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THE TRRL QUIET HEAVY VEHICLE

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1. Introduction

The principal objective of the QHV Project[1] at the start in 1971 was to produce two demonstration articulated vehicle tractors powered by diesel engines, one at 32.5 tonnes (32 tons) gross vehicle weight and 158 kW (212 bhp) and the other capable of 44.7 tonnes (44 tons) gvw with 261 kW (350 bhp).

For both vehicles the specification was:-

(1) Noise level to be reduced by at least 10 dB(A) with a target level of 80 dB(A). This target level was to apply for the BS 3425 : 1966 test and under any normal operating conditions.

(2) Target levels for the major vehicle components were set as in Table 1.

(3) The target noise level inside the driver's cab to be 75 dB(A).

(4) The vehicles should comply with all current and proposed legislation.

(5) Account was to be taken of the possible introduction of significant sources of additional noise, eg exhaust brakes.

(6) All essential research features of the vehicles were to be capable of incorporation in practicable production vehicles.

An important objective was the evaluation of the additional costs of quietening the final demonstration vehicle and the establishment of the variation in cost for various degrees of quietening.

The project was managed by TRRL who also carried out the tyre noise research. In addition to contributions from the manufacturers, research on noise reduction was done at the Institute of Sound and Vibration Research (ISVR) at Southampton University, the Motor Industry Research Association (MIRA) and at the National Engineering Laboratory (NEL).

Experimental vehicles in the two categories described above were built based upon Leyland (32.5 tonnes) and Foden/Rolls Royce products and a final demonstration vehicle based upon the latter is now undergoing long term trials.

The first Foden vehicle supplied for the project was a 6AR6 tractor with the capability of 44.7 tonnes (44 tons) and powered by a Rolls Royce Eagle 261 kW (350 bhp) turbo-charged diesel engine. Experimental work was carried out by MIRA and ISVR on the major components of this vehicle until it was superseded by a later design based on a 4 x 2 chassis and a gross vehicle weight capability of 38 tonnes. This vehicle, shown in Plate 1, was the basis for the research vehicle and with re-engineered quietened components, also for the final demonstration vehicle on which an engine maximum power of 320 bhp was specified.

TABLE 1

Upper limits of sound levels to be emitted by vehicle components

Research Organisation	Source	Maximum level dB(A)	
		at 1 m	at 7.5 m
ISVR	Engine including gearbox	92	77
	Air intake, exhaust system (computer modelling)	84	69
MIRA*	Cab noise (internal)	- 75 dB(A) -	
	Cooling system	84	69
	Exhaust system (development of practical systems).	84	69 (Additional target of 90 dB(C))
NEL	Final cooling fan design	84	69
TRRL	Tyre road surface noise	-	75-77

2. Research on the Rolls Royce Engine

Early in the programme Rolls Royce showed that a revised turbocharger and camshaft profile gave useful reductions in open pipe exhaust noise. This standard of engine, shown in Plate 2, was the baseline for further work.

Noise measurements were made with the engine and gearbox installed in an anechoic test cell at ISVR. It was loaded by a dynamometer mounted beyond the rear wall of the cell so that the noise from the dynamometer itself was substantially eliminated.

The effect of engine speed on noise is shown in Figure 1 and was typified by a change of slope at about 1700 rev/min. Evidence from cylinder pressure spectra, the effects of load, boost and speed on noise and the influence of timing gear noise all pointed towards overall noise being mainly determined by combustion noise at low speeds and by mechanical noise at higher speeds. This meant that a reduction in maximum engine speed from 2100 to about 1900 rev/min would give a 3 dB(A) benefit in noise; the final demonstration engine used a maximum of 1950 rev/min.

Lead covering tests were carried out to establish the rank order of the various noise sources which for this engine was (1) structure, (2) sump, (3) front end, (4) gearbox and (5) inlet. As an example of the data obtained, Figure 2 shows the spectra of noise distribution from the front of the engine obtained during the lead covering tests while Figure 3 gives the A-weighted spectra of the uncovered engine running at maximum load and peak torque speed. The noise from the front of the engine was shown to emanate largely from the timing gears and oil pump plus its drive gear. The aluminium gear case and sump were efficient noise radiators contributing to front end noise and they were also important at the sides of the engine. Replacing the timing gears by a chain drive resulted in a reduction of front end noise of 7 dB(A) on average.

As a result of these tests the timing gears were removed to the rear on the ISVR research engine where their effect was screened by the mass of the engine and the gearbox.

Vibration measurements were made at a grid of accelerometer positions on the engine surface. Typical engine surface vibration patterns are illustrated in Figure 4 for 200 Hz where one of the major bending modes occurred. Because phase information was not obtained (one transducer being used in sequence) Figure 4 does not represent vibration mode shapes but rms vibration amplitude over the surface.

The frequency content of the noise and vibration of the principal sources on the engine may be summarised as follows: Major engine structure plate modes occurred at 200, 350 and 500 Hz. There were also several peaks controlled by the response of the sump at 620, 730 and 1100 Hz and resonances of the structure occurred at 1.45, 1.6 and 1.9 kHz; the tappet cover was important at 2.8 kHz. The basic structure was a major noise radiator mainly due to the large area rather than a particularly high vibration level. The sump was the second most important radiator having high vibration levels in the frequency range 2.5 to 4 kHz.

From these and other findings the basic design of the quieter version of this engine was established.

2.1 Research engine (revised structure)

A research engine was designed by ISVR based upon a new cylinder block/crackcase structure, the cylinder head design being unchanged.

Figure 5 shows cross-sections of the original Rolls Royce Eagle and the revised ISVR structures, the basic difference between them being that the revised crankcase employed a bed plate containing the lower halves of the bearing saddles for greater stiffness. The bottom deck of the cylinder block was moved down to the lower end of the cylinders to form a more rigid section at this point. The crankcase design also provided flat sides to enable close fitting damped panels to be attached. The research engine is illustrated in Plate 3.

2.2 Characteristics of ISVR/Rolls Royce research engine

(i) Overall results: the noise level spectra of the standard and research engines are compared in Figure 3. Overall noise reductions of 9 dB(A) were obtained when averaged round the engine.

(ii) Noise radiation was greater on the near side where part of the surface was exposed for the attachment of auxiliaries.

Vibration level on the sump of the revised engine was reduced by up to 20 dB compared to the original engine above 2.5 kHz, and about 10 dB in the 0.5-2 kHz range. The lower vibration levels due to the bedplate contributed about half of the improvement up to 2 kHz while the sump isolation was the major contributor above that frequency.

(iii) Fuel pump and exhaust manifold: the noise levels of the Maximec fuel pump and exhaust manifold were reduced by 2.2 and 4 dB(A) respectively but despite these reductions they contributed proportionally more to the overall level of the research engine than to the original engine.

(iv) Rocker covers and damped panels: The vibration levels on the isolated rocker covers were an average of 31 dB lower than the original covers but this resulted in only 3.5 dB(A) reduction in noise level 1 ft above the engine. The effect of the laminated damped panels on radiated noise was complex, and depended on the frequency band and the location of the panel. The vibration level on some panels was greater than on the underlying structure, particularly at frequencies below 500 Hz. At frequencies between 2.5 kHz and 10 kHz the damped panels attached to the sides of the revised structure vibrated at amplitudes up to 20 dB less than the underlying structure. The average noise reduction on the offside of the engine due to the panels was 1.8 dB(A).

3. Exhaust system

The open pipe exhaust noise of the Rolls Royce engine supplied at the start of the project was 109 dB(A) at 2100 rev/min measured 7.5 m from the end of the pipe and 60° off the axis. With the later type turbo-charger and modified camshaft using reduced valve clearances, a reduction of 9 dB(A) in open pipe exhaust noise was obtained thus easing the difficulty of designing a quieter exhaust system.

Initial designs of the exhaust silencing system were prepared using Prof P O A L Davies' computer package[2]. However, practical development meant that a total of 22 versions of experimental silencers were built before the final design was obtained. An early system comprised two separate boxes each containing two extended inlet/outlet expansion chambers and had a total volume of 100 litres. Back pressure of this system was just within the target but it failed to meet the noise target by 10 dB(A). A 1 m long absorption section was added which improved the reduction by 6 dB(A) and only increased the back pressure by a small amount; this system was used to enable the quietening programme for the whole vehicle to be continued with a sufficiently subdued exhaust noise.

Subsequent designs were aimed at achieving improvements at lower frequencies, overall noise reductions, improved design of perforated bridges, and reduced silencer case noise, using entirely reactive silencer designs.

The size of the holes and the percentage open area in the perforated sections proved to be critical factors in the performance of the silencers, and holes of 3 mm diameter with percentage open area between 5 and 10 per cent were were adopted. The silencer pair finally fitted to the Foden vehicle is shown in Figure 6 together with test results.

The spatial average this system met the dB(A) target over the speed range from 1000 rev/min to over 2100 rev/min. The secondary dB(C) target was exceeded as the speed of the engine was reduced due to difficulty in silencing the firing frequency components of exhaust noise.

The maximum exhaust back pressure was well within the allowable maximum.

The weight of the final silencer pair was 81.7 kg and the volume (mainly determined by back pressure) was 186 litres.

4. Cooling system

The cooling system on the original Foden tractor was of the thermo syphon impeller-assisted type and used an 8 bladed axial fan.

The tests carried out by MIRA to determine the baseline cooling system characteristics concluded that in order to meet the QHV cooling system noise target the overall noise level of the standard system would need to be reduced by about 25 dB(A). A new design of cooling system was therefore required.

The fan duty in the research vehicle was considerably increased by the need to drive cooling air through the engine enclosure and a double inlet multi-vane centrifugal fan was chosen by MIRA, driven by an hydraulic motor coupled via flexible pipes to an engine driven hydraulic pump. The layout of this cooling system is shown in Figure 7. Although rig results indicated that this should have met the cooling and noise targets, when installed in the vehicle the cooling performance was well below the requirement for extreme European temperatures. After considerable efforts to improve this system, a new cooling system based on a mixed flow fan designed by the National Engineering Laboratory eventually provided the required noise and cooling performance on the demonstration vehicle. The fan rotor is illustrated in Plate 4.

5. Tyre Noise

Tyre noise has been studied at TRRL with the object of reaching an understanding of the mechanism of tyre noise and of specifying a tyre design for the Quiet

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Heavy Vehicle which would meet the target noise level of 77 dB(A).

The main conclusions of the work on tyre noise as it affects the QHV project are:-

(i) Radial-ply tyres are some 3-4 dB(A) quieter than cross-ply tyres of similar tread patterns.

(ii) The quietest tread patterns were those having 5 or more circumferential ribs and no pronounced shoulder pattern to produce strong tonal effects.

(iii) Traction tyres or dual purpose tyres having pronounced edge serrations are noisier than the tyres described in (ii) by up to 5 dB(A) in the dry and exhibit subjectively annoying tonal sounds. On wet surfaces the differences between the tread patterns tend to reduce.

(iv) A typical radial-ply tyre with 5 circumferential ribs met the QHV tyre noise target on dry surfaces up to 80 km/h, exceeding it by 2-5 dB(A) at 100 km/h. On wet surfaces the target is exceeded by 9-13 dB(A) at 80 km/h and 11-15 dB(A) at 100 km/h. On the TRRL pervious macadam surface at 80 km/h the QHV target level was met in the dry and exceeded by 6 dB(A) in the wet.

6. External and cab noise of Foden vehicle

The spectra of noise emitted during acceleration tests conducted in accordance with the procedure of BS 3425 : 1966 are shown in Figure 8 where F1 indicates the initial vehicle and F4 the final research vehicle fitted with the ISVR engine. Thus the maximum overall noise level on this test had been reduced from 92 dB(A) to 83.5 dB(A) at this stage and the demonstration vehicle now returns a figure of 80 dB(A).

Similarly the noise level in the cab was reduced from a maximum of 92 dB(A) to 78 dB(A) in the final research vehicle and is 72/75 dB(A) in the demonstration vehicle.

The passing noise of the final research vehicle coasting, cruising and accelerating in the appropriate gear is shown in Figure 9.

7. Demonstration Vehicle

The final demonstration vehicle, completed in 1978, included a version of the quiet Rolls Royce engine built to production standards, commercially made examples of the silencers, the new cooling system and other components.

The external noise of the vehicle had been reduced to 80 dB(A) and the cab noise to 72 dB(A) under normal operating conditions, levels which are below those set in the original targets for the project.

The cost penalty of quietening this vehicle to 80 dB(A) has been estimated at 7-8 per cent of the initial cost. The tare weight penalty in a production vehicle is likely to be 3 per cent which represents 1 per cent of a fully laden tractor-trailer combination. The vehicle is illustrated in Plate 5 while Plate 6 shows the upper enclosure in a tilted position and the Rolls Royce engine.

In considering the cost and weight penalties of components of this vehicle, the engine enclosure is at the top of the list with the cooling system a close second. The exhaust system represents a small problem in these respects. It is worth noting that if the engine enclosure could be removed or extensively ventilated then a much less bulky, cheaper fan could be used. However this would imply a considerably noisier vehicle or a dramatically quieter engine.

Measurements of the vehicle noise levels in various stages of assembly are shown in Table 2.

TABLE 2

Average measured noise levels for acceleration tests (dB(A))
(Measured with a sound level meter)

Vehicle condition \ Test condition	BS 3425 : 1966 2nd high		70/157/EEC 2nd overdrive	
	nearside	offside	nearside	offside
Full enclosure without fan	81.8	79.0	-	-
Full enclosure with fan	82.3	79.5	80.3	80.0
Upper and lower side enclosure but no under-tray and with fan	84.8	84.0	83.8	82.8
No enclosure with fan	87.3	86.0	86.7	85.5

Thus for the measurements with the fan running, the enclosure accounted for 5 dB(A), reducing the nearside noise level of the vehicle from 87 to 82 dB(A) in the BS test.

With no bottom enclosures the noise level of the vehicle was 84 dB(A). The standard Foden Fleetmaster cab provided some degree of shielding of the upper surfaces of the engine, but was not of course as effective as the fibre glass shields fitted to the QHV.

Measurements were also made to determine the effect of shielding the upper parts of the engine, the exhaust outlet pipe and the gearbox. This was done by lagging with a combination of lead sheet and fibre glass. None of the treatments applied to the exhaust enclosure and the top of the engine had any discernable effect on the noise levels. However, when all of the enclosure was removed, and the gearbox wrapped in lead and fibreglass the noise level of the vehicle in the BS test was reduced by 1 dB(A) by this covering.

8. In-Service Trials

The demonstration QHV has been operated by a general haulage contractor since November 1979 being treated as far as possible as a normal fleet vehicle, the principal difference being that its operation is closely monitored and a test at TRRL of its noise and fuel consumption is performed during its 3 monthly service. Results from the first five months of this trial are summarised in Figures 10-12. Average trip load factor was 0.43 with the majority of journeys under 100 km. As the trial progresses it is intended that these factors will be increased. Fuel consumption shows the variability typical of trials on the road, even when divided into fully laden and unladen conditions. The curves shown are for a computer simulation of this class of vehicle with no special adjustment for noise reduction features. It can be seen that the practical results obtained are consistent with the computed curves except that unusually good fuel consumptions have been measured at high average speed. Further analysis will be required before this can be confirmed and special attention to the type of load involved (if any) may help to explain this if aerodynamic drag is fortuitously low.

In general the operating economics of the vehicle are looking quite favourable with only a small effect on maintenance costs due to the quietening features though it is too early in the trial to give any comprehensive information on this.

Reference 1. Tyler, J W. TRRL quiet vehicle programme. Quiet Heavy Vehicle (QHV) Project. Department of the Environment, Department of Transport TRRL Supplementary Report 521.

Reference 2. Davies, P O A L. Silencer Design Computer Package SECT F - Compeda Ltd Stevenage.

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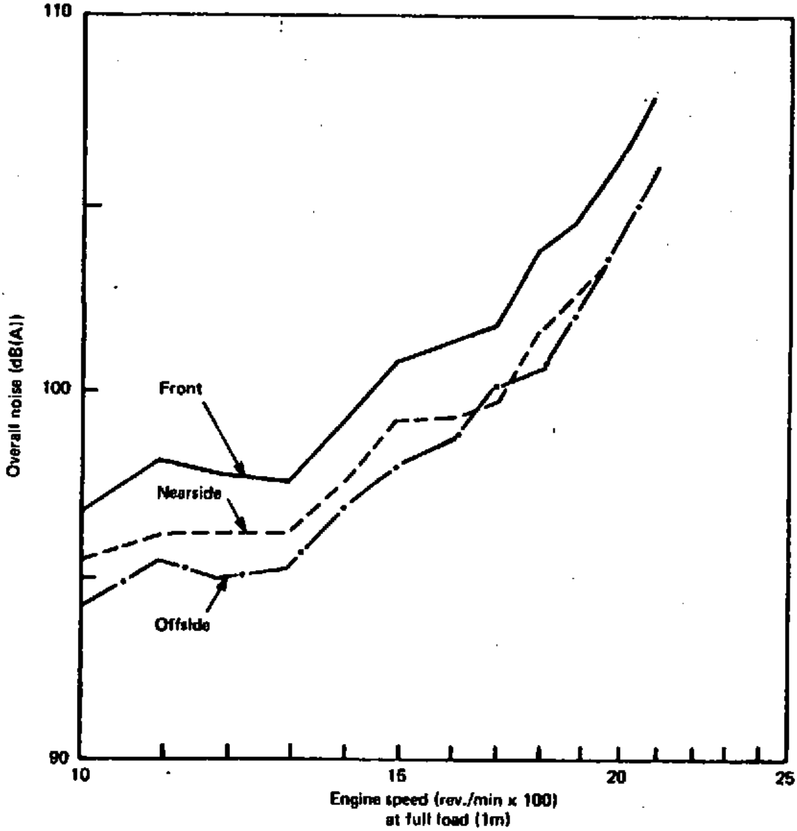


Fig. 1 OVERALL NOISE vs ENGINE SPEED
(Rolls Royce standard engine)

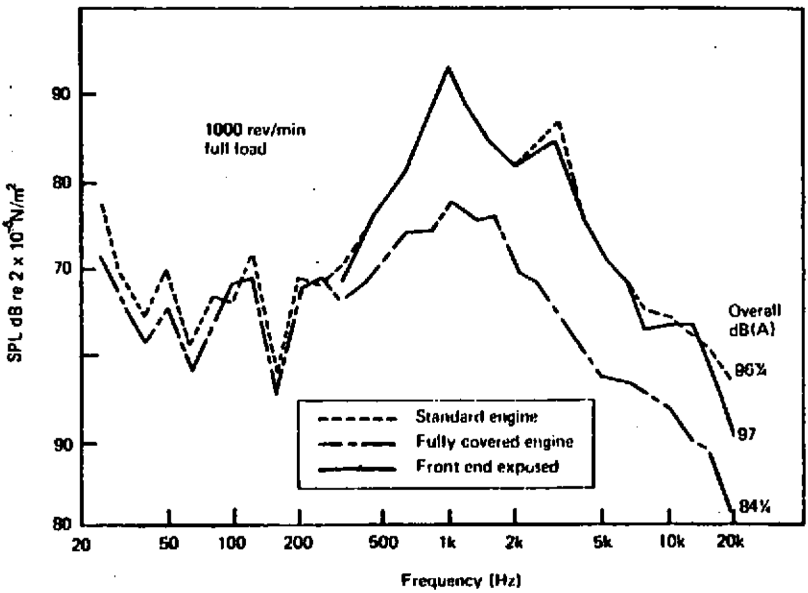


Fig. 2 NOISE CONTRIBUTION FROM FRONT OF ORIGINAL ROLLS ROYCE EAGLE ENGINE

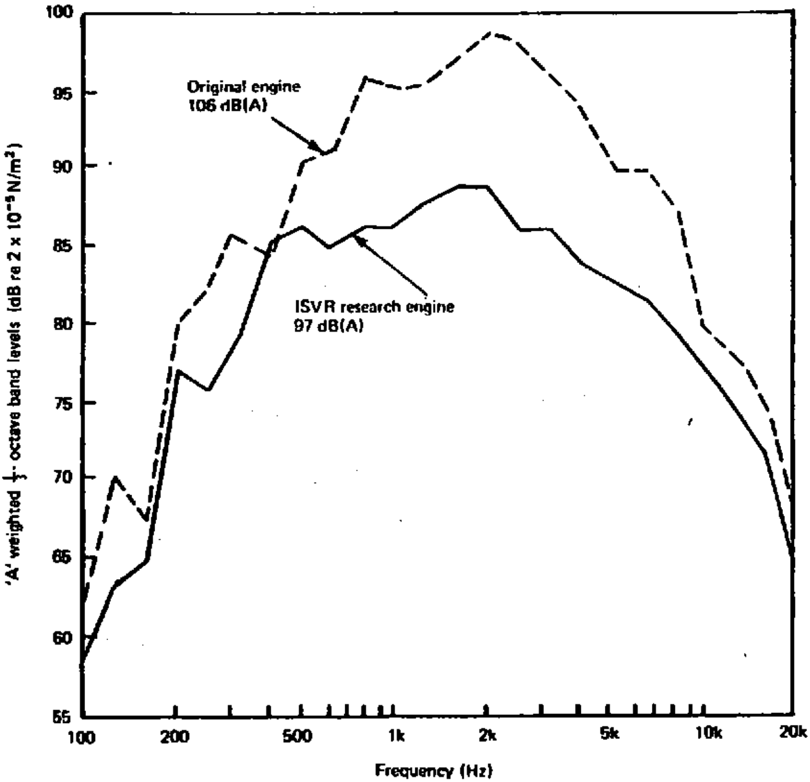
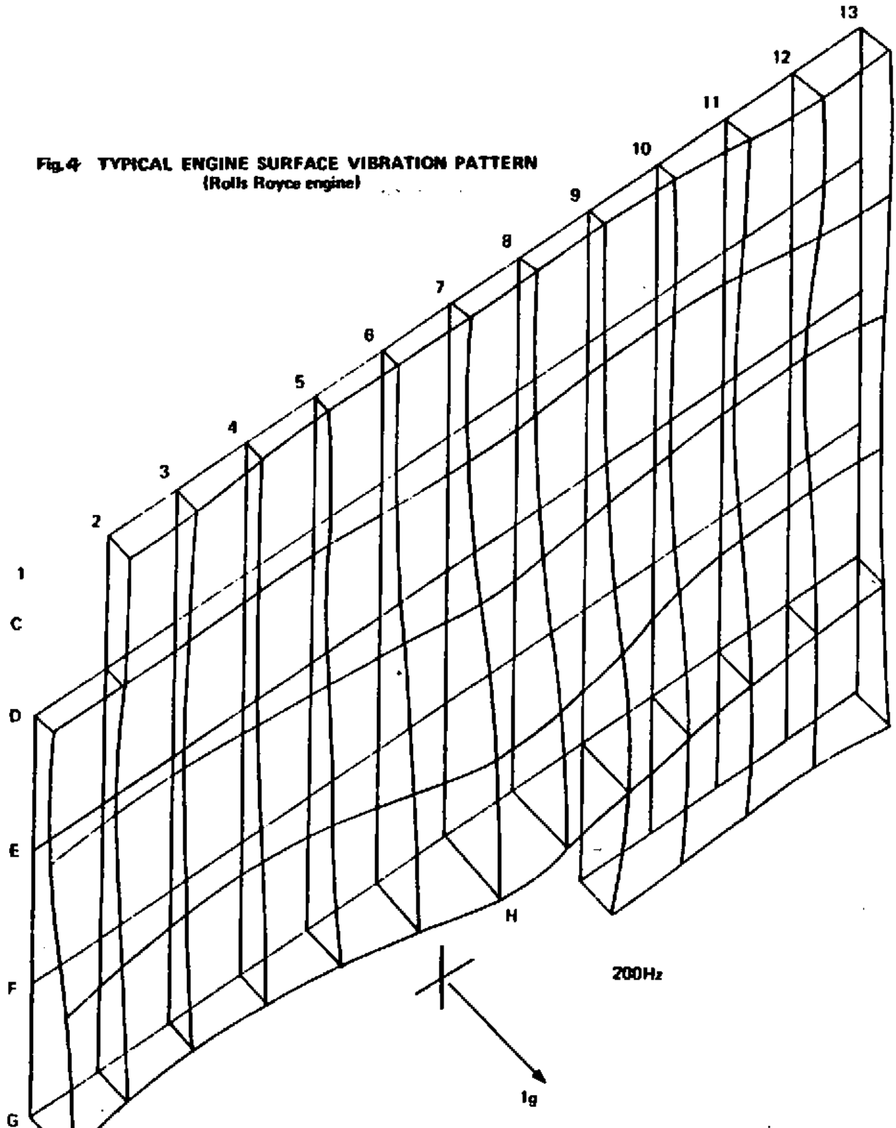


Fig. 3 COMPARISON OF SPECTRA OF ORIGINAL AND RESEARCH ROLLS ROYCE EAGLE ENGINE AT RATED CONDITIONS (Near side)

Fig.4 TYPICAL ENGINE SURFACE VIBRATION PATTERN
(Rolls Royce engine)



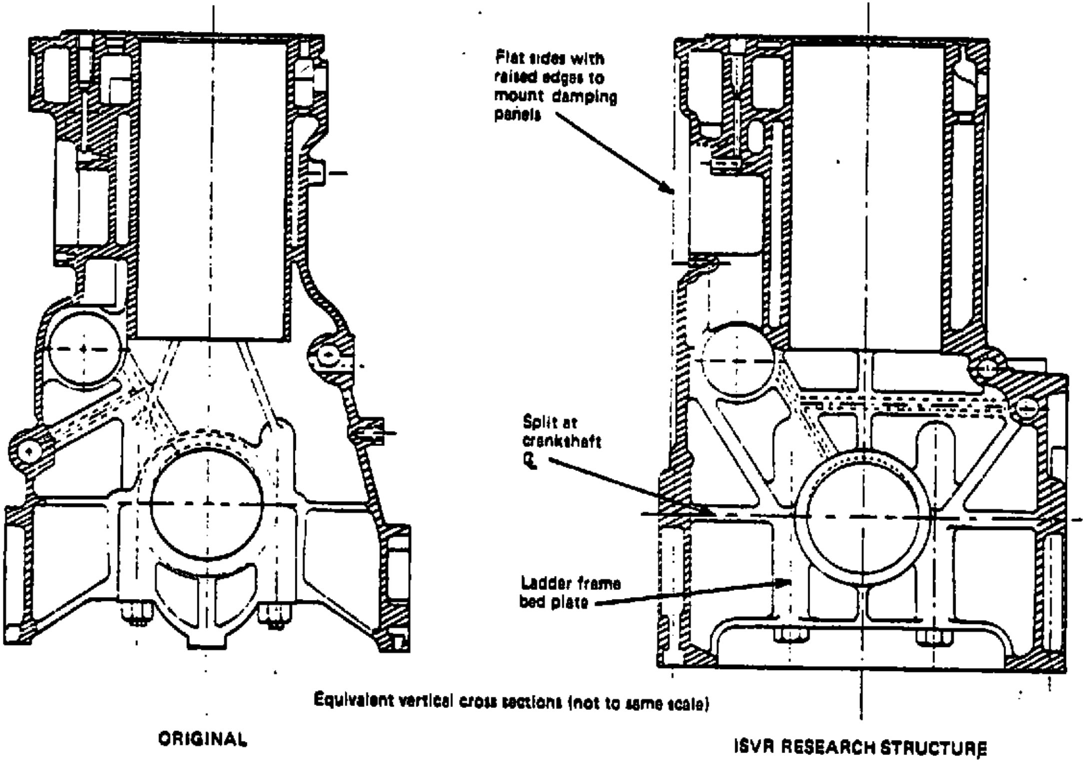


Fig. 5 COMPARISON OF ENGINE STRUCTURES
(Rolls Royce engine)

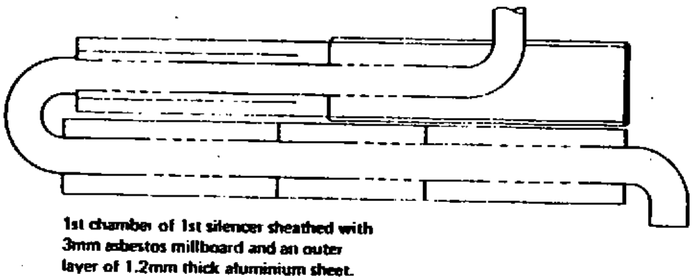


Fig. (a) FINAL ISVR/MIRA SILENCER DESIGN

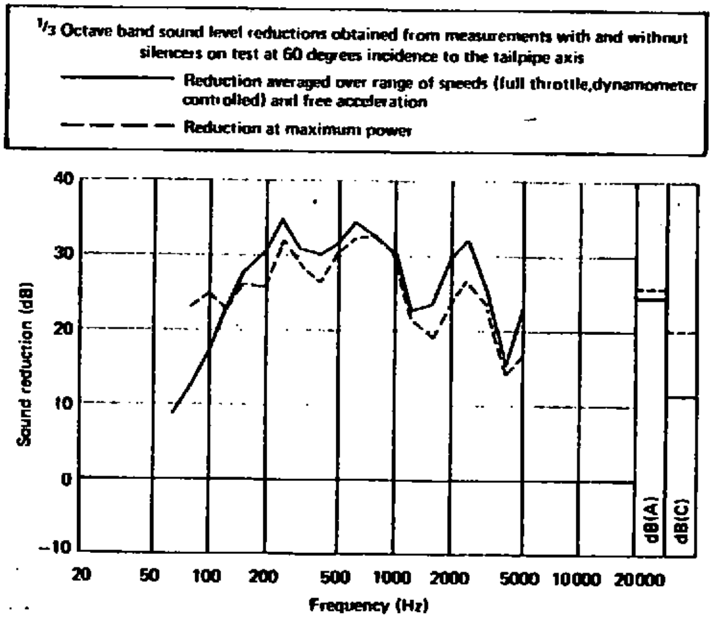


Fig. 6:(b) TEST RESULTS OF FINAL MIRA/ISVR SILENCER DESIGN

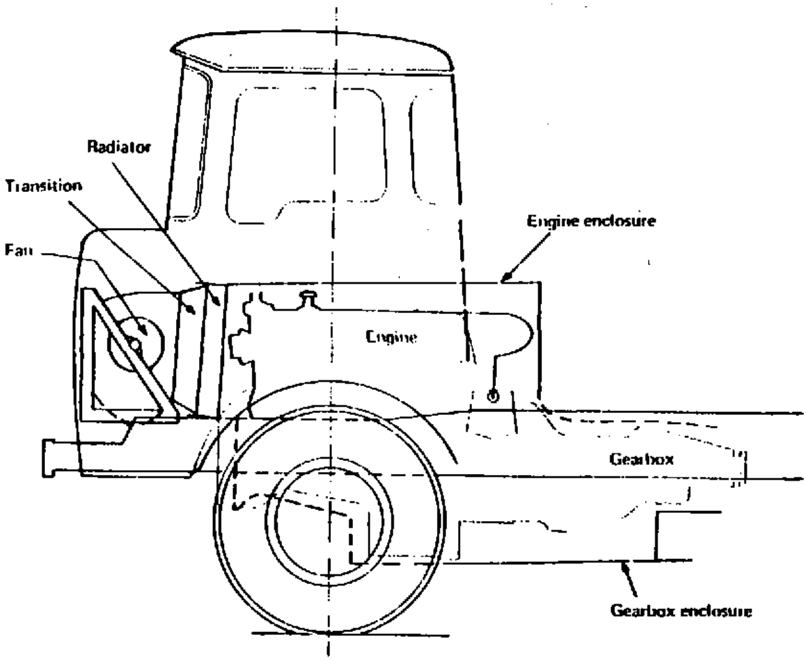
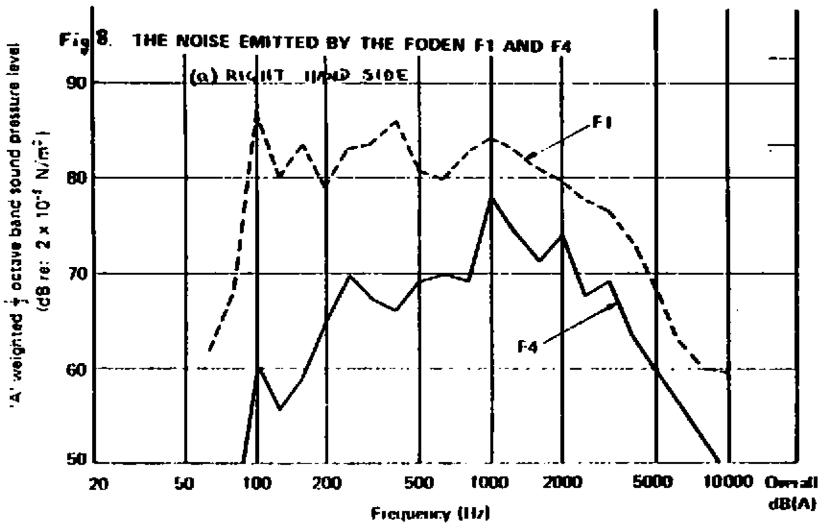
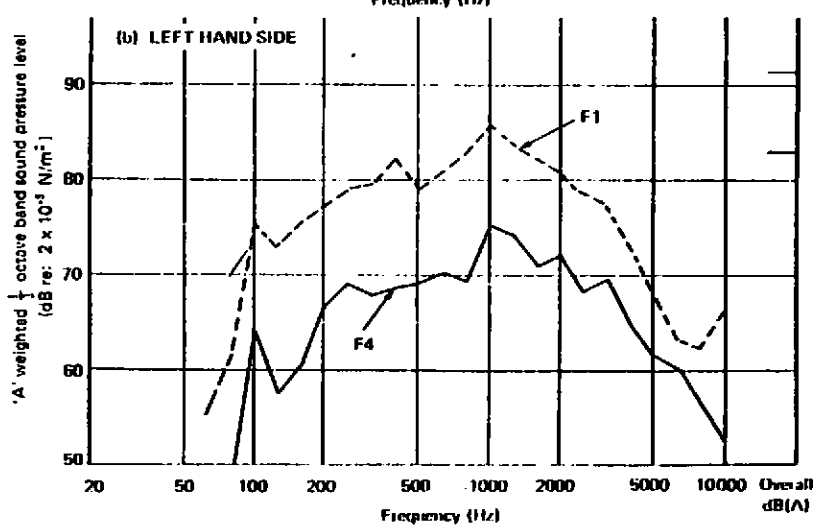


Fig. 7 GENERAL ARRANGEMENT OF THE FRONT END OF THE FODEN F2 SHOWING CENTRIFUGAL FAN COOLING PACKAGE

Fig 8. THE NOISE EMITTED BY THE FODEN F1 AND F4





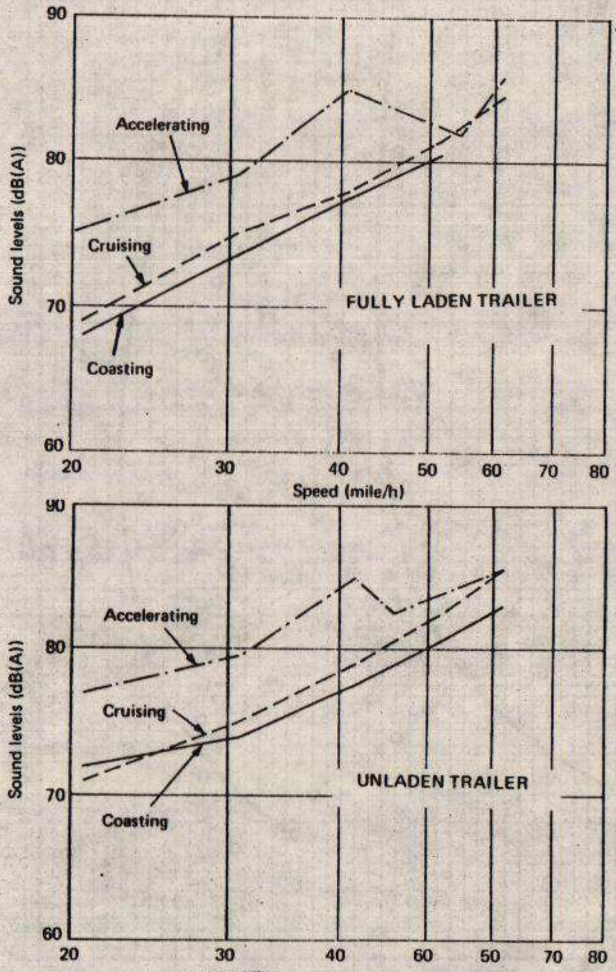


Fig. 9 PASSING NOISE OF F4 WITH TRAILER

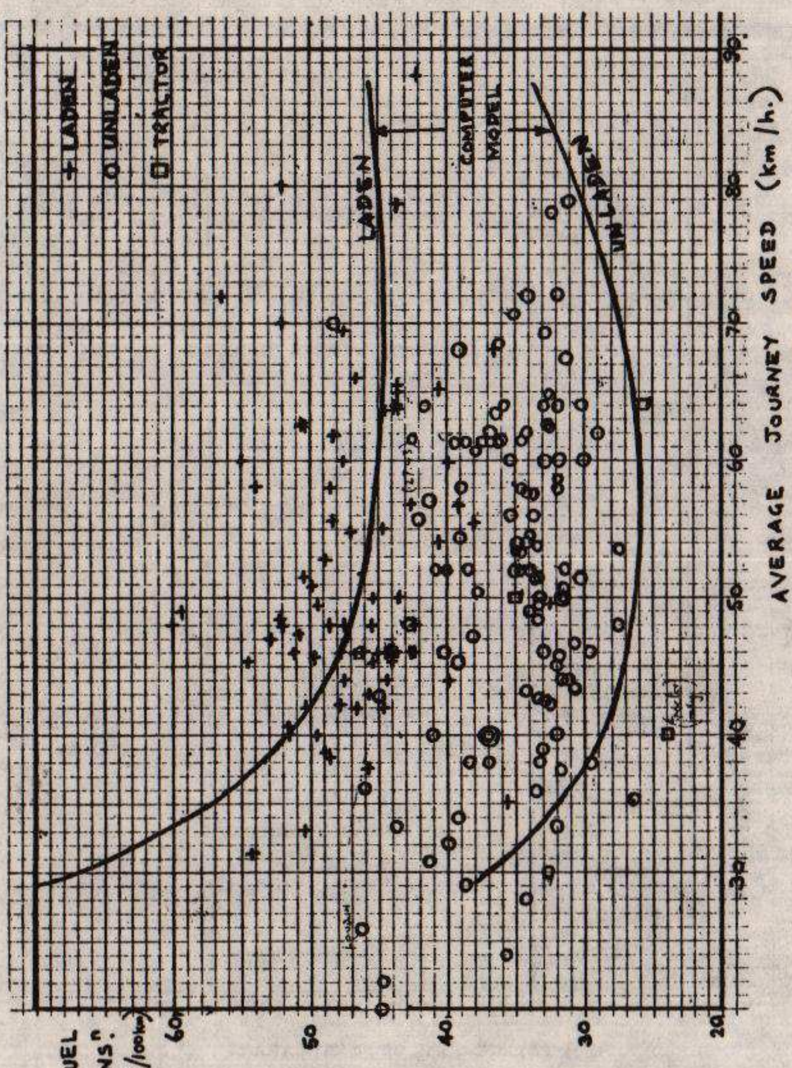


Fig. 10. FUEL CONSUMPTION OF QHV.

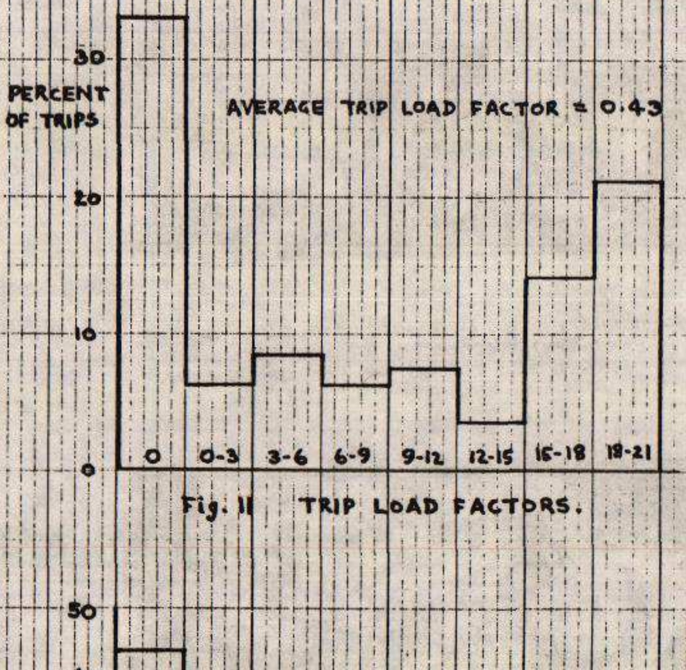


Fig. 11 TRIP LOAD FACTORS.

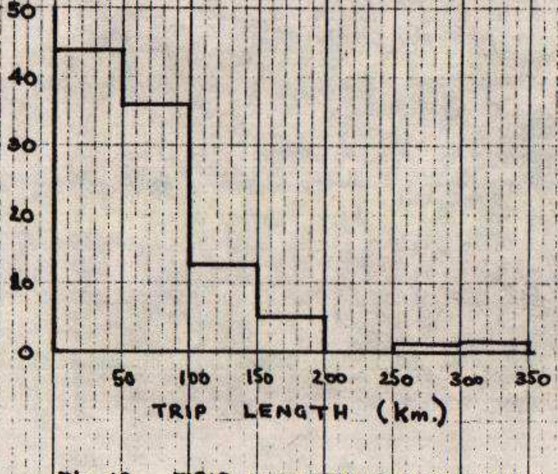


Fig. 12. TRIP LENGTH DISTRIBUTION.

Neg. no. B260/79

Plate 1 ORIGINAL FODEN 4 x 2 TRACTOR WITH EARLY DESIGNS OF EXPERIMENTAL COOLING AND EXHAUST SYSTEMS (F3). THIS VEHICLE FORMED THE BASIS OF THE FINAL RESEARCH VEHICLE F3

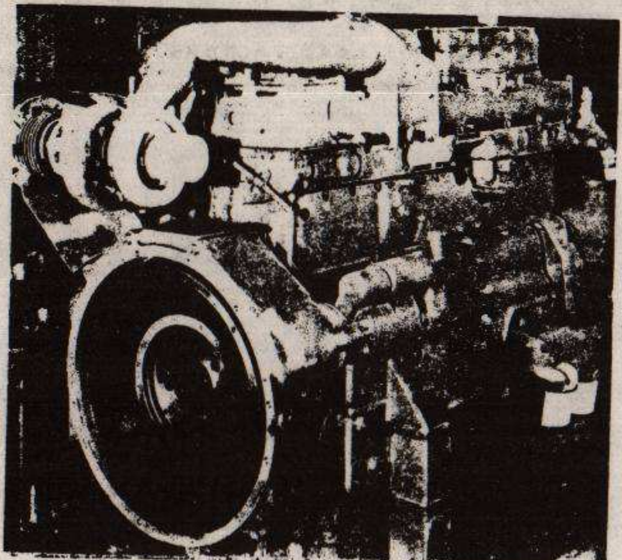


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Plate 2 ORIGINAL ROLLS ROYCE EAGLE ENGINE

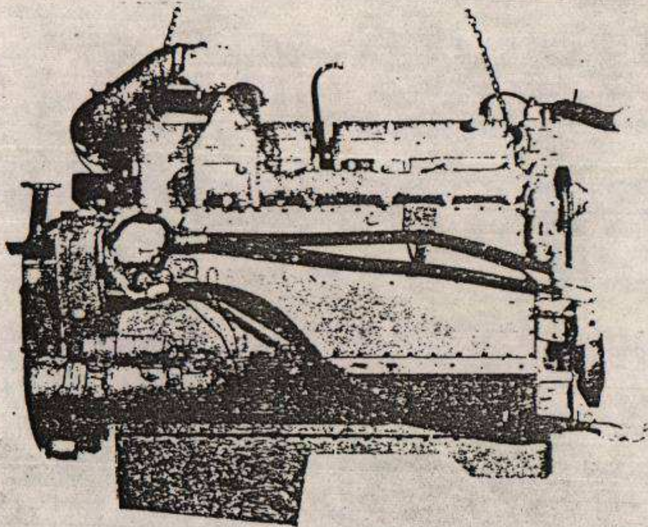

Plate 3 ISVR/ROLLS ROYCE RESEARCH ENGINE SHOWING LAMINATED DAMPING PANELS, BEDPLATE AND ISOLATED SUMP

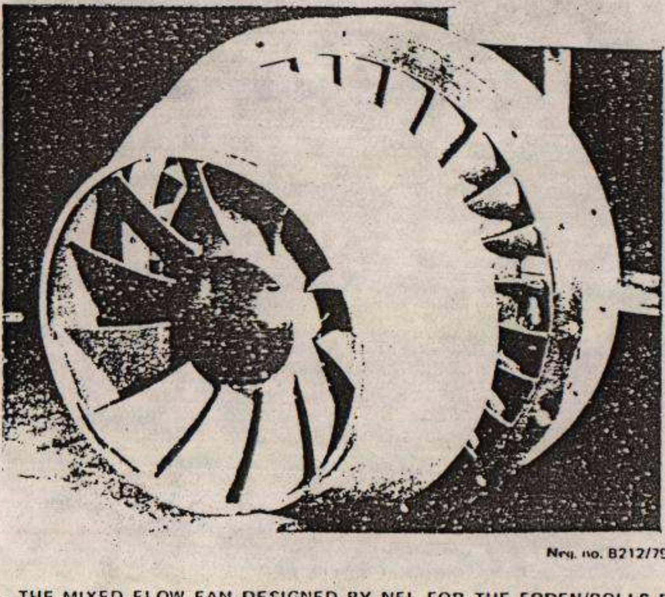

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Plate . THE MIXED FLOW FAN DESIGNED BY NEL FOR THE FODEN/ROLLS ROYCE VEHICLE (Impellor at front, stator at rear)

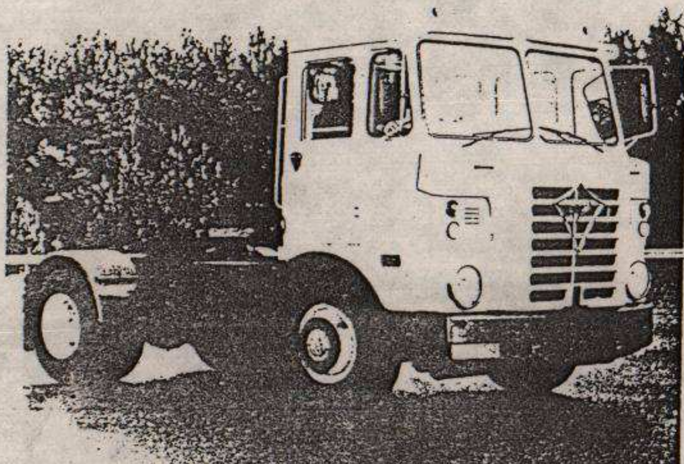

Plate FODEN/ROLLS ROYCE DEMONSTRATION QUIET HEAVY VEHICLE

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Plate 6 VIEW OF DEMONSTRATION VEHICLE SHOWING ROLLS ROYCE QUIET ENGINE AND UPPER ENCLOSURE IN FULLY RAISED POSITION