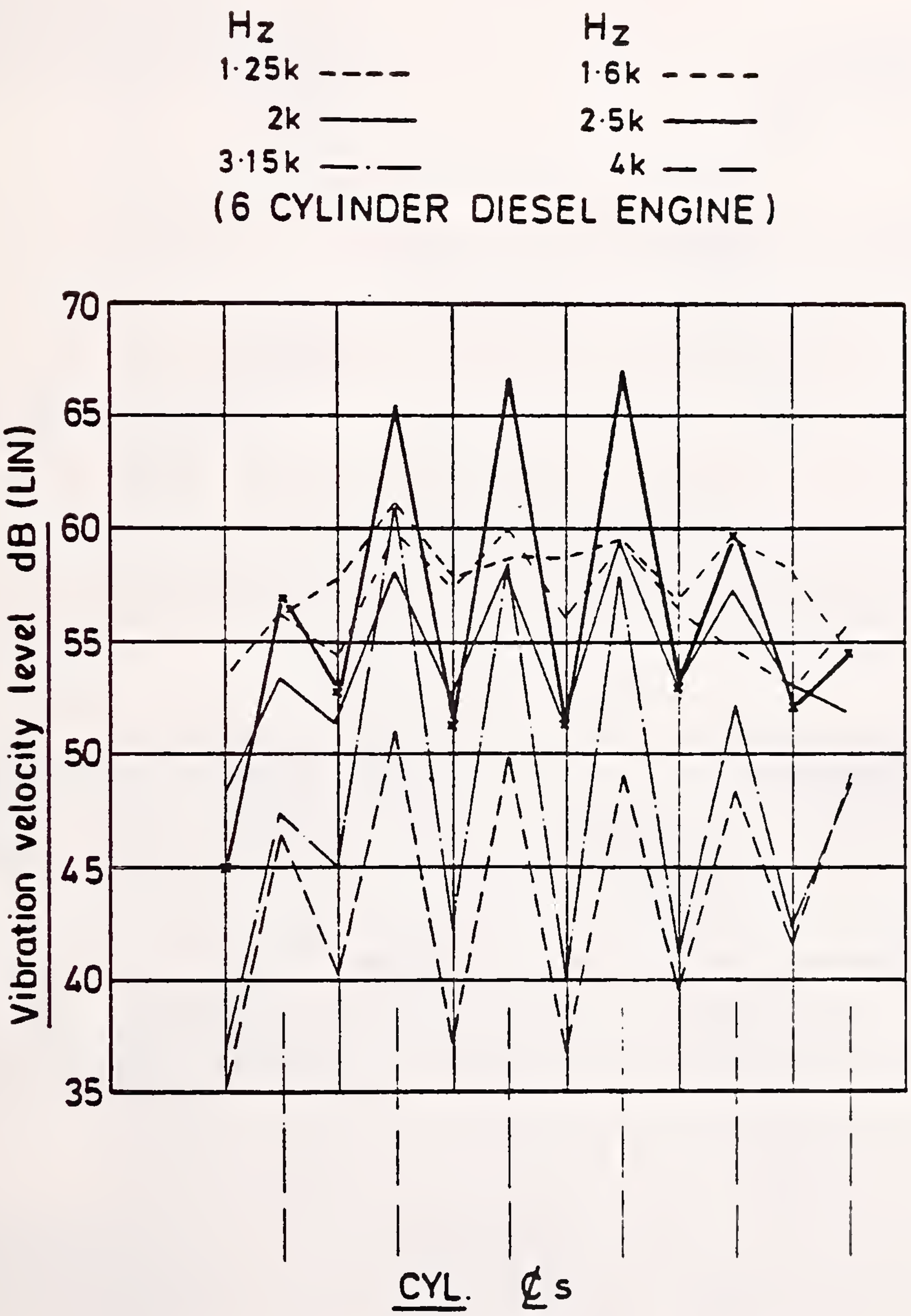


FIGURE 18. CRANKCASE PANEL VIBRATION AT VARIOUS 1/3 rd. OCTAVE CENTER FREQUENCIES

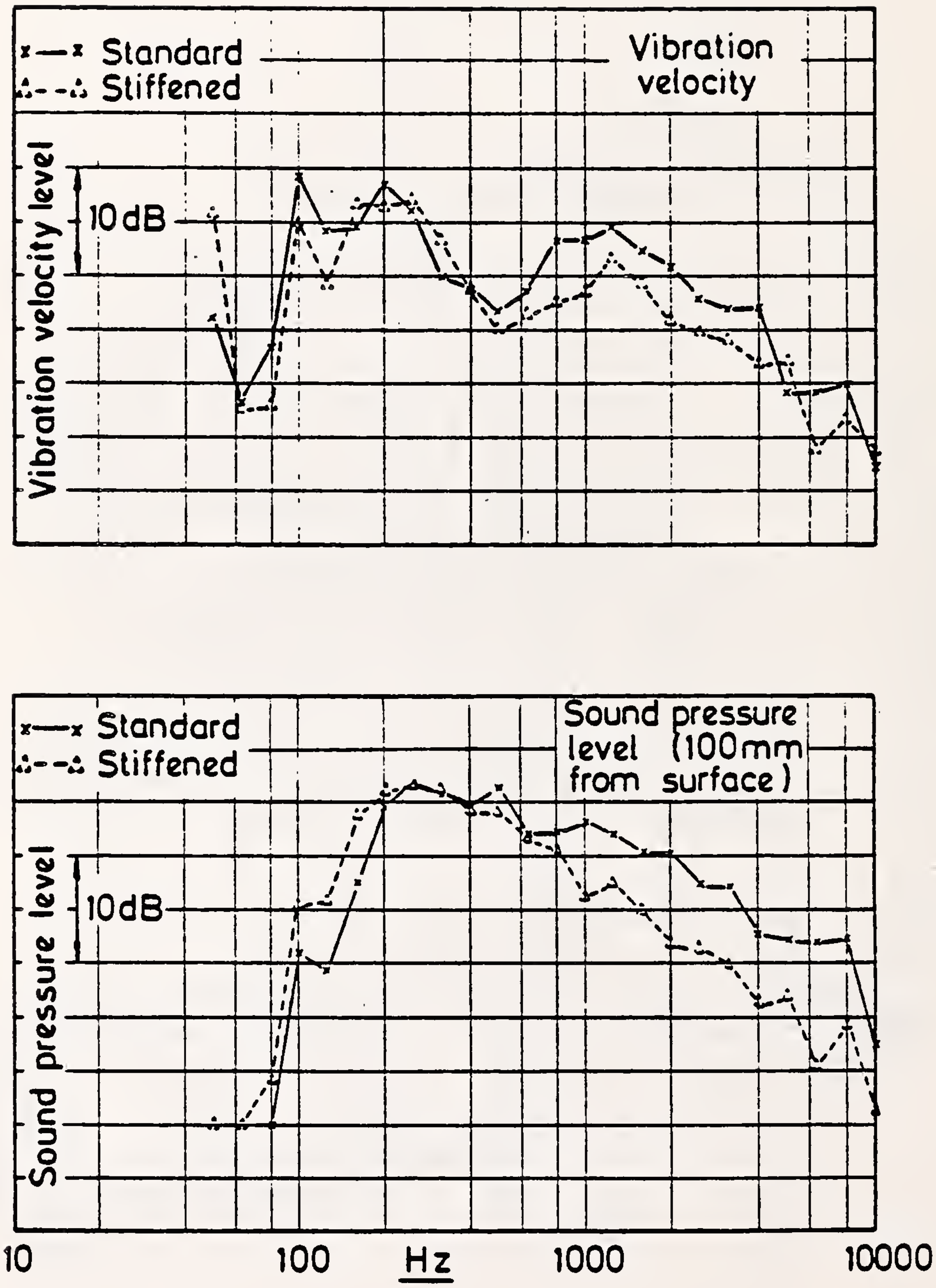


FIGURE 19. EFFECT OF RIBS ON CRANKCASE PANEL VIBRATION AND NOISE

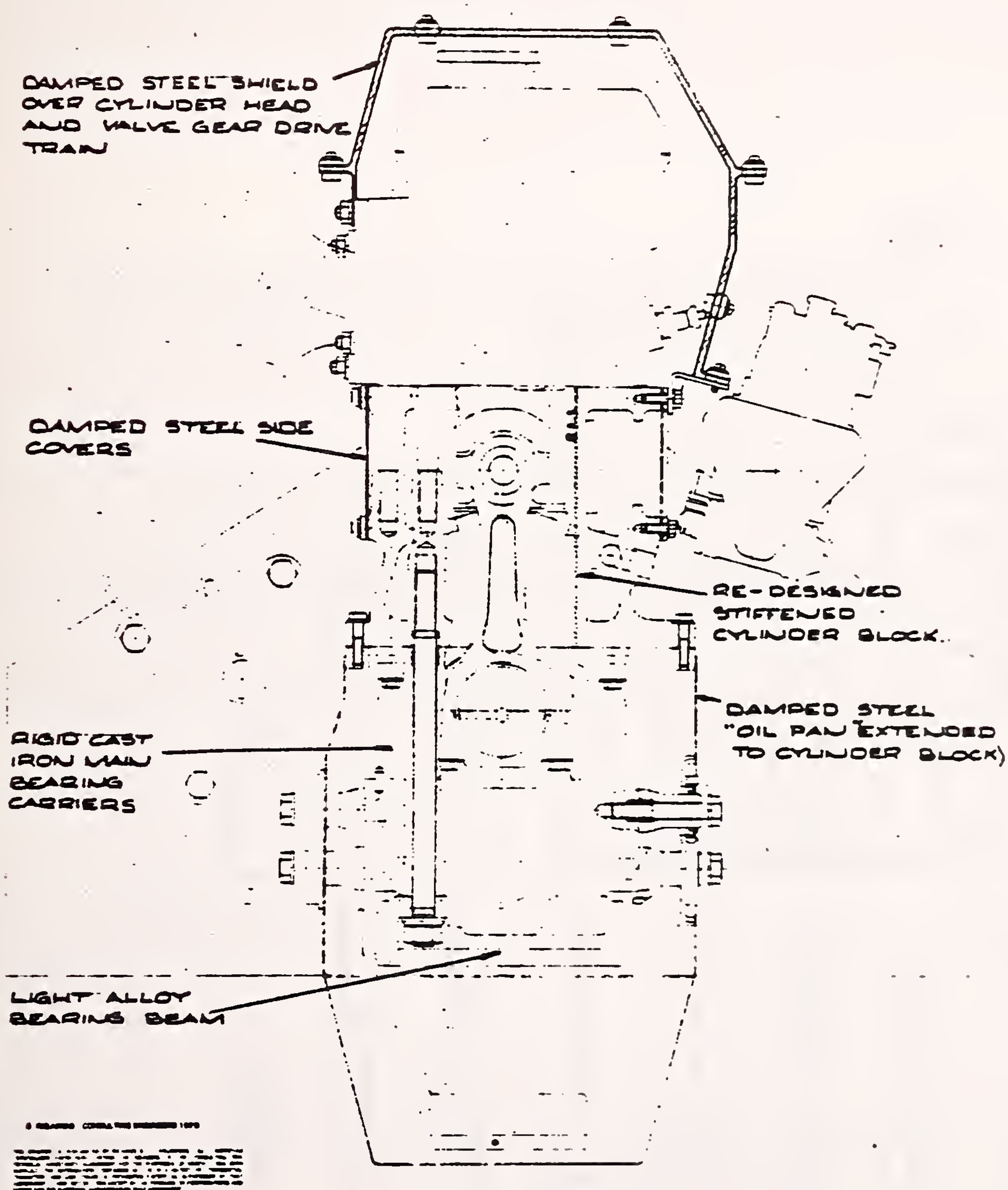


FIGURE 20. RICARDO LOW NOISE ENGINE BASED ON EXTENSIVE STRUCTURAL RE-DESIGN

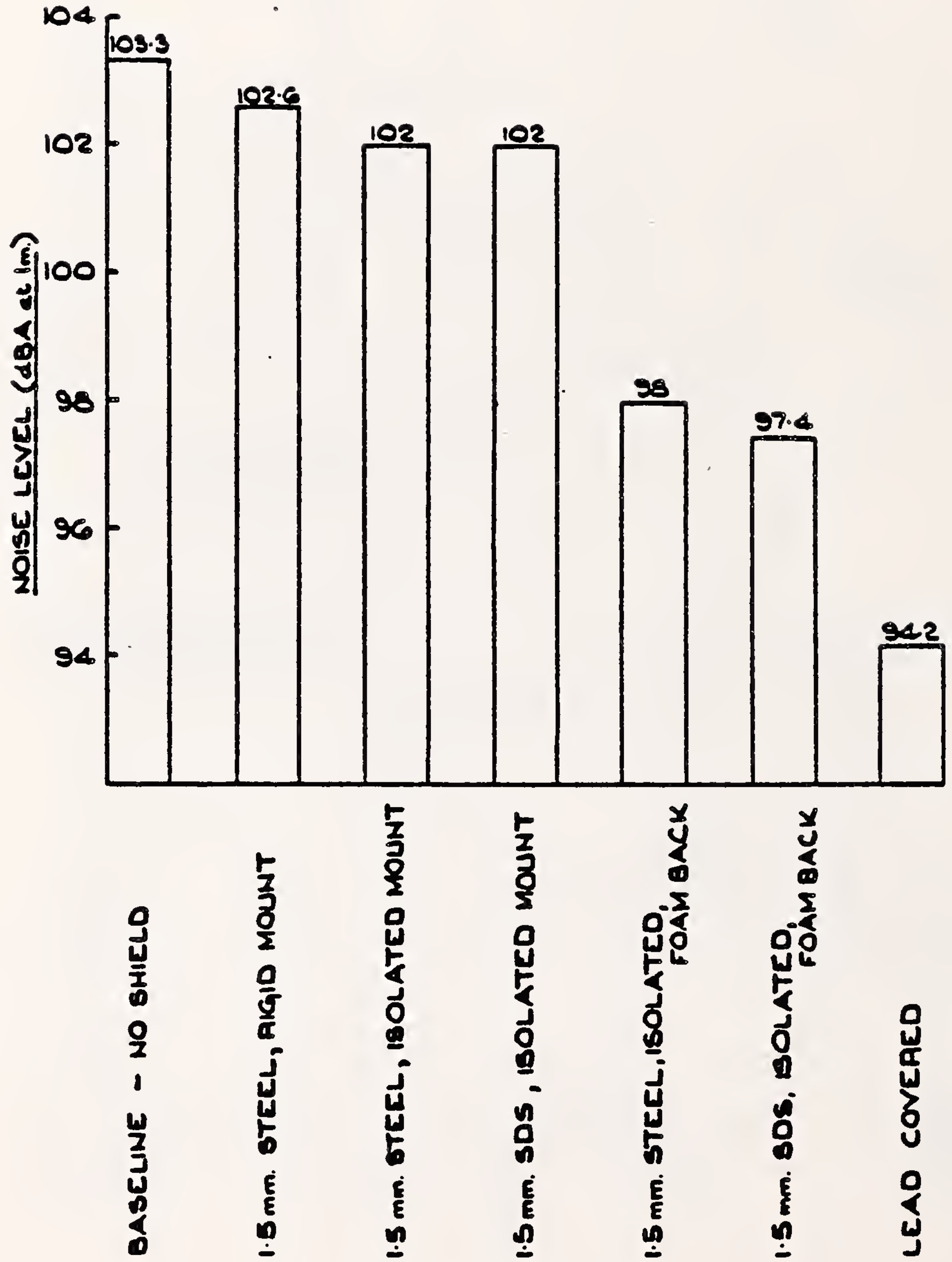


FIGURE 21. EFFECT OF VARIOUS CLOSE FITTING SHEILDS ON CRANKCASE SOURCE NOISE (REMAINDER OF ENGINE LEAD COVERED)

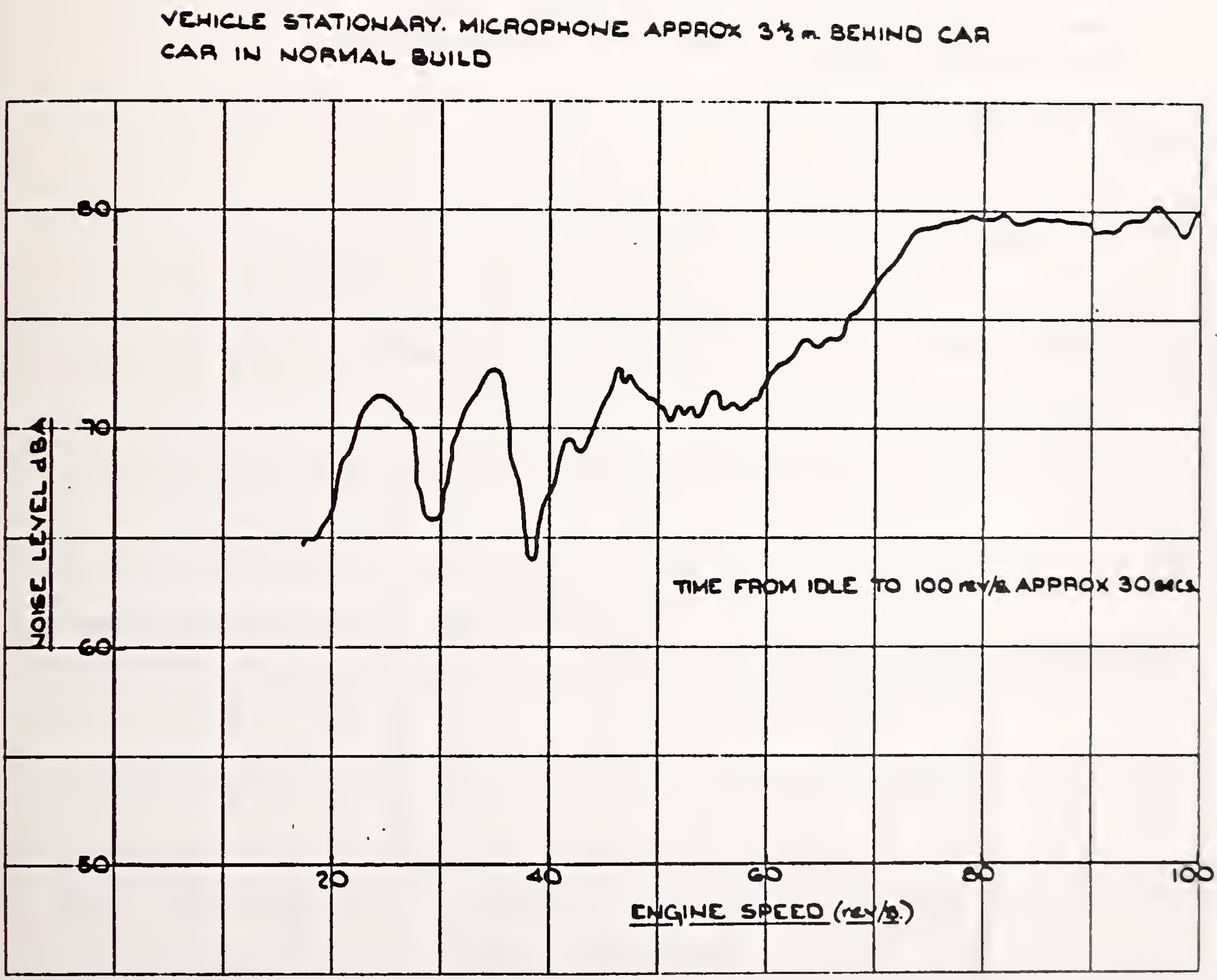


FIGURE 22. SAAB 99 GL: EXHAUST NOISE V ENGINE SPEED (SHOWING EFFECT OF EXHAUST RESONANCES)

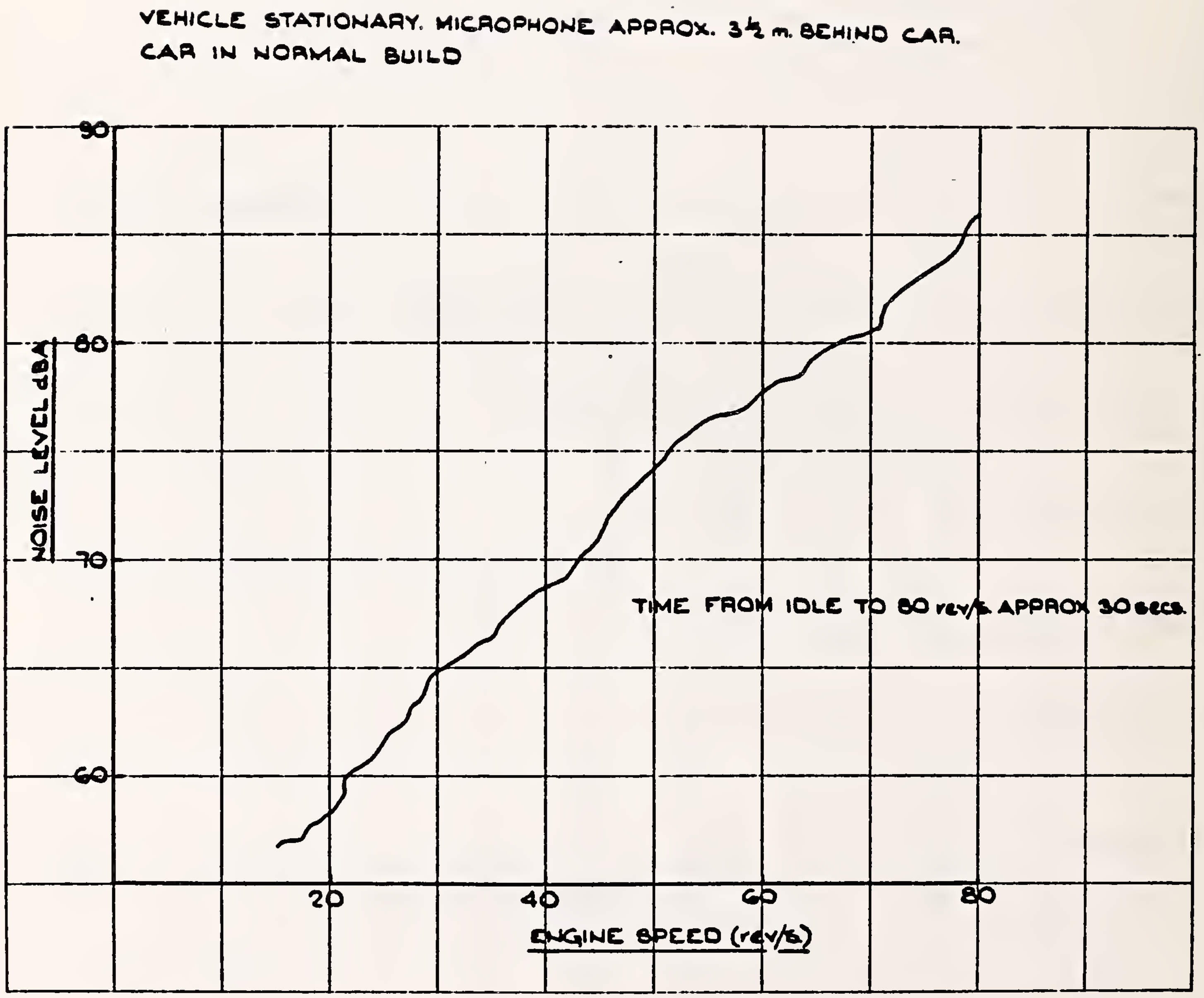


FIGURE 23. PEUGEOT 504 GLD: EXHAUST NOISE V ENGINE SPEED (SHOWING ABSENCE OF SIGNIFICANT EXHAUST RESONANCES)

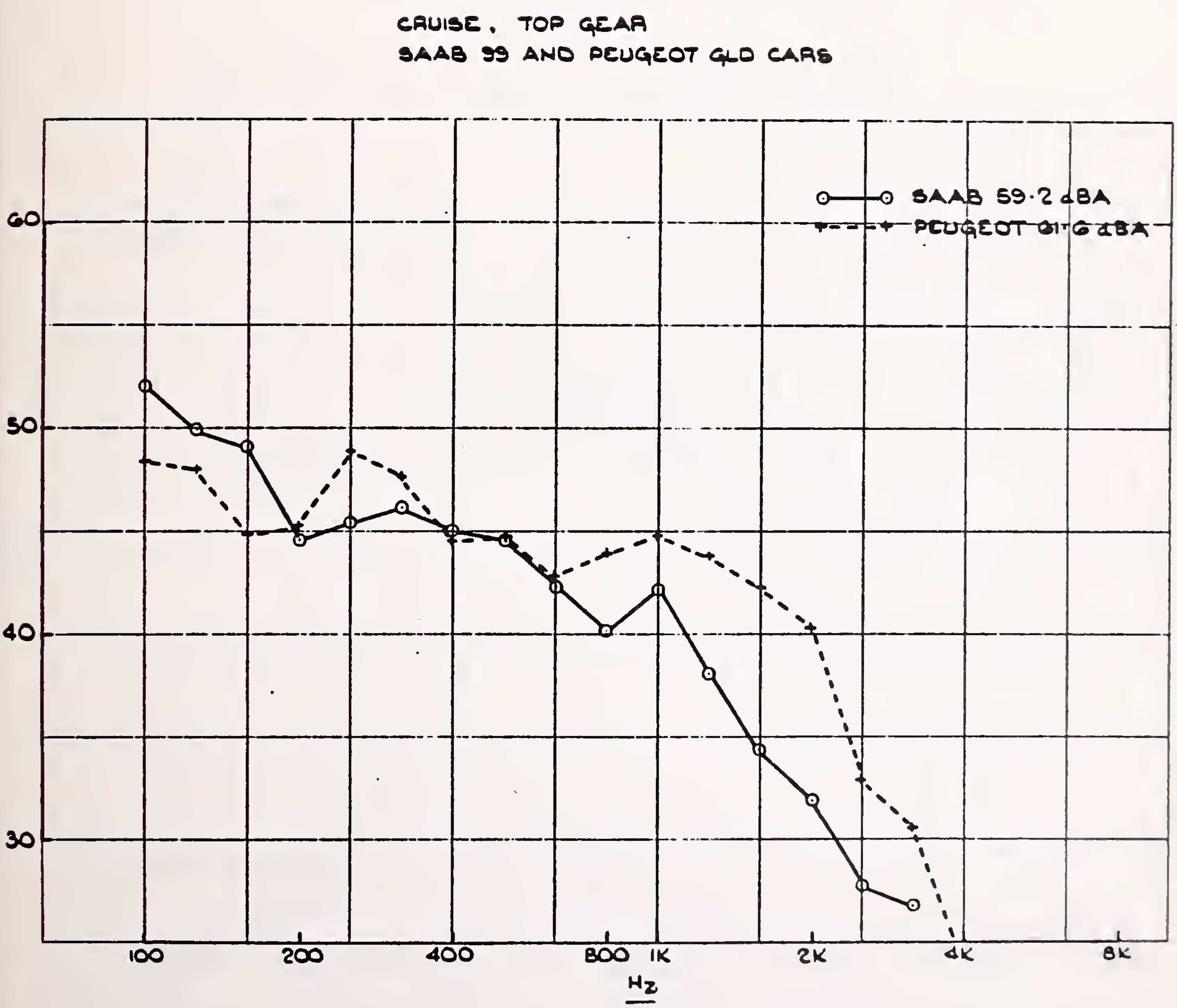


FIGURE 24. INTERIOR NOISE AT 50 km/h

SPEED, TOP GEAR
SAAB 99 AND PEUGEOT 504 GLD CARS

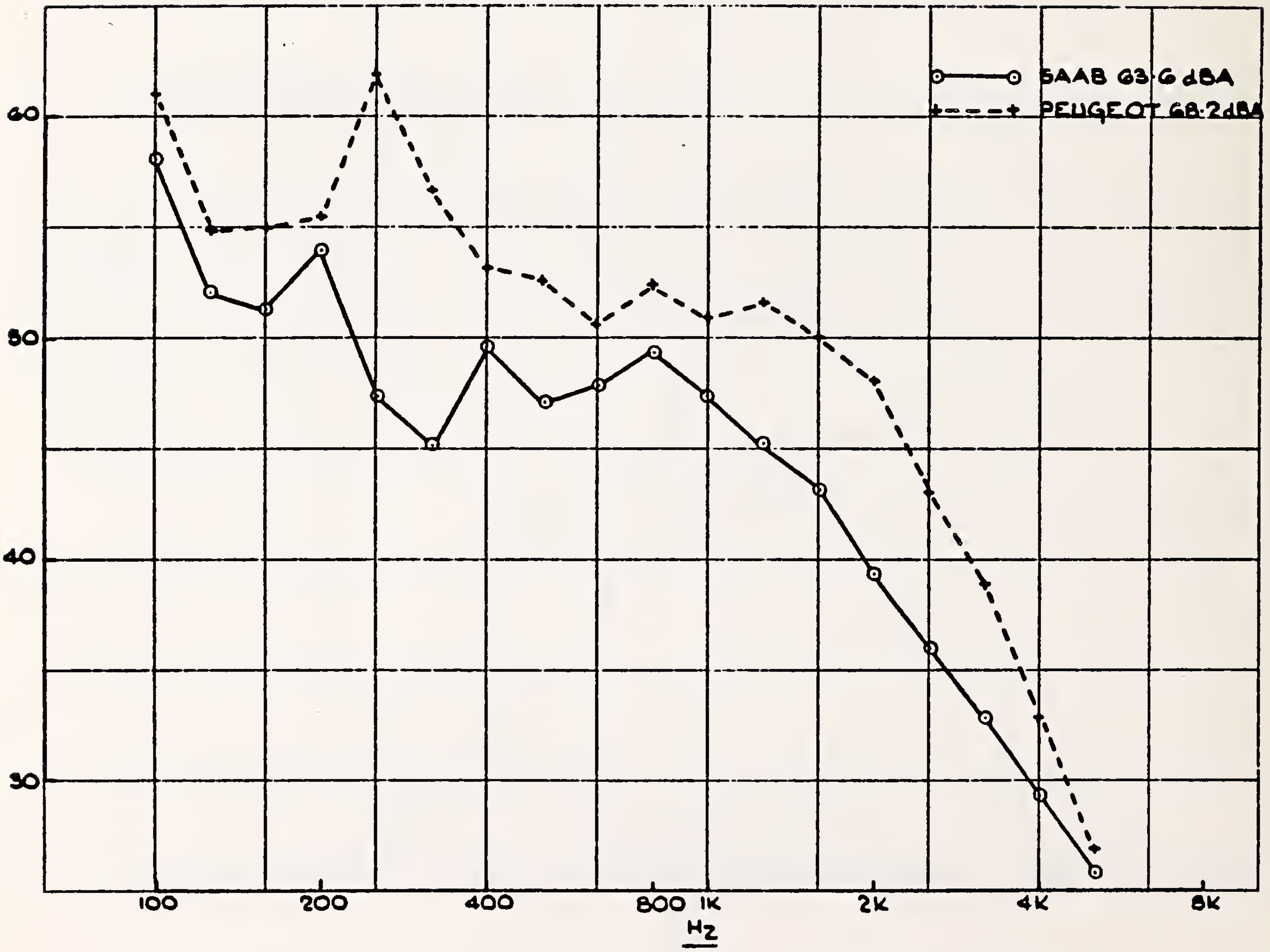


FIGURE 25. INTERIOR NOISE AT 80 km/h STEADY

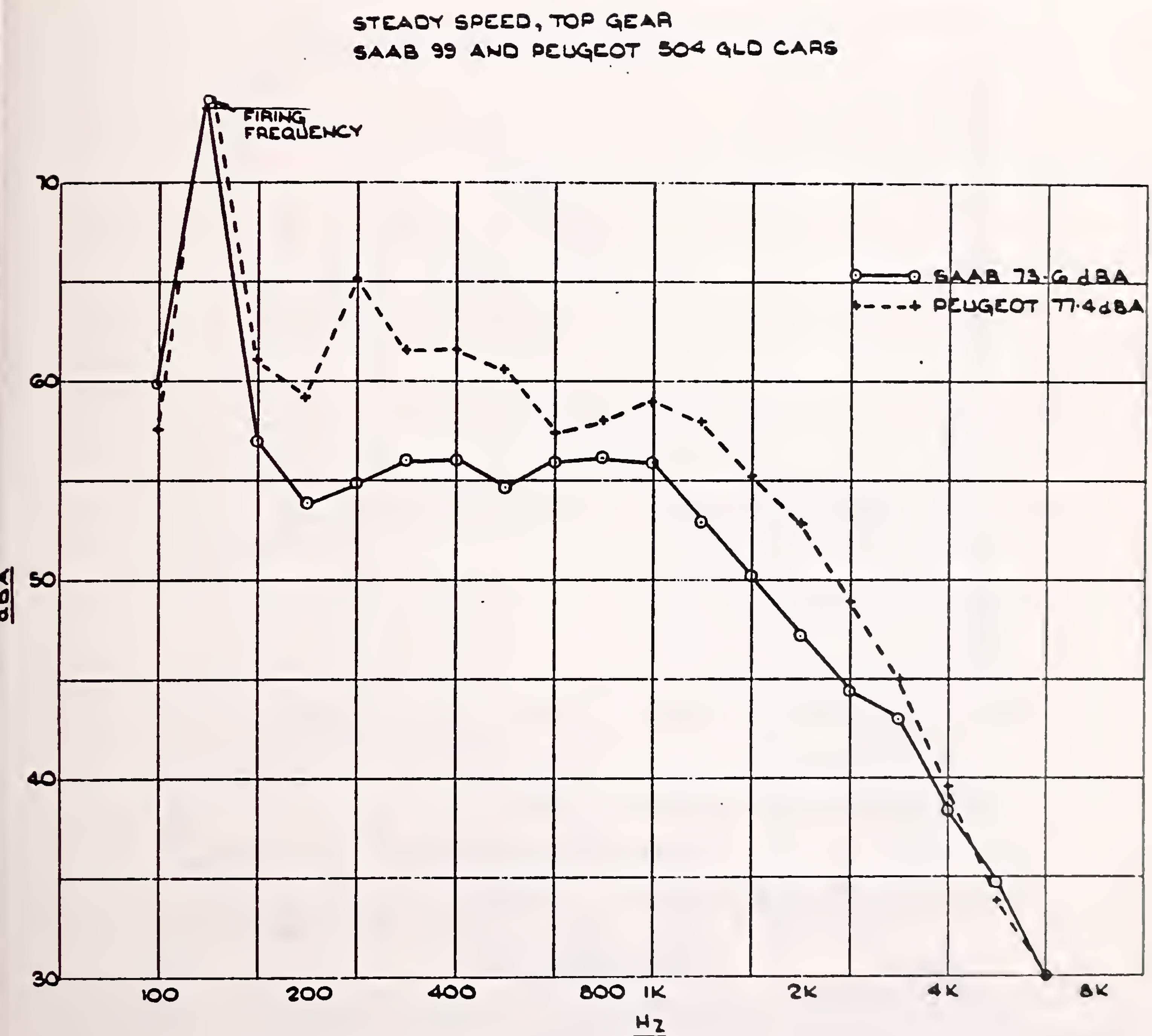


FIGURE 26. INTERIOR NOISE AT 112 km/h

1100 kg. GASOLINE CAR

COMPOSITE FUEL CONSUMPTION = $\dfrac{1}{\dfrac{0.55}{LA4} + \dfrac{0.45}{HIGHWAY}}$

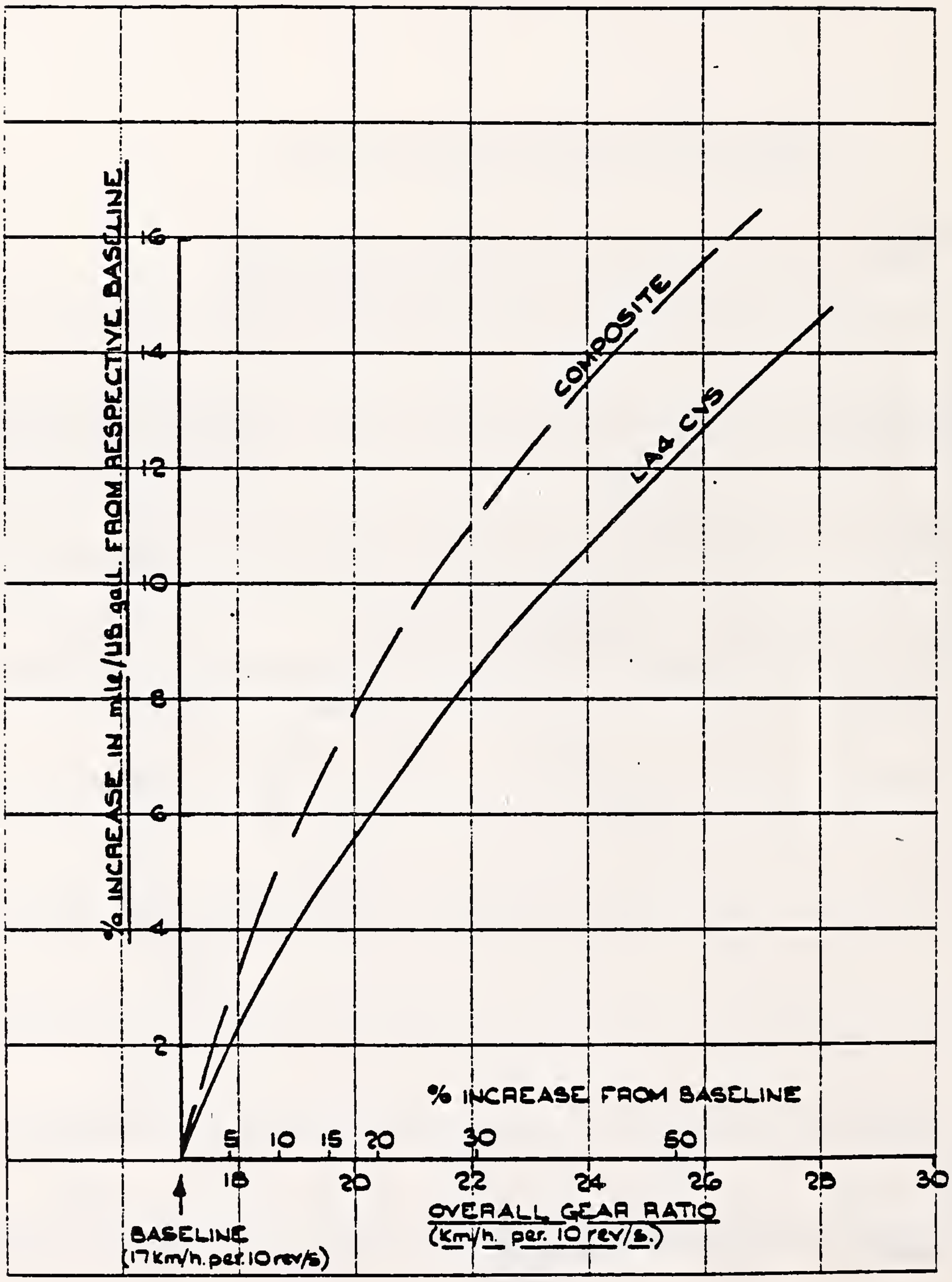


FIGURE 27. PREDICTED EFFECT OF OVERALL FINAL DRIVE RATIO ON COMPOSITE AND LA4 FUEL CONSUMPTION

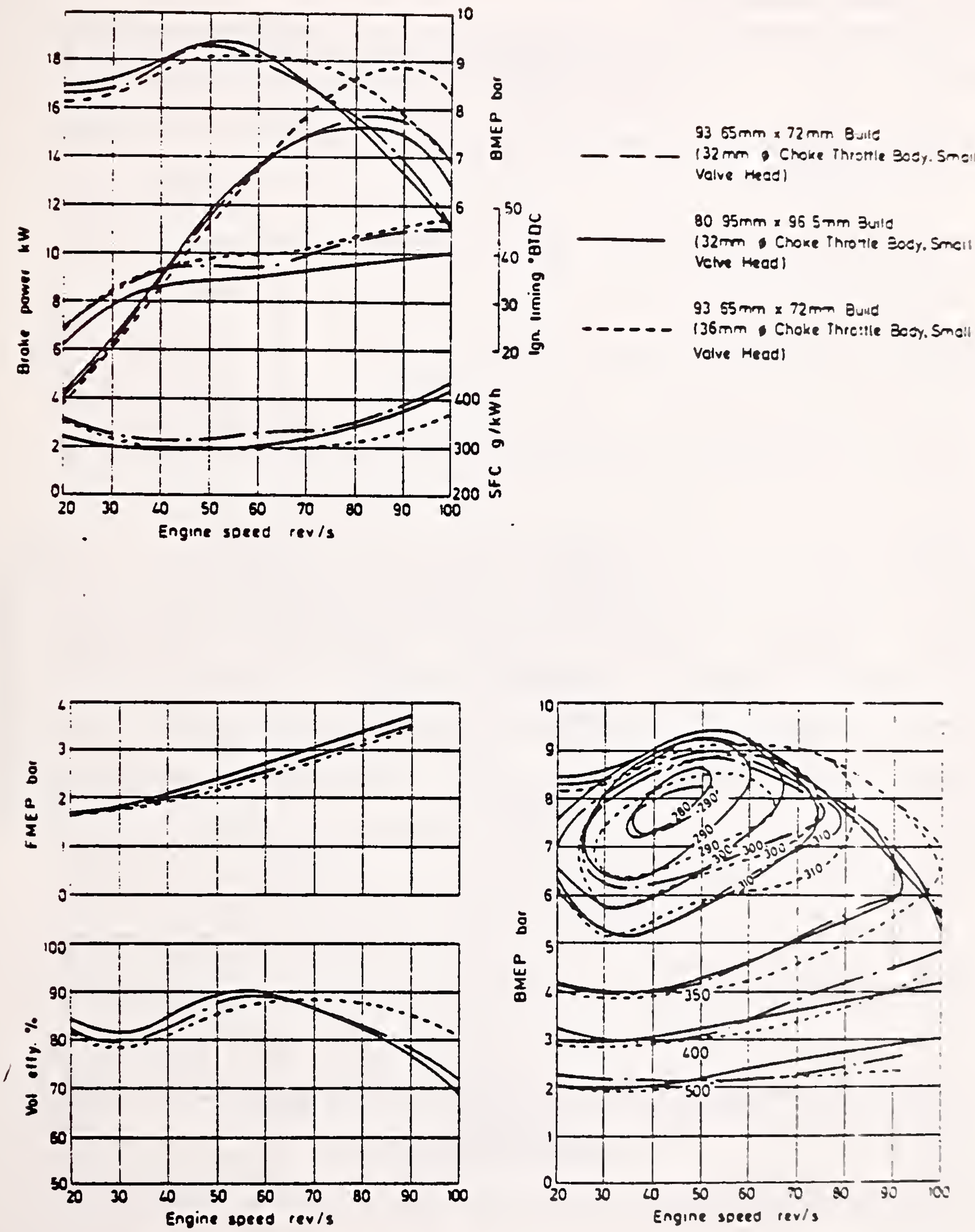


FIGURE 28. SINGLE CYLINDER RESEARCH ENGINE (SPARK IGNITED GASOLINE TESTS WITH VARIOUS BORE/STROKE COMBINATIONS

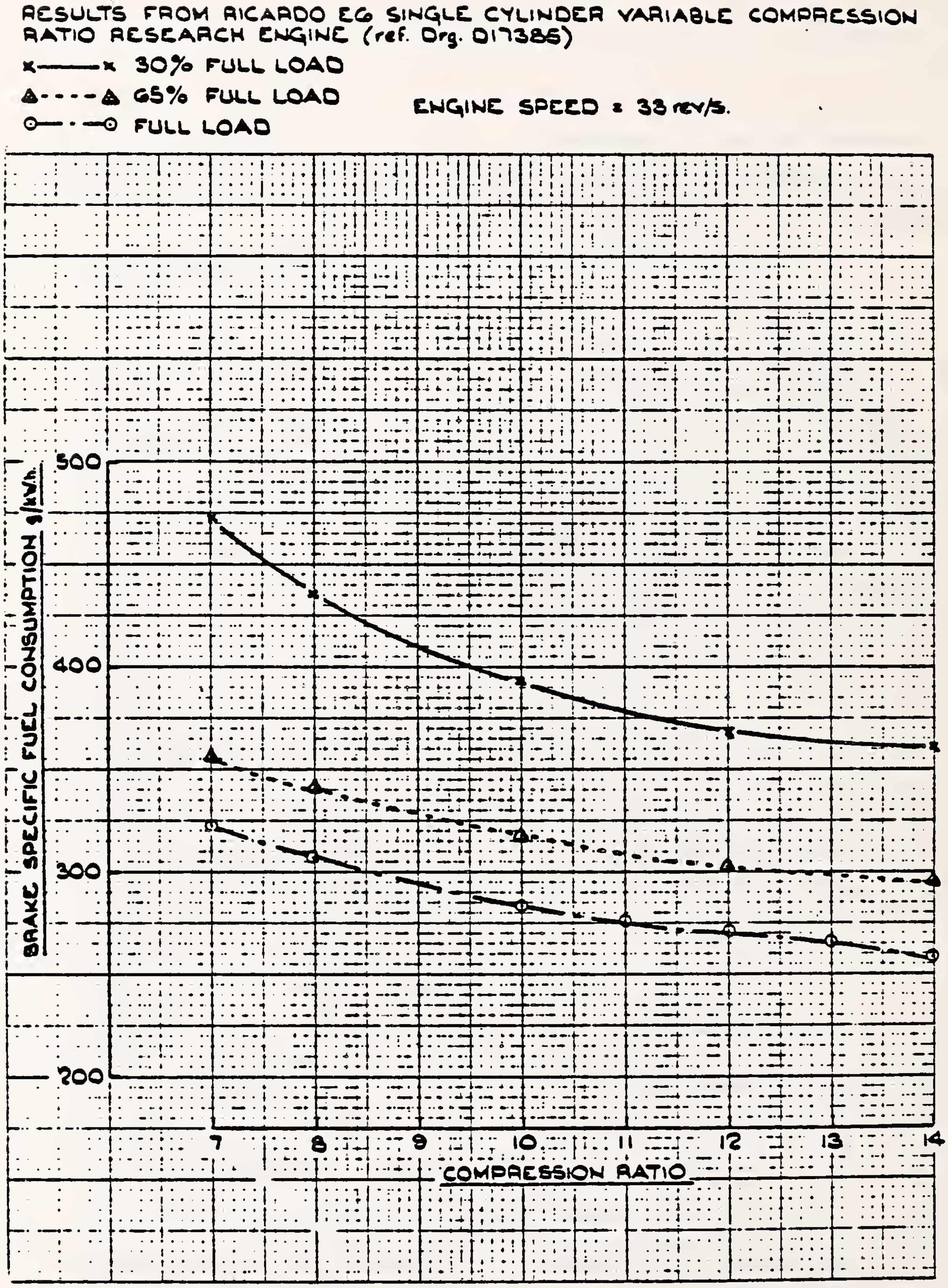


FIGURE 29. EFFECT OF COMPRESSION RATIO ON SPARK IGNITED GASOLINE ENGINE FUEL CONSUMPTION

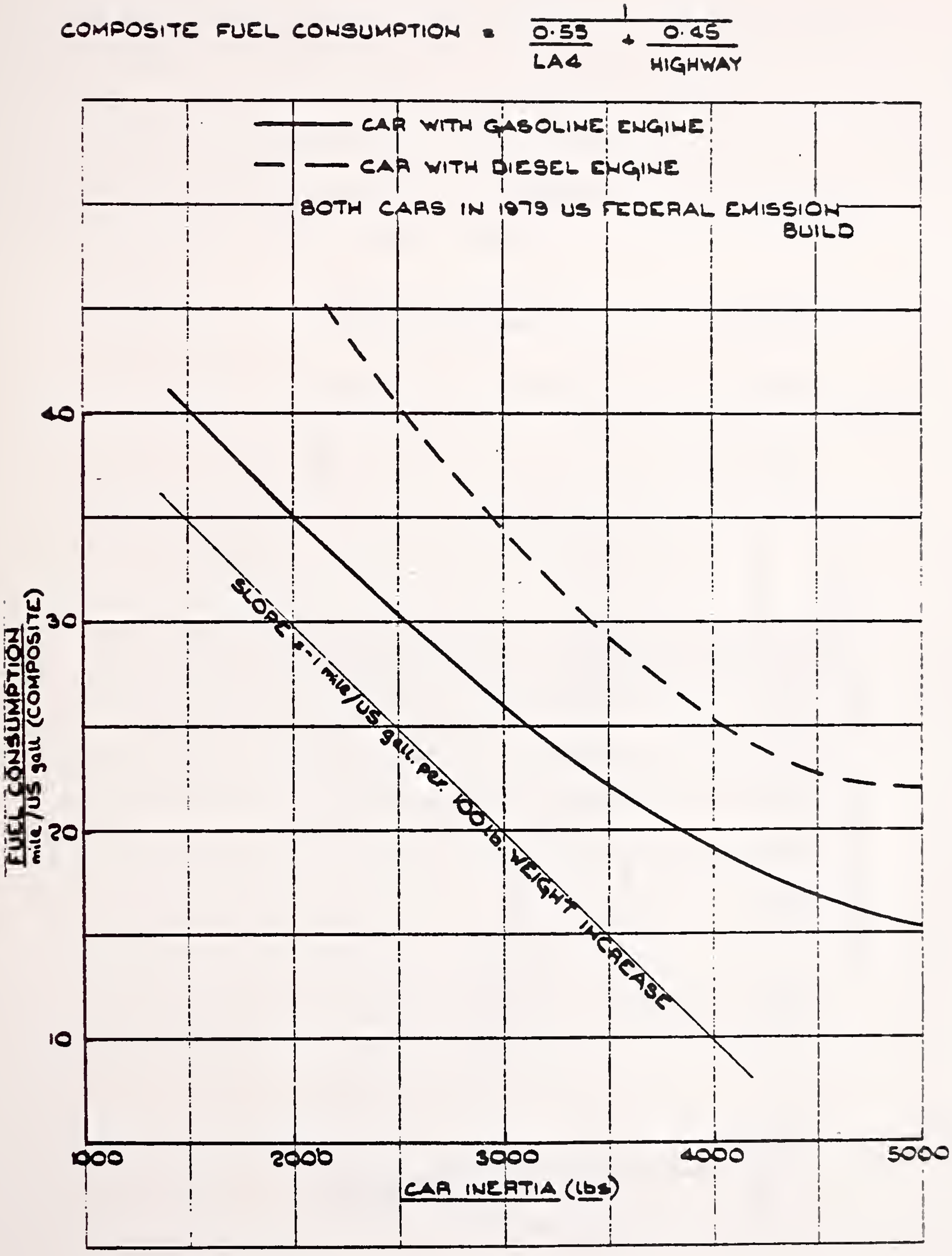


FIGURE 30. PREDICTED EFFECT OF VEHICLE INERTIA WEIGHT ON LA4/ HIGHWAY COMPOSITE FUEL CONSUMPTION

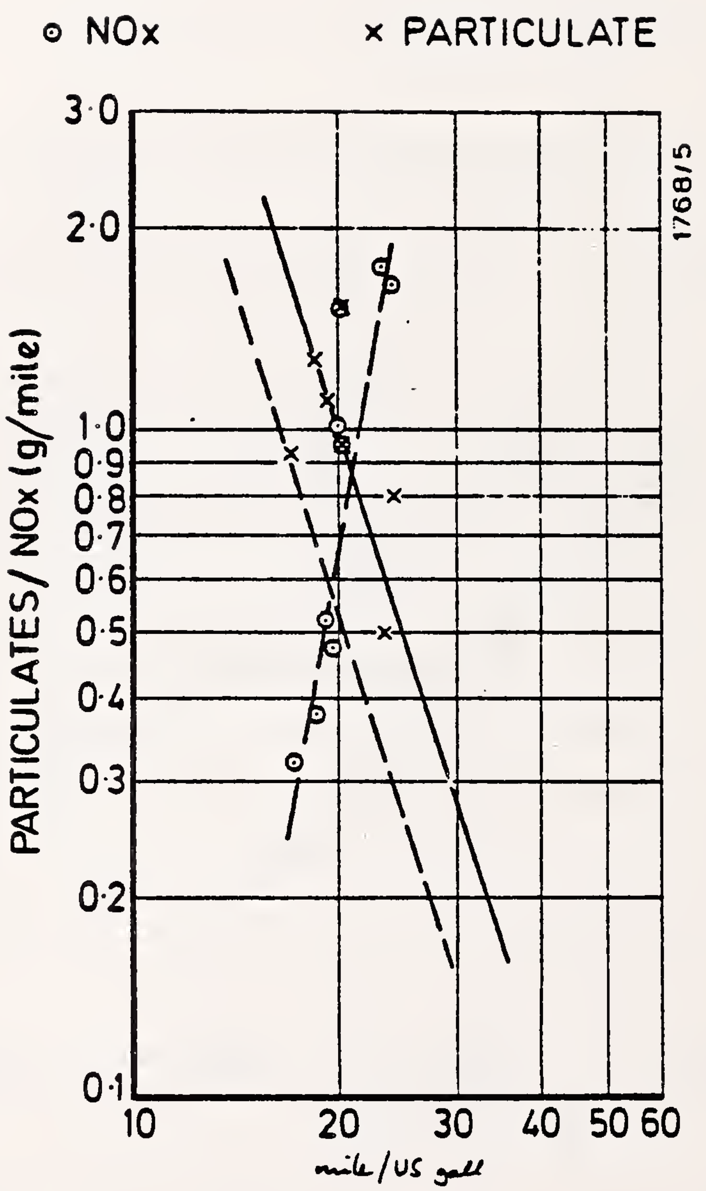


FIGURE 31. NOx/EXHAUST PARTICULATE/FUEL ECONOMY TRADE-OFF FOR A DIESEL PASSENGER CAR WITH MODULATED EXHAUST GAS RECIRCULATION AND INJECTION RETARD OVER F.T.P. URBAN CYCLE

# APPENDIX B
# REPORT OF NEW TECHNOLOGY

The work performed for this report, while leading to no new inventions, has provided detailed information on noise, noise control, and fuel economy characteristics of small, high speed internal combustion engines. Particular attention is given to tradeoffs between noise control measures and their effect on engine performance, cost, and weight.

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