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VEHICLE NOISE LEGISLATION - A TRUCK INDUSTRY VIEWPOINT

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1. INTRODUCTION

The paper describes work carried out by a U.K. truck manufacturer in meeting current and proposed EEC vehicle noise directives. An outline is given of the methods adopted of attenuating vehicle drive-by noise, both experimental and in production. In addition the paper is offered as a means of conveying the experiences of a manufacturer dealing with acoustic measurements in the field.

2. LEGISLATIVE REQUIREMENTS

2.1 EEC Vehicle Noise Directives.

Because of environmental considerations and public demand for ever quieter vehicles, especially heavy trucks, continual revision of noise legislation will continue into the next century. This will happen throughout the developed world but at the present time, is particularly relevant to European truck makers.

A Compulsory reduction in drive-by noise means that after October 1995 a vehicle type approval certificate will not be issued without compliance with revised directives.

Table 1 - EEC Directive 92/97/EEC. (Heavy Goods)

Vehicle Categories (Goods)	Maximum Drive-By Noise Level dB(A) Current	Amended
GTW > 3.5Tonnes < 75KW	81	77
> 3.5Tonnes > 75KW < 150KW	83	78
> 3.5Tonnes > 150 KW	84	80

Note: A correction factor of 1dB (subtractive) is allowed for equipment error. All further values contained in this paper will be of measured results only.

2.2 Impact of Revised Directives.

Over a period of twenty five years heavy vehicle pass-by noise will have been reduced from 92 to 80dB(A). A 38 or 44 tonne gross weight heavy lorry, by the mid 1990's, will have to be quieter (under test conditions) than a passenger car was in 1970. This obviously presents a formidable challenge to the manufacturers.

3. VEHICLE DEVELOPMENT TRENDS

3.1 Increased Power

In parallel with reductions in acceptable drive-by noise has come demands for increased power and performance. Until the late 1970's an articulated vehicle of 200 - 230BHP would have been considered adequate for most operating conditions. Today 400 - 500BHP machines are common. Increased power and weight carrying capacity has in turn led to larger and heavier duty transmissions and auxiliary equipment.

3.2 The Company Product Range

Foden Trucks is a manufacturer of commercial vehicles ranging from 17 to 44 tonnes GTW (Gross Train Weight). Power unit options range from 134KW (180BHP) to 347KW (465BHP).

3.3 Government Assisted Projects

Foden Trucks has benefited from Government assistance to the commercial vehicle industry over a number of years. During the 1970's support was provided for the production of an experimental vehicle with a drive-by level below 80dB(A). More recently public assistance was allocated to industry for the development of quieter production vehicles for the 1990's (QHV 90 Project). This resulted from the Armitage Report "People, Lorries and the Environment" and the subsequent white paper. Both projects highlighted the level of development necessary to comply with anticipated directives.

3.3.1 The QHV90 Project. Foden Trucks was contracted (by the Transport Road Research Laboratory) to construct two heavy vehicles designed to meet a (corrected) pass-by limit of 82dB(A) and 88dB(C). The (C) weighted target, although not in any directive, was included because evidence suggested that low frequency noise can cause annoyance to people in their homes.

Table 2 - Vehicle Specifications.

	Vehicle No. 1	Vehicle No. 2
Vehicle Type:	S106T Tractor (Double Drive)	S106TS Tractor (Twin Steer)
Engine Type:	(A) 10 litre Six Cylinder, Turbocharged 242KW (325BHP)	(B) 12 litre Six Cylinder, Turbocharged 224KW (300BHP)
Gearbox:	9 Speed Constant Mesh	9 Speed Constant Mesh
Rear Axle:	Single Reduction Hypoid	Single Reduction Hypoid
Cab:	Glass Reinforced Plastic (GRP)	Glass Reinforced Plastic

Note:
Both engines were modified to reduce sound emissions as part of the QHV 90 Project. Engine B was more extensively developed however.

3.4 Test Procedures.

3.4.1 Test Area. Dimensions for an approved test site are shown by Fig.1 (Appendix). The test surface must be hard and flat.

3.4.2 Test Method. The test vehicle must approach the measurement zone at 3/4 of maximum power rated speed or 50KPH (whichever is slower). Upon reaching the line A-A maximum acceleration is to be applied until the line B-B has been passed. Sound level measurements, taken at a distance of 7.5m from track centreline, must be in RMS, frontal and fast response modes.

Table 3 - Test Equipment

Precision Sound Level Meter	B & K Type	2231
FM Reel to Reel Level Recorder	B & K Type	7006
Single Channel Frequency Analyser	B & K Type	2143
Narrow Band Spectrum Analyser	B & K Type	2031

Note:
1. The single channel (real time) analyser was used to display recorded broad band spectra for general assessment. 2. The narrow band analyser was normally used to examine discrete frequencies by means of Fourier Transforms.

4. DISCUSSION OF EXPERIMENTAL RESULTS

4.1 Vehicle Noise.
Vehicle noise consists of six primary sources, listed as follows :-

1. Exhaust Gas Pulses.
2. Engine (Combustion and Mechanical Processes).
3. Powertrain/Transmission.
4. Cooling System
5. Air Induction System
6. Ancillary Equipment

Two secondary sources are also present :-

7. Wheel and Tyre.
8. Wind and Aerodynamic

It should be noted that as primary sound attenuation progresses, the lesser components become more significant and often more difficult to reduce.

4.2 Methods of Noise Source Ranking.
4.2.1 Identification. In order to identify and rank the various noise sources two methods were employed; shielding (with absorption or barrier materials) and decoupling.

A composite material, of 1mm lead and 25mm of ceramic quilt (L/Q), was used extensively to clad potential noise sources. Lead, because of its great density, has a high transmission loss factor. Ceramic fibres possess high sound absorption coefficients. Polyurethane foam/lead composites (F/L) were also used as an alternative to L/Q. Both of these materials by virtue of excessive cost and weight were limited to experimental use only.

Drive-by noise measurements were taken before and after cladding the various noise sources. The effects of component shielding were assessed both cumulatively and separately. Calculated differences in magnitude could only be used as a guide to source ranking however, as some sound energy would always penetrate the covering material affecting the meter reading.

4.2.2 Calculations. The equations for determining transmission loss factors and the addition of coincident sound waves are shown in the Appendix.

4.3 Experimental Results.
4.3.1 Exhaust Attenuation. Experimental exhaust attenuation was carried out by using a battery of four absorption mufflers, connected in series and clad in lead sheet to reduce casing noise. These provided a datum from which proprietary silencers could be assessed.

4.3.2 Air Intake Attenuation. Air induction noise was quietened by using a single absorption muffler and then cladding the system with L/Q quilt.

4.3.3 Power Unit/Powertrain Attenuation. The engine and transmission were quietened by overwrapping with L/Q sheeting. A chassis mounted acoustic undertray was used to attenuate oil pan (sump) noise. Acoustic (L/Q) side screens were employed to block crankcase and turbocharger noise.

Table 4 - Experimental Drive-By Noise Reduction.

Noise Source	Acoustic Attenuation	Drive-By Level dB(A)
Standard Vehicle	None	88.5
Exhaust System	4 Absorption Silencers	86.0
Gearbox	L/Q Cladding	85.3
Engine Sump	Acoustic Undertray	83.3
Engine Timing Case	F/L on Upper Cab Grille	82.9
Engine Crankcase	L/Q Panels	81.8
Rear Axle Bowl	L/Q Wrapping	81.4
Rear Engine	Rear Cab Enclosing Panels (F/L Lined)	80.1
Air Intake System	Absorption Silencer	79.4

See Fig.2 (Appendix).

4.3.4 Drive-By Spectrum. Frequency spectra, shown by Figures 4 & 5 (Appendix), display 1/3 octave recordings of vehicle drive-by noise before and after experimental sound attenuation.

Fig.4 (Standard Condition) shows that the dominant frequency, produced by 1/3rd order engine firing, was approximately 80HZ. Other frequencies highlighted were; gearbox noise at 800HZ & 2KHZ and turbocharger whistle at 8KHZ.

Fig.5 (Experimental Condition) gives resultant and much flatter spectrum, obtained after application of treatments specified (Table 4).

5. PRODUCTION METHODS

5.1 Exhaust Attenuation.
5.1.1 Exhaust Noise. Test results showed exhaust noise to be of primary importance. Its attenuation alone was sufficient to enable a standard production vehicle to meet the 88dB(C) project target. Suitable engine and gearbox encapsulation could then reduce measured drive-by noise to 80dB(A).

5.1.2 Silencer Construction. Absorption (Passive) silencers contain ceramic or mineral fibres* Depending on size, they are capable of attenuating broad band noise with limited gas restriction. A disadvantage however lies in the deterioratioj of the core material (disputed by modern manufacturers). Consequently additional conditioning tests are required for certification.

Reactive silencers function by diffusing engine gas pulses through internal baffles and expansion chambers. Anticipated frequencies, predominantly from 1/3rd order firing, are cancelled in Helmholtz chambers. Lower frequencies (longer wavelengths) require larger volumes however. Increased sound attenuation increases gas restriction (back pressure) which if excessive can reduce fuel economy, cause particulate problems and reduce engine life.

The available combinations of diffusion and expansion chambers are endless. Consequently the design of reactive silencers is empirical and success depends on trial and error.

Note: A theoretical method for determining silencer performance was developed as part of the QHV90 Project at Loughborough University. Known as LAMPS, the technique requires the use of finite elements tk determine gas flow dynamics. Its validity however, requires very accurate measurement of pressure and temperature gradients across the exhaust system.

5.2 Cooling System
Cooling system noise is principally caused by the fan. If engaged (by an air temperature sensing viscous drive) the sound produced will be excessive. The magnitude of this sound is proportional to the cube power of fan tip speed.

Table 5 - Axial Fan Generated Noise.

Fan Diameter(mm) and Type	Fan Speed (RPM)	Resultant Noise dB(A)
610 7 Bhaded	1900	74.6
660 8 Bladed	2700	82.6
711 6 Bladed	2500	84.8
762 8 Bladed	2500	86.3

Note:
Research at The National Engineering Laboratory (NEL Report PR2 ESCP/01) emphasised the importance close fan tip to fan ring clearance. If engaged fan noise became mandatory for vehicle certification however mixed flow fans would be needed. Such an arrangement was developed for the QHV 80 Project.

5.3 Partial Enclosures
Comparative results shown by Table 3 Section 5 highlight the level of encapsulation (and silencer improvement) needed to produce a drive-by of 80dB(A). Partial enclosures were assessed as a more practical alternative.

Table 6 - Effects of Partial Enclosures (Ref. Fig.3 Appendix)

Noise Source	Acoustic Attenuation	Drive-By Level dB(A)
Standard Vehicle	None	88.5
QHV90 Engine Development	Timing Case Modified	87.4
Exhaust System	Improved Silencer (Reactive)	84.6
Engine Crankcase etc.	Lined Wheelarch	83.8
Engine Rear	Partial Rear Enclosure	83.6
Engine Sump	Acoustic Sump Cover	83.2

Note:
1. The specificatioj of acoustic lining material was made according to criteria of low cost and weight, durability and heat and oil resistance. A high density heat resistant polyurethane foam, lined with a 2mm thick, air permeable(thermoplastic weave was considered to be most suitable.

2. The acoustic sump cover was fabricated from laminated steel lined with 25mm heat resistant foam. Stepped bolts were used for attachment.

3. All suppliers considered were identified by the MIRA Materiahs Handbook.

5.4 Attenuating Engine Noise at Source
5.4.1 Internal Modifications. Engine B was extensively re-developed as part of the engine manufacturer's QHV 90 Project. Modifications included combustion chamber redesign, the incorporation of two stage (pilot) fuel injection and fitment of a floating idler pulley bearing.

The effect of these modifications alone was to produce a reduction in vehicle drive-by noise of 1.2dB (85.2 to 84.0dB(A)).

5.4.2 External Modifications. Sound energy absorption and vibration isolation techniques were also used on engine B with the following results:-

Table 7 - Engine Modifications

Acoustic Component	Measured Change (dB)
Front Pulley Cover	Nil
Isolated Rocker Cover	Nil
Isolated Inlet Manifold	Negligible
Isolated Sump Pan	1.1
Right Hand Crankcase Panel	0.7
Left Hand Crankcase Panel	1.0

Note:
1. The combined effect of internal and external engine modifications, together with improved exhaust silencing was a measured (at TRRL) drive-by level of 80.5dB(A).

2. Modifications to the upper engine, whilst producing significant results on bench testing, were obscured by the vehicle cab enclosure.

3. Sump pan vibration, a pronounced source of noise with most engines, was isolated by means of a compressed rubber ring located with a detachable flange and tapered bolts. Two engine manufacturers provide proprietary acoustic sump covers. Isolated attachment is required however.

6. CONCLUSION

The paper indicates the level of development work undertaken by part of the UK truck industry in order to produce environmentally acceptable commercial vehicles. Particular reference has been made to the government supported QHV 90 Project and its benefits to industry. Significant factors emerged from this and earlier work concerning the impact of revised noise directives.

In terms of cost and weight the components found necessary to comply with directive EEC 424/84 were relatively modest; Approximately £100 cost and 10 to 15kg weight being added per vehicle. Further reductions below 84dB(A) (corrected) require substantially increased effort and expense.

The most significant reduction in vehicle noise was produced by improvements in exhaust silencer performance. After considerable comparative assessment of alternatives against a known datum, an effective silencer was obtained which could be used with a of variety engines and arrangements. An optimum compromise between exhaust restriction and acoustic attenuation appeared to have been reached. (The LAMPS Programme may provide further improvements, although manufacturers believe silencers will need to be increased in size).

The use of partial enclosures were necessary on an untreated engine in order to reach a (corrected) drive-by level of 82dB(A). However the reduction of engine generated noise at source from 100 to 90dB(A) (measured at 1m) with the techniques described eliminated this. Unfortunately the effects of such modifications on fuel economy and vehicle serviceability have not been fully assessed. If these modifications remain unavailable, the level of external noise treatment and attendant costs in reaching drive-by levels at or below 80dB(A) will be considerable

Estimated costs (QHV 90 Project) for complete engine/powertrain encapsulation vary between 2 and 3 percent of total vehicle costs. Weight penalties of 100 to 120kg will also be incurred. A detailed study carried out in 1978 (QHV 80 Project) estimated that operating and servicing costs for a heavy vehicle would be increased by 1 percent.

In addition to the efforts made by vehicle and engine manufacturers in reducing noise the need for component suppliers to contribute has become apparent. For example, current gearbox manufactures intend to keep transmission noise at least 10dB below vehicle drive-by noise. Modifications to gear tooth machining have already been necessary, so as to maintain this standard, rather than just to increase life or improve performance. Nevertheless acoustic enclosures may eventually be required for gearboxes.

Ancillary equipment, such as compressor blow-off valves and brake systems can add 1/2 to 1dB to previously attenuated drive-by noise. General squeaks and rattles also become very noticeable at 80dB(A). Brake system silencers, compressor blow-off valve enclosures and chassis rubbing pads will be needed if problems from intermittent noise are to be avoided.

During field work it soon becomes apparent how unpredictable sound level measurements can be. Although the theory of acoustics is precisely mathematical, environmental conditions can affect significant changes whilst being very difficult to quantify. It was noted during the QHV 90 Project that test measurements at different (certified) test sites varied by as much as 3dB. A universal (ISO) test surface has now been adopted. Despite this, surface water alone can add 1 to 2dB to test measurements. Ambient temperature changes can be almost as important. (Two engine manufacturers have stated that with a turbocharged engine a fall of 10degC in ambient temperature will cause an increase in drive-by noise of 0.5dB, due to ignition retardation; Ref B.S. AU141a). There appears to be a distinct advantage in carrying out acoustic tests in a warm, dry climate.

Unfortunately most UK truck manufacturers are compelled to carry out noise testing in Britain. Increasing competition has also forced them to be highly flexible in producing cost effective vehicles for the transportation of goods. Ultimately however the benefits to society of continual development follows a law of diminishing returns, advancements gained being outweighed by costs incurred. Also exterjal factors, such as road/tyre noise, outside the control of the vehicle manufacturer eventually predominate. The next noise directive, scheduled for 1995/96, probably reaches that point.

7. ACKNOWLEDGMENTS

The author would like to thank:-

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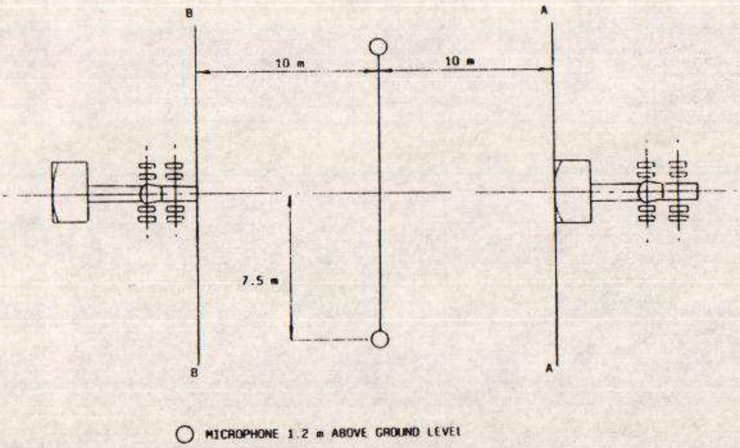
8. REFERENCES

(1) J R HASSAL & K ZAVERI, Acoustic Noise Measurement.
(2) J A HOLLMAN, T MARSHALL & A K HADDOCK, Fundemental Principals of Noise.
(3) D T ASPINALL, D HAVERGILL & G ROBERTSON, Acoustic Materials Handbook.

VEHICLE NOISE LEGISLATION

9. APPENDIX

Figure 1 - Vehicle Drive-By Noise Test Area EEC/424/84



Equation 1
Transmission Loss Factor (Cremers)

$$1/Y = \left(1 + \frac{WMS}{2PC} Z \frac{(F)^2}{FE} \operatorname{Cos}0 \operatorname{Sin}^4 0\right)^2 + \frac{W^2MS^2}{4P^2C^2} \operatorname{Cos}0 \left(\frac{-(F)^2}{FC} \operatorname{Sin}^4 0\right)^2$$

Where

MS	-	Surface Mass
FC	-	Critical Frequency
Z	-	Damping Factor
0	-	Angle of Incidence
Y	-	Transmission Coefficient
P	-	Density
W	-	Frequency

Transmission Loss TL given by

$$TL - 10 \text{ LOG } (1/Y)$$

Empirical Equation Derived by Brazely

$$TLmean - 10 \text{ LOG } \left(1 + \left(\frac{W\,MS\,\operatorname{Cos}0}{2PC}\right)^2\right)$$

(Density Term Very Important)

Equation 2
Addition of Decibels

$$SPL\ (dB) = 10 \text{ LOG } \frac{P1^2}{P0^2} \qquad - \text{ i}$$

$$\text{or} \quad 20 \text{ LOG } \frac{P1}{P0} \qquad - \text{ ii}$$

Where

SPL	-	Sound Pressure Level
PO	-	Minimum Reference Pressure
P1	-	Measured Pressure Level

Also

$$PTot = P1 + P1P2 + P2 \qquad - \text{ iii}$$

Where

PTot - Total Sound (Air) Pressure

(P1 and P2 are Measured Sound Pressures)

Assuming no constructive wave interference

$$P1P2 = \text{Zero} \qquad - \text{ iv}$$

9. APPENDIX

Figure 2 - Experimental Enclosures

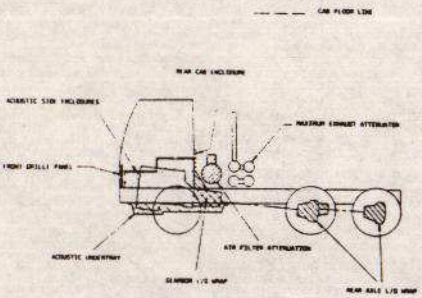


Figure 3 - Production Enclosures

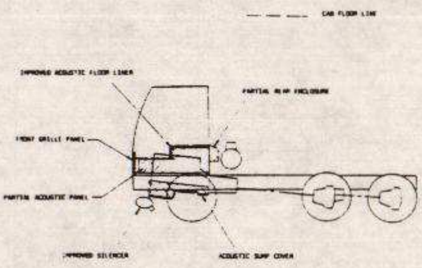


Figure 4 - Drive-By Spectrum (Standard Vehicle)

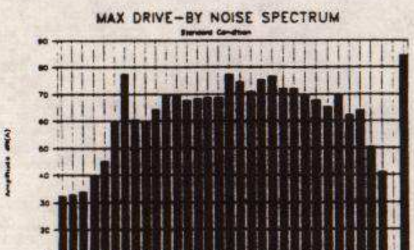


Figure 5 - Drive-By Spectrum (Experimental Vehicle)