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**Urban Mass
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Noise Control of the Contemporary Transit Motorbus

Tri-County Metropolitan Transportation
District of Oregon
Portland Oregon

May 1984
Final Report

UMTA Technical Assistance Program

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16. Abstract

A broad range of topics relating to the noise control of ordinary transit motorbuses is presented. The work is an outgrowth of Portland's Transit Mall engineering. Topics include noise ratings of various makes and models, source analysis and treatment, testing techniques, performance benchmarks, environmental noise prediction, busyard sound barrier, and noise control strategies.

Two Service Bulletins detailing noise treatment kits have been previously published:
Noise Reduction Retrofit for a "New Look" Flxible Transit Bus,
UMTA-OR-06-0005-80-1, September 1980.
Noise Reduction Retrofit for a "New Look" GMC Transit Bus,
UMTA-OR-06-0005-81-1, November 1981.

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PREFACE

From 1975 to 1982, many programs and projects related to transit motorbus noise control were carried out in Portland, Oregon, under the supervision or with the cooperation of Tri-Met. Most of the funds for this work came directly or indirectly from UMTA. The greater amount of the work was done under Grant Contract OR-06-0005 which was primarily funded by the Urban Mass Transportation Administration, with some support from the Environmental Protection Agency. The intent of this report is to gather together all the valuable engineering information that flowed from this time, going beyond the requirements for a final engineering report by OR-06-0005.

The contents of this report will be of interest to those transit bus operators, government agencies, manufacturers, planners, and academicians seeking acoustic information for predicting, controlling, and reducing bus noise. They will also find a great deal of useful information on related topics of automotive engineering pertaining to the transit motorbus such as cooling system behavior, fuel mileage, road performance, and engine aspiration. Successful bus anti-noise treatments and a busyard sound barrier are described.

Acknowledgement is made of the special contributions by Gary Brentano and Richard Woods of the Tri-county Metropolitan Transportation District of Oregon (Tri-Met) who managed most of the projects.

The work has been directed by the U.S. Department of Transportation through the efforts of Patrick J. Sullivan for the Urban Mass Transportation Administration.

Conversion Table to SI Units

This publication uses customary English units for the convenience of engineers and others who use them habitually. The table below is for the reader interested in conversion to SI units. For additional information see:

(1) NBS LC1078, Dec., 1976, "The Metric System of Measurement".
(2) Z210.1-1976, "ASTM/IEEE Standard Metric Practice".

Quantity	To convert from	To	Multiply by
Length	inch	m (meter)	2.540×10^{-2}
	foot	m	3.048×10^{-1}
	mile	m	1.609×10^{3}
Area	in^2	m^2	6.452×10^{-4}
	ft^2	m^2	9.290×10^{-2}
Volume	in^3	m^3	1.639×10^{-5}
	ft^3	m^3	2.832×10^{-2}
	gallon	m^3	3.785×10^{-3}
Temperature	°F	°C	$t_{°C} = (t_{°F} - 32)/1.8$
T. difference	$\Delta t_{°F}$	K	$\Delta T_K = \Delta t_{°F}/1.8$
Mass	pound	kg	4.536×10^{-1}
	ounce	kg	2.835×10^{-2}
Pressure	psi	Pa	6.895×10^{3}
	in H_2O	Pa	2.488×10^{2}
	in Hg	Pa	3.386×10^{3}
	mmHg	Pa	1.333×10^{2}
Energy	Btu	J	1.055×10^{3}
	MBtu	J	1.055×10^{9}
	kWh	J	3.600×10^{6}
	ft • lbf	J	1.356×10^{0}
	kilocalorie	J	4.187×10^{3}
Power	Btu/h	W	2.931×10^{-1}
	hp	W	7.457×10^{2}
Flow	gal/min	m^3/s	6.309×10^{-5}
	ft^3/min	m^3/s	4.719×10^{-4}
Density	lb/ft^3	kg/m^3	1.602×10^{1}
	lb/gal	kg/m^3	1.198×10^{2}
Heat Capacity	Btu/(lb • °F)	J/(kg • K)	4.187×10^{3}
	$Btu/(ft^3 \cdot °F)$	$J/(m^3 \cdot K)$	6.707×10^{4}

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1. INTRODUCTION

1.1 BACKGROUND

In order to assist in the rejuvination of downtown Portland, Oregon, a 10-block transit mall was constructed in the heart of the city. In the planning of the mall, the Environmental Impact Statement predicted a 9 dBA increase in ambient sound levels. In order to make the mall acceptable to the general public and to negate this dire engineering forecast, a program of bus noise control was initiated under the joint sponsorship of the USDOT UMTA Office of Technical Assistance and the Environmental Protection Agency. The program focused on the noise levels of existing TRI-MET buses and on measures that could reduce those levels. TRI-MET's investigations revealed that its buses were not noisy when compared to newly manufactured vehicles nor to vehicles operated by other transit systems. Nevertheless, tests and treatment of GMC and Flxible R New Look buses indicated that modest improvements could be made in the noise levels of well maintained vehicles.

This report summarizes the work and findings of the TRI-MET efforts, providing information on pertinent noise sources and the control of their outputs, both in general and specifically as applied to the two makes of buses comprising the bulk of the TRI-MET fleet.

The following material is divided into two major sections: the first deals with contributing noise sources per se and their control, and the second deals with specific automotive factors which enter into any consideration of the bus noise problems.

The compilation of TRI-MET investigations provides much information applicable to specific and immediate problems. In addition, this experimental information would seem to be useful to many in the bus transit field who need a broad understanding of bus noise sources and their control.

1.2 TYPICAL TRI-MET BUS

This report deals with the GMC and Flxible R New Look buses, the most common buses in the TRI-MET fleet. Typical characteristics of the bus are:

Weight..........	24,000 lb empty
Average age.....	7 years
Expected life...	15 years
Purchased with..	Mostly federal and state funds
Number wheels...	4 (rear are dual-tired)
Number seats....	45
Length..........	40 feet
Width...........	8 1/2 feet
Engine..........	Naturally aspirated vee 2-stroke cycle 570 cubic-inch (8 cylinder) Diesel
Drive...........	Torque converter Automatic 2-step planetary transmission Rear axle differential

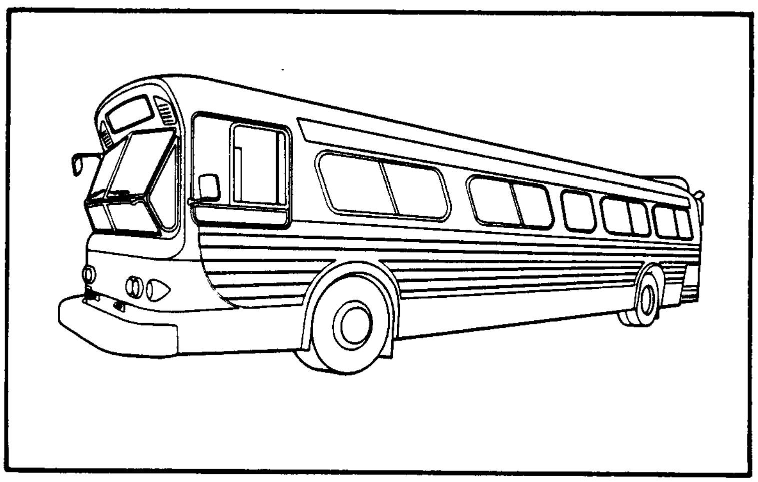


FIGURE 1.

A TYPICAL TRANSIT MOTORBUS

Operational reliability has the highest priority in bus design. Thus, the diesel engine is the natural choice for the motorbus power plant. In addition, the necessity for frequent, smooth speed changes to moderate road speeds from standing starts on what might be steep hills, and, again, because of an association with long life and operational reliability, an automatic transmission is the diesel engine's natural companion.

The bus engine and cooling system are located under the rear seat in a notch formed in the reinforced monocoque body. Access is through a rear door and right side door. Although mountings have been provided, the compartment's bellypan is absent.

The radiator fan is mounted on the front of the engine and faces toward the left side of the bus. The engine drives it through a thermally-automatic modulated speed hydrodynamic clutch.

The underslung exhaust muffler is located just forward of the engine compartment. The exhaust pipe runs up the left rear corner of the bus to an outlet aimed skyward.

The current makeup of the TRI-MET fleet is given in Table 1.

TABLE 1.

THE TRI-MET BUS FLEET
November, 1982

Year	Qty	Make	Model	Engine	Trans	Seats	Width-in.	Tare-lbs.
1961	5	GMC[1]	TDH-4517	6V-71N	VH	45	96	18,500
1961	5	GMC	TDH-5301	6V-71N	VH	53	102	20,380
1963	22	GMC	TDH-4519	6V-71N	VH	45	96	18,900
1963	20	Flx[2]	M50HYM	6V-71N	640	53	102	21,780
1964	15	GMC	TDH-4519	6V-71N	VH	43	96	18,500
1964	15	GMC	TDH-5303	6V-71N	VH	53	102	21,080
1965	15	GMC	TDH-4519	6V-71N	VH	45	96	19,500
1966	20	GMC	TDH-4519	6V-71N	VH	45	96	19,200
1970	3	GMC	T6H-4521A	6V-71N	VH	45	96	21,380
1971	25	GMC	T8H-5305A	8V-71N	VH	51	102	21,860
1971	50	Flx	111DD-D51	6V-71N	VH-9	43	94	21,300
1972	135	Flx	111DC-D061	8V-71N	VS2-8	42	102	21,500
1973	20	Flx	45102-8-1	8V-71N	VS2-8	42	102	22,100
1973	20	GMC	T8H-5307A	8V-71N	VS2-8	49	102	22,620
1974	80	Flx	53102-8-1	8V-71N	VS2-8	49	103	23,600
1976	100	AMG[3]	10240B-8	8V-71N	V-730	49	102	24,510
1980	11	GMC	T6H4523N	6V-71N	V-730	40	96	21,200
1981	87	C/I[4]	286.02	NHHTC-290	HT-740D	61	102	36,500
1982	75	GMC	T80204	6V-92TA	V-730	43	102	36,900
	723							

[1]General Motors
[2]Flxible®
[3]American General
[4]Crown-Ikarus

Engine	Stroke-cycle	Cylinders	Configuration	Aspiration
6V-71N	2	6	vee	natural
8V-71N	2	8	vee	natural
NHHTC-290	4	6	pancake	turbocharged
6V-92TA	2	6	vee	turbocharged

1.3 The Mission

Major transit ridership is composed of people going to and from their daily jobs in the central business district. As many as 50% of the downtown commuters will be riding the bus during the morning and afternoon rush hours. Were it not for public transit, overwhelming peak hour automobile traffic congestion with such undesirable side effects as overlength trip times, air pollution, and declining property values would significantly increase together with the attendant side effects. Transit malls are an approach to abating or perhaps reversing this deteriorating automobile traffic situation.

The typical transit motorbus is on a "stop-and-go" mission. After boarding riders, it accelerates at, or nearly at, full throttle until again merged with traffic. It cruises with this flow until coming to the next predetermined stop. Its average road speed is less than 20 mph.

Even though they are constantly being adjusted, bus routes generally follow the radial pattern of their predecessors, the electric trolley cars. They are like spokes of a wheel, radiating outward along busy arterials from the central city hub to turnarounds in relatively quiet residential neighborhoods.

Portland is not unlike many U.S. cities. Although the terrain is largely flat, there are numerous short segments of steep streets along bus routes in the range of 10% to 15% grade that must be ascended and descended.

Table 2 gives representative operating statistics.

TABLE 2.

PER BUS TRI-MET OPERATING STATISTICS
FY 1977-79

Annual mileage	38,459 miles
Annual deployed time	2,714 hours
Service factor	[1]271 days per year
Annual usage	75%
Average road speed	[2]16.7 mph
Daily mileage	142 miles/day

[1] Based on 10 hours per day average.
[2] Based on running time being 85% of deployed time.

One of the problems with the typical motorbus is the inherent noisiness of its diesel engine. Investigation show that, while the typical motorbus canot be made downright quiet without expensive major alternation, certain adroit refinements can minimize bus noise. At the very least, attention to particular maintenance targets can prevent individual faulty buses from "sticking out like sore thumbs."

2. NOISE AND NOISE CONTROL

2.1 General

Sound levels are usually expressed in terms of A-weighted decibels. The sound level meter consists of a microphone, an amplifying electronic network, and a readout. A decibel of sound is 20 times the logarithm (power of ten) of the ratio of the measured sound pressure to a reference sound pressure. The reference is the threshold of human hearing. A sound having a thousand times the pressure of the human threshold, would have a decibel rating of 60 dB. A-weighting means that the signal is sent through a filter that down-grades low frequency sound corresponding to the response of the human ear.

One does not use simple arithmetic with decibels. Every time sound pressure doubles, the sound level increases by 3 dBA. 80 dBA + 80 dBA = 83 dBA.

If one sound is 10 dBA louder than another, the sound level of the two together is not detectably more than the greater sound. When a particular noise source is 10 dBA less than the overall noise, then it is unimportant.

Distance has a predictable effect on sound level. Each time one doubles the distance to a sound source radiating from a point in an echo-free space above a reflecting plane, the sound level drops by 6 dBA. If the source were a line of sound, doubling distance would drop the level by 3 dBA.

When trying to stop noise by putting a box around the source, one should keep these things in mind:

- A lot of noise comes through a small hole. An air leak is usually a noise leak too.

- "Live" panels (they ping when you strike them) can be "killed" by stiffening them or by gluing something heavy to them.

- The walls of the box have to be relatively heavy. Usually a pound or two per square foot is enough for an engine enclosure.

- Some kind of sound absorptive lining inside of the box is good because it reduces echoes inside. If it becomes less noisy inside the box, it will become less noisy outside the box.

- If you can't put a complete box around a source, at least try to put up barriers to cut off line-of-sight noises. This is better than leaving open holes.

Noise radiates in all directions from a typical bus, but not evenly. Less noise radiates forward because the bus body is a barrier to engine noise in the rear. The bus is noisier on the left side than on the right, not because the exhaust stack is on that side, but because the radiator opening is there...an open door to noise from the engine compartment. Fortunately, the boarding passengers are exposed to right side noise which is quieter and easier to treat than left side noise.

Engine noise is sensitive to both speed and power. It has a harsh, unpleasant quality. It is the most difficult noise source to treat and is therefore usually the dominant source after the overall bus has been treated. It can be helped by the engine manufacturer through smoothing the onset of combustion, by stiffening the block, and by curing "hot-spots". This takes a long time and is expensive. Other treatments are turbocharging and encapsulation. The simplest treatment is to reduce speed and power output.

The fan makes a whirring sound, coming from where the blade tips chop the air. The sound is highly sensitive to fan speed, so a modulating speed drive is better for noise control than an on-off drive. At least sometimes you will hear the fan if it has an on-off drive, but you might never be bothered by fan noise with a modulating speed drive. Nothing much can be done to cure fan noise except to slow the fan down. This is most easily done at the factory where a larger radiator can be incorporated to compensate for lower fan speeds.

Exhaust outlet noise is more mellow and musical. Most uninitiated believe they are hearing exhaust noise and blame the muffler when a loud bus goes by. It might well be a break in the exhaust piping upstream from the muffler, or more often, it is just plain engine noise they are hearing. Exhaust noise is relatively easy to treat. All one needs is a proper muffler. With turbocharged engines, there is less need for muffling, but there is more need for low back pressure. This leads to the problem of finding room for the adequate muffler. To have low back pressure and good silencing, a muffler can't be kept small.

Muffler and exhaust pipe shell noises might be revealed as treatment of the louder engine, exhaust, and fan noises progresses. Common practice is to double the skin of the muffler body when used with naturally aspirated engines. The need is less with turbocharged engines. Pipe shell noise does not ordinarily require treatment, but there might be significant noise radiating from flexible tube sections. At times, significant noise can escape pipe joints when old style u-bolt-and-saddle clamps are used. Modern stretchable overlapping band clamps solve this problem.

Tire noise is unimportant at speeds below 40 mph for the ordinary bus. Even after treatment, tire noise is masked out at speeds below 30 mph. Body noise can be a nuisance if there are loose doors and windows.

Inside the bus, no one needs to be told that it is louder in the rear than up front. The source of noise heard is almost always the engine...not the exhaust and not the fan. The noise from the structureborne path is at least as great as that from the airborne path. It is important to keep the rear seat engine compartment access hatch well fitted and sealed tight. And, replace those worn out engine mounts.

A plot of the noise that a typical bus can put out is shown in Figure 2. The sound measurements were made under standard conditions set out in Appendix A in which the observer (microphone) was 50 feet from the bus track centerline. The bus, under full acceleration, makes its first, automatic, upshift just as the rear of the bus passes by.

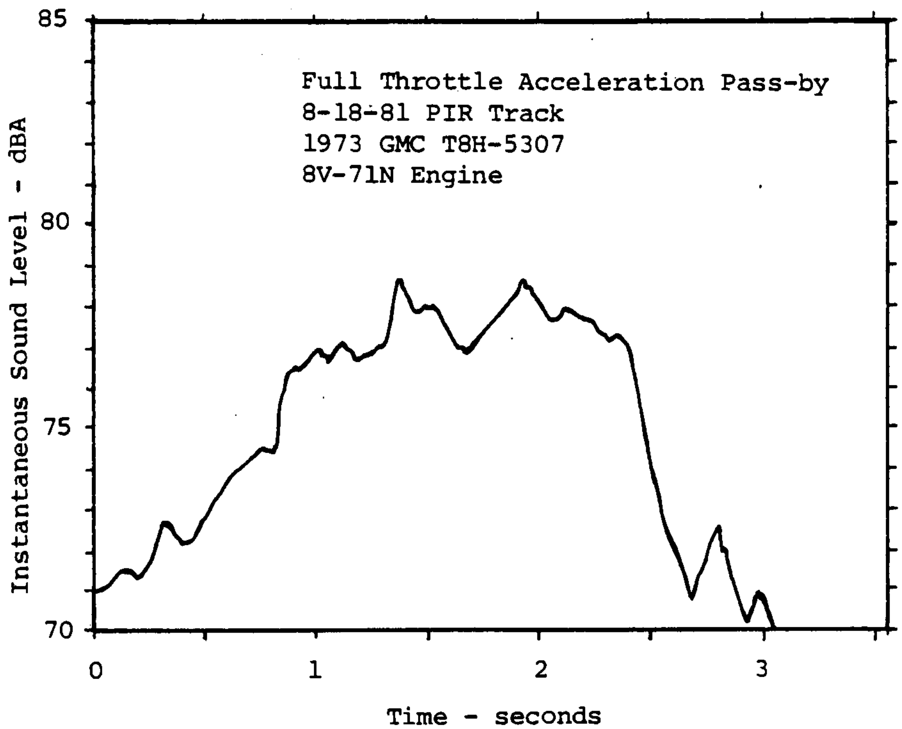


FIGURE 2.

SOUND LEVEL DURING BUS PASS-BY[1]

[1] Data courtesy of Freightliner Corporation.

The sound from a motorbus covers a wide spectrum as seen in Figure 3. Its main energy spreads from 100 Hz to 3,000 Hz with a high concentration from 500 Hz to 1,000 Hz. While some peaks coincide with multiples of engine firing order, others do not.

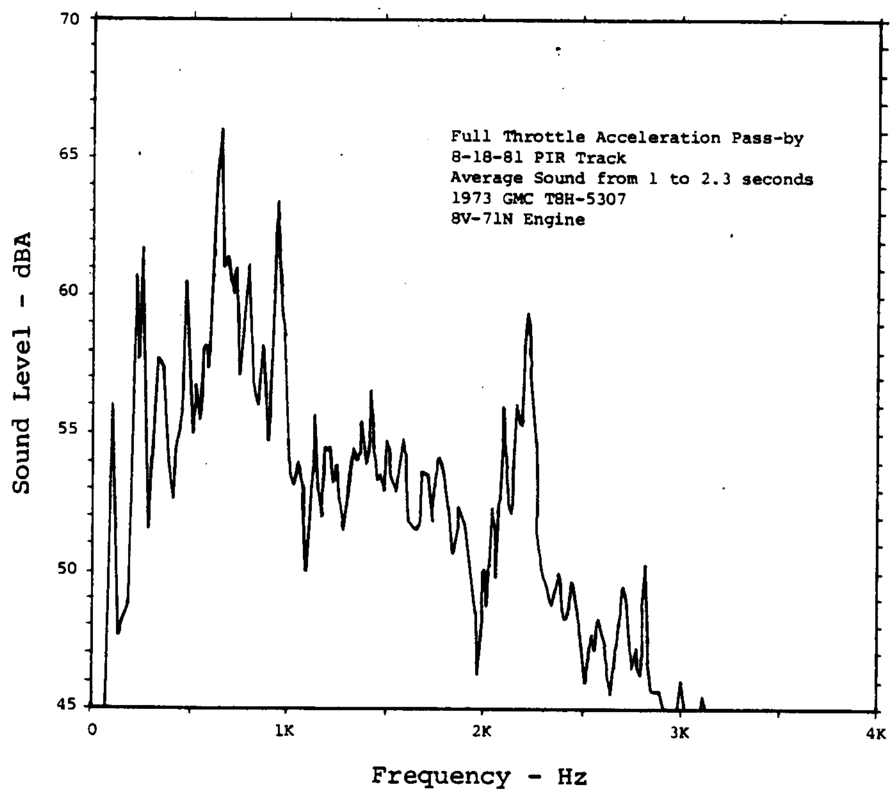


FIGURE 3.

SOUND SPECTRUM OF A MOTORBUS[1]

[1] Data courtesy of Freightliner Corporation.

As can be seen in Figure 4, sound from the typical motorbus does not diverge evenly. The left side is louder than the right due to the engine compartment sound escaping through the radiator opening, not because of the exhaust outlet being on that side. The front is shielded from the engine compartment sound by the bus body. Sound level in front is less than that at the rear for the same measurement distance away. These attributes are shown in Figure 4, in which distance to bus for equal sound level contours around a bus are plotted.

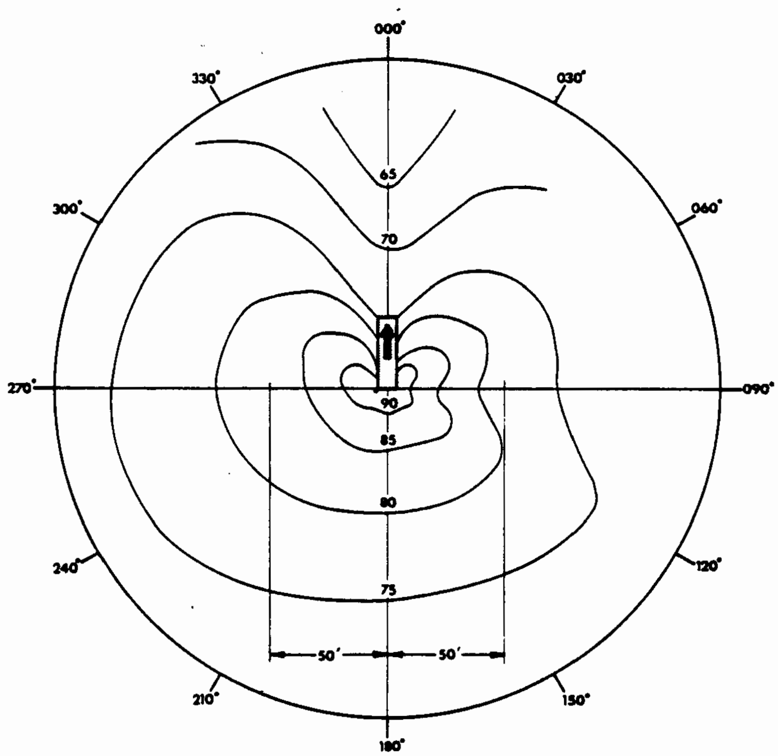


FIGURE 4.

DIRECTIVITY OF BUS SOUND

Parked on Asphalt Apron
Portland International Airport July 1975
1972 Flxible® 111DC-D061
8V-71N Engine Idling at 2,150 rpm
Contours of Equal Sound Level - dBA

One can expect sound to spread with distance away from a motorbus on ordinary pavement in accordance with simple theory. Figure 5. shows how sound from a bus decreases with distance as if the engine compartment sound were a point source... and the only source of consequence.

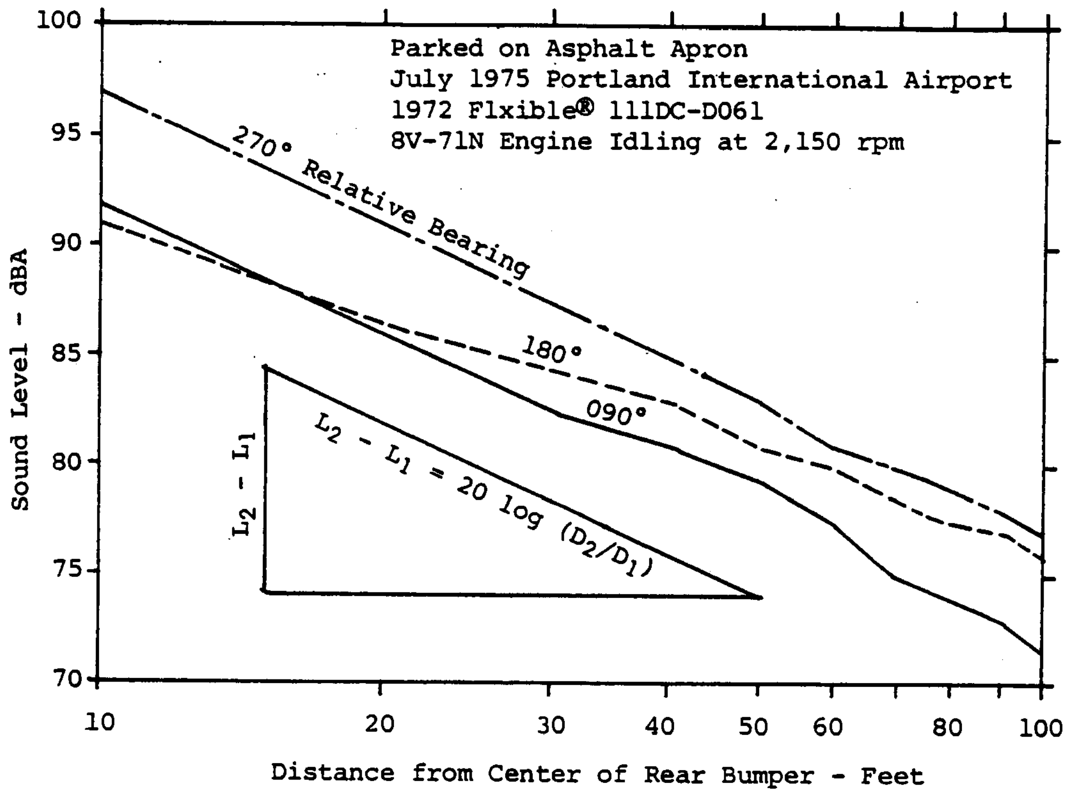


FIGURE 5.

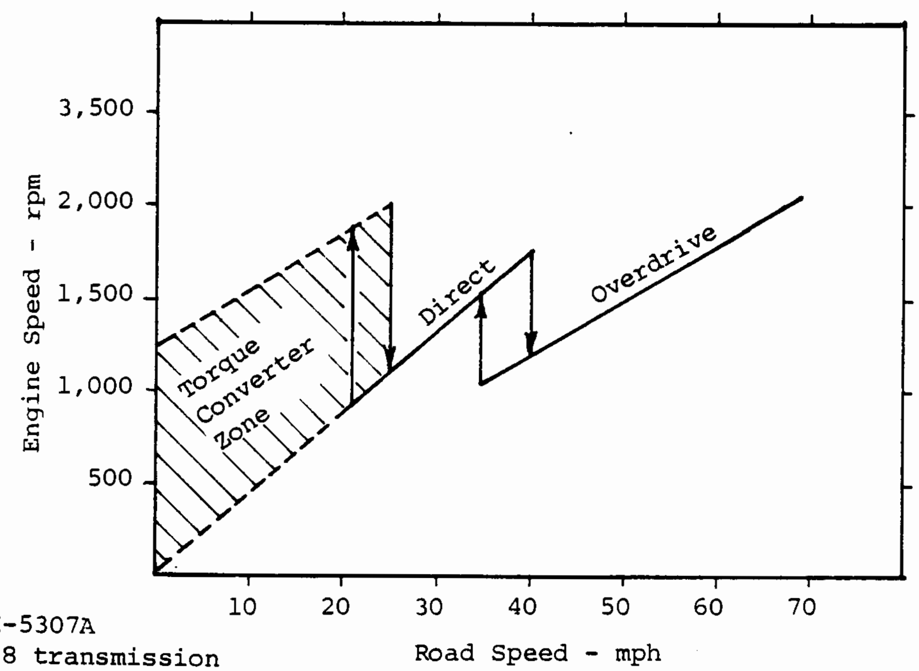
VARIATION OF BUS SOUND LEVEL
WITH DISTANCE

2.2 Noise Rating Procedure

The general strategy of a noise rating procedure is to operate the bus in a repeatable manner so as to produce its loudest noise. The SAE and EPA procedures place the rear of the bus just past, but close to, the microphone when the first upshift occurs so as to benefit from the strong rearward radiation pattern at the moment of peak sound. A detailed description of the test method is presented in Appendix A. The SAE procedure adopted by the Environmental Protection Agency when it formulated a proposed federal standard for new bus noise limits is given in Reference 1.

These procedures emphasize engine-related sounds and avoid the contribution of tire sound. At high speeds, tire sound can be dominant. Vehicle manufacturers cannot do much about tire sound, and in any event buses spend most of their time at no more than moderate road speeds.

Normally, bus sound becomes louder as the engine turns faster. Like its heavy truck counterpart, the transit motorbus diesel engine's overspeed governor is ordinarily set at about 2,100 rpm. However, unlike the truck engine, the bus engine rarely reaches this speed. As Figure 6 shows, the automatic transmission upshifts when the engine reaches about 1,750 rpm. Road speeds must exceed 60 mph before the engine can go over 2,000 rpm while the bus is moving. While parked, on the other hand, the engine is often idled at governed speed to hasten the time for air brake reservoir replenishment.



1973 GMC T8H-5307A
Allison VS2-8 transmission
"direct" 1.04:1
"overdrive" .72:1
Axle ratio 5.143
Tires 12.5 x 22.5, 490 rev/mile
Tested 5-28-80

[1] Reference 3.

FIGURE 6.

SPEED CHART OF A TYPICAL TRANSIT MOTORBUS

A noise rating test should reflect real-world bus operation. For instance, one might wonder what difference it would make if the passby noise rating were based on a "pullaway" test instead of a first upshift test since this is what it is like for a person near a bus stop as the bus gets underway. Figure 7 comes from a 1975 experiment to find the sensitivity of the peak sound level to the starting point (the distance uprun from the microphone's position, perpindicular to the rear bumper).

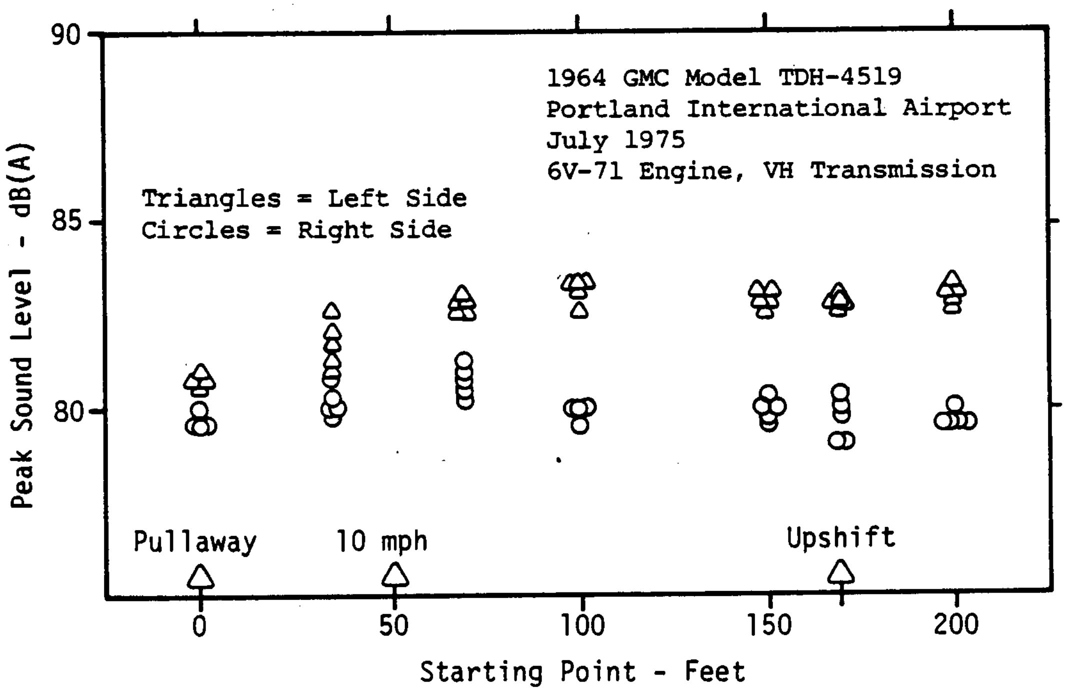


FIGURE 7.

SENSITIVITY OF PEAK SOUND LEVEL TO POSITION OF START

It doesn't seem to matter much where the starting point is set, except that noise ratings tend to fall off somewhat for starting points less than 50 feet. The experiment was repeated in August 1978 using a 1972 Flxible [R] Model 111DC-D061 with an 8V-71 engine and a VS2-8 transmission. Results confirmed the earlier finding.

If a bus has a fan clutch, the EPA rating procedure requires that it be artificially engaged fully when doing the test. This might stand to reason if on-off clutches were prevalent, but most buses either have modulated speed clutches or no fan clutch at all. Only under most unusual circumstances will modulated speed clutches cause the fan to turn at full speed. If this happens, the fan can be the dominant source of sound. Fan sound is steeply sensitive to fan speed. At lesser speeds, fan sound quickly subsides. Locking up a modulated speed fan clutch can mask the improvement made by a noise reduction treatment that would be discernible under ordinary circumstances.

Also consistent with "worst-case" thinking, the EPA procedure only considers the sound from the louder of the two sides of the bus for the exterior rating. The louder of the front or the rear seat is taken for the interior rating. The interior test is done while operating the bus in the same manner as for the exterior rating.

The noise from the right side of a bus is of more import because that is the side closest to boarders and other pedestrians. The value of a noise reduction treatment that accomplishes a lowering of right side sound, but not the left, would go unrecognized by a left-side-only rating.

A more realistic "operational" rating procedure would leave the fan drive in the "as-is" condition when doing the tests and average right and left side levels to obtain the exterior rating. The front and rear seat levels would be averaged to give the interior rating.

The levels proposed in the past by the EPA for a uniform, federal new-bus standard are:

	EXTERIOR	INTERIOR
First year	83 dBA	86 dBA
Fifth year	80 dBA	83 dBA
Seventh year	77 dBA	80 dBA

Experiments have shown that the interior noise levels of a bus are not appreciably affected by artificially locking up a fan drive.

The International Organization for Standardization (ISO Standard) microphone distance for noise-rating heavy vehicles is 7.5 meters, about 25 feet or half the standard distance used in the U.S. Tests have verified that it is valid to translate an ISO rating into a U.S. rating by simply subtracting the 6 dBA that the free-field distance rule would suggest.

Table 3 sets forth a collection of bus noise ratings taken over the span of this report. Some generalities are:

- One can expect repeat tests of the same bus usually to give the same ratings within 1 dBA. Tests made years apart might result in as much as a 2 dBA change, depending on the rating procedure.

- One can expect a group of in-service buses from the same batch of manufacture to fall into a 3 dBA range of noise ratings.

- Tests on smooth cement pavements can result in noise ratings 2 dBA higher than tests made on ordinary porous asphalt pavements.

- Buses with the 6V-71N engine do not always have higher noise ratings than those with 8V-71 engines.

- Newer buses are, if anything, slightly louder than older buses.

- Exhaust leaks can increase noise ratings by at least 4 dBA

- Turning the fan on can increase the noise rating of a bus by as much as 3 dBA.

- 4-stroke cycle engines tend to be louder than 2-stroke cycle engines.

- Turbocharged engines tend to be quieter than naturally aspirated engines.

- Bus noise ratings have little to do with their make or model.

- Lumping all the various buses of a typical fleet mix together would give the following noise ratings:

	EXTERIOR		INTERIOR	
	[1]EPA	[2]Opr	[3]EPA	[4]Opr
[5]Average	83½	80½	82½	79
Range	80-86½	78-86	76-89½	73½-82½

[1] Left side. Fan on.
[2] Average of left and right sides. Fan as-is.
[3] Rear seat. Fan on.
[4] Average of front and rear seats. Fan as-is.
[5] Rounded up to the nearest ½ dBA.

TABLE 3.

A COLLECTION OF BUS NOISE RATINGS
1975-1982

Bus	Year	Make	Model	Engine	Noise Rating[1]					Test Date
					Exterior			Interior		
					EPA					
					Fan On	As-Is[2]	Opr[3]	EPA	Opr[4]	
537	1964	GMC[5]	TDH-4519	6V-71N	86	86	83½	85	79½	Jun 1975
537	1964	GMC	TDH-4519	6V-71N		85	84	84½		Nov 1977
538	1964	GMC	TDH-4519	6V-71N		83½	81	84		Nov 1977
549	1965	GMC	TDH-4519	6V-71N		79	78½	83		Nov 1977
553	1965	GMC	TDH-4519	6V-71N		81½	79	85		Nov 1977
612	1971	Flx[6]	111DD-D51	6V-71N		81½	81	80½		Nov 1977
635	1971	Flx	111DD-D51	6V-71N	82	81½	81	84	79	Jun 1975
[9]635	1971	Flx	111DD-D51	6V-71N		85½	85	84		Nov 1977
304	1972	Flx	111DD-D61	8V-71N	80	79½	79	75	73½	Jun 1975
322	1972	Flx	111DD-D61	8V-71N		80½	80	82		Nov 1977
324	1972	Flx	111DD-D61	8V-71N	82			83	79½	Oct 1978
330	1972	Flx	111DD-D61	8V-71N		81	80	81		Nov 1977
330	1972	Flx	111DD-D61	8V-71N		79½	78½			Feb 1979
339	1972	Flx	111DD-D61	8V-71N		81	80	80½		Nov 1977
357	1972	Flx	111DD-D61	8V-71N		80	79½	83	80	Aug 1978
[10]357	1972	Flx	111DD-D61	8V-71N		82½	82	82	79½	Sep 1978
357	1972	Flx	111DD-D61	8V-71N		80				Oct 1978
357	1972	Flx	111DD-D61	8V-71N	80	79½	78½			Oct 1979
364	1972	Flx	111DD-D61	8V-71N		80½	79½	81		Nov 1979
370	1972	Flx	111DD-D61	8V-71N		80½	79½	84		Nov 1977
374	1972	Flx	111DD-D61	8V-71N		78½	78	80		Nov 1977
[11]802	1973	GMC	T8H-5307A	8V-71N	84	84	83			Jun 1980
803	1973	GMC	T8H-5307A	8V-71N	84½	83½	81			Jun 1980
805	1973	GMC	T8H-5307A	8V-71N	83	80	77½			Jun 1980
807	1973	GMC	T8H-5307A	8V-71N	83	80½	78			May 1980
104	1974	Flx	53102-8-1	8V-71N		81	78	81		Nov 1977
109	1974	Flx	53102-8-1	8V-71N		80½	77½	79		Nov 1977
115	1974	Flx	53102-8-1	8V-71N		78½	77½	79		Nov 1977
118	1974	Flx	53102-8-1	8V-71N	81½	80	79½	76		Jun 1975
118	1974	Flx	53102-8-1	8V-71N		81	77½	78		Nov 1977
1000	1976	AMG[7]	10240B-8	8V-71N		83½	83½	85½		Nov 1977
1007	1976	AMG	10240B-8	8V-71N	85	83	82½	82	78	Oct 1979
1021	1976	AMG	10240B-8	8V-71N		82½	81½	85		Nov 1977
[12]1027	1976	AMG	10240B-8	6V-92TAC	84	81	79½	80½	76	Oct 1979
1038	1976	AMG	10240B-8	8V-71N	86½	83	83	83		Nov 1977
1042	1976	AMG	10240B-8	8V-71N		82	81	87		Nov 1977
1053	1976	AMG	10240B-8	8V-71N		81½	81	85		Nov 1977
1060	1976	AMG	10240B-8	8V-71N	84½	83½	82½			Jun 1977
[13]1165	1976	AMG	10240B-8	VT-903		78	77½	85	77	Sep 1978
714	1981	C/I[8]	286.02	NHHTC-290		82	80½	89½	82½	Apr 1982

[1] Rounded up to the nearest ½ dBA.
[2] Same as EPA standard procedure except that fan clutch left in normal mode.
[3] Average of left and right sides with fan normal.
[4] Average of front and rear seats.
[5] General Motors.
[6] Flxible®.
[7] American General.
[8] Crown-Ikarus.
[9] Suspect exhaust leak.
[10] Tested on wet smooth cement. All other tests on dry asphalt concrete.
[11] Known to have exhaust leak.
[12] Engine conversion.
[13] A Seattle METRO bus. Engine conversion.

Standard procedure is to use a "fast" sound level meter response setting. One might wonder what difference it would make if the meter were set on "slow" instead. "Slow" makes the rating about 1 dBA lower. Such a comparison was made at the Portland International Raceway track in August, 1979. The test bus was #341, a 1972 Flxible [R] Model 111-D061 powered by an 8V-71T engine with 71C5 injectors. The fan drive was operating normally. Ambient temperature was 75°F. The bus was equipped with a partial engine compartment lining kit and a Donaldson 11180 muffler in tandem with a Donaldson "Super Stack". Results were:

	Noise Rating - dBA	
Meter Setting	Left Side	Right Side
Fast Slow	76½ 76	74 73
Difference	½	1

2.3 SOURCE CONTRIBUTIONS

A bus is a chorus of individual noises. These can be grouped as engine compartment noise, fan noise, exhaust outlet noise, muffler and exhaust pipe shell noise, and tire and body noise.

2.3.1 Engine Sound

As Table 4 indicates, the sound energy from a bus diesel engine is distributed over a broad spectrum of frequencies. The bulk of the energy is in the range of 500 to 4,000 Hz.

TABLE 4.

SPECTRAL DISTRIBUTION OF BUS ENGINE SOUND
4 Feet to Right Side of Engine
2,500 ft^3 Dyno Room, .36 sec Reverberation Time
Detroit Diesel 8V-71N, Model 7087-420
221 bhp, 2,150 rpm

Center Frequency	Sound Level
31.5	89
63	73
125	89
250	90
500	95
1,000	97
2,000	97
4,000	96
8,000	89
16,000	82
31,500	63
	102.3 dBA

Engine load consists of internal friction as well as external resistance.

$$THP = BHP + FHP$$
$$FHP = pN + qN^3$$

Where:
THP = Total engine horsepower
BHP = Brake horsepower
FHP = Friction horsepower
N = Engine speed - rpm
p = Coulomb factor
q = Viscous factor

For the 8V-71N engine, $p = .0205$ $q = 4.5 \times 10^{-9}$

For instance, friction horsepower for the 8V-71N engine at 2,000 rpm is 77 hp.

The sound from a bus engine increases with engine speed as Figure 8 shows.

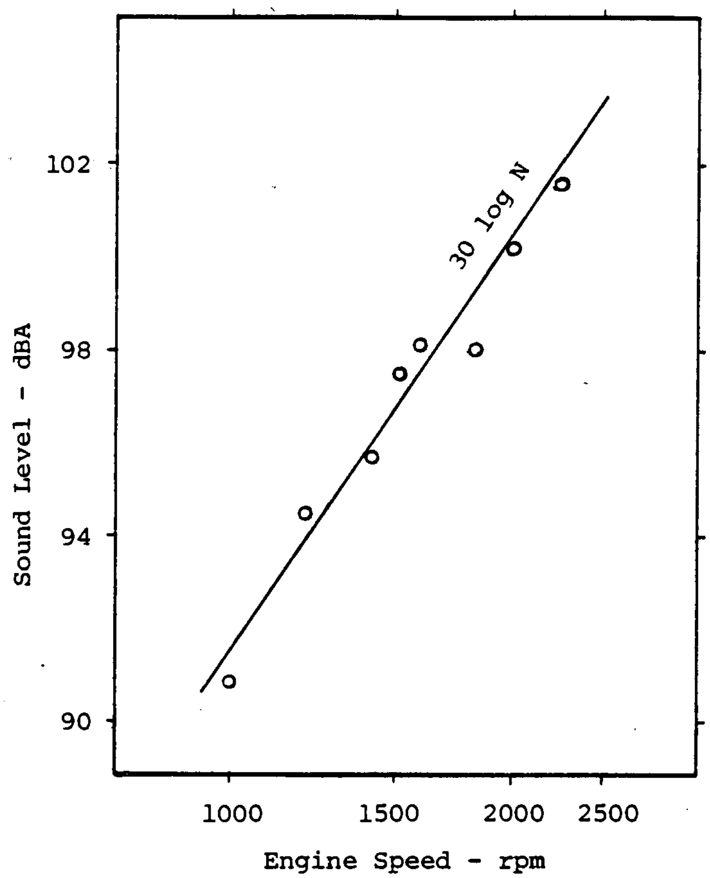


FIGURE 8.

SPEED SENSITIVITY OF 8V-71N ENGINE SOUND
4 Feet to Right Side of Engine
2,500 ft^3 Dyno Room, .18 sec Reverberation Time
Detroit Diesel 8V-71N, Model 7087-4020
Zero Dynamometer Load
Tri-Met, April 1982

A bus engine's sound also increases with load, as Figure 9 shows. But, its sound is far more sensitive to engine speed than to its engine load. Raising the engine's output from no throttle to full throttle at any speed results in raising its sound level by less than 3 dBA.

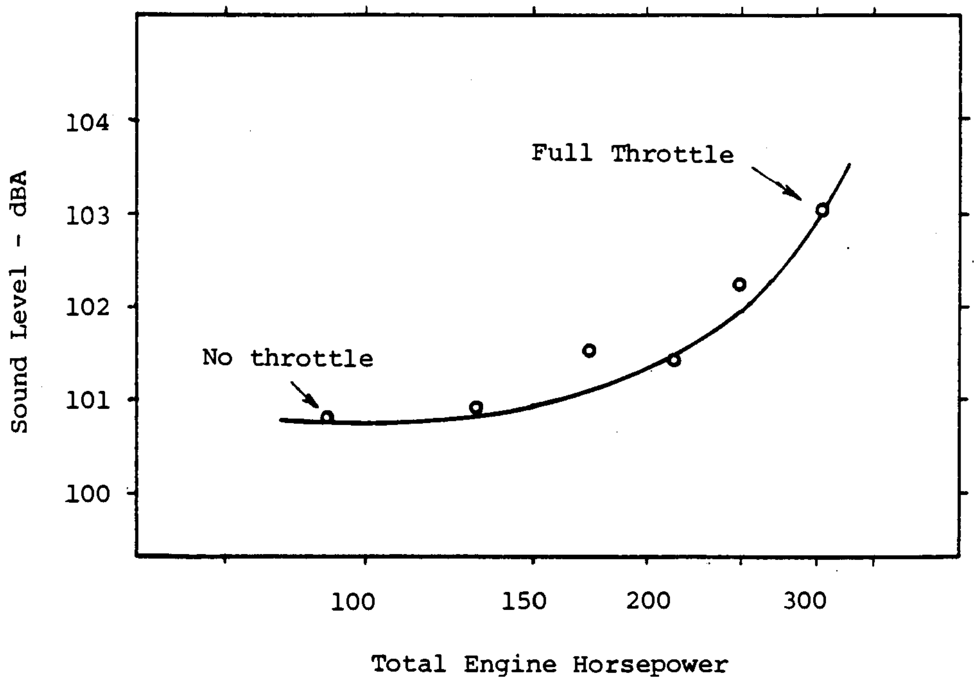


FIGURE 9.

LOAD SENSITIVITY OF 8V-71N ENGINE SOUND
4 Feet to Right Side of Engine
2,500 ft^3 Dyno Room, .18 sec Reverberation Time
Detroit Diesel 8V-71N, Model 7087-4020
Constant 2,150 rpm Engine Speed
Tri-Met, April 1982

Engine sound may be thought of as consisting of two sounds: one from a speed-sensitive source and the other from a load-sensitive source.

$$L_N = a \log N + C$$

$$L_{hp} = b \log THP + Q + C$$

Where:
- L_N = Sound level of the speed-sensitive source - dBA
- L_{hp} = Sound level of the load-sensitive source - dBA
- a & b = Sensitivity factors
- Q = A constant relating the two sources
- C = A constant characterizing the acoustic environment
- N = Engine speed - rpm
- THP = Total engine load - hp

For the 8V-71N engine:
- a = 30
- b = 37.5
- Q = 4.3

Tests in the field generally confirm that engine sound is quite sensitive to engine speed. As seen in Figure 10, the trend of data indicates a senitivity of 24 log N. In other words, doubling the engine's speed increases engine sound by about 7 dBA. Whatever is done to lower engine speed has a strong potential for reducing bus noise.

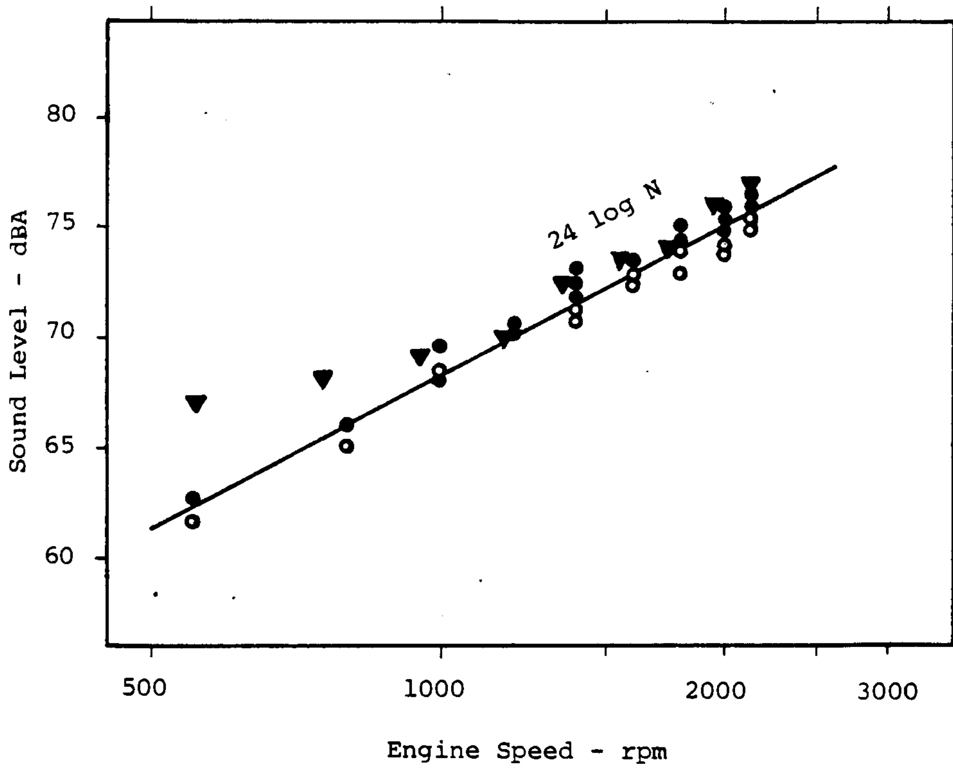


FIGURE 10.

INSTALLED ENGINE SOUND SPEED SENSITIVITY
Parked, No-Load Engine Revving
50-Foot Microphone Distance
Fan Not Turning

Circles: Open = Right Side; Closed = Left Side
Mix of Buses 118 (8V-71N), 304 (8V-71N), and 635 (6V-71N)
Rear of Bus on Microphone Perpendicular
Portland International Airport, July 1975

Triangles: Bus 324 (1972 Flxible® with 8V-71N) Left Side
Rear of Bus 30 Feet Past Microphone
Portland International Raceway Dragstrip, October 1978

During an engine's compression stroke, there is a certain rate of pressure increase within the cylinder. Upon the onset of combustion, this rate gives way to a more rapid rate. Fundamentally, the suddenness of the rate changeover is related to the loudness of the engine. Conventional wisdom holds that since turbocharging tends to smooth the changeover, it has a quieting effect. While laboratory tests seem to confirm this, the effect is modest and varies from case to case as indicated by Table 5.

TABLE 5.

[1]QUIETING EFFECT OF TURBOCHARGING
Dynamometer Room Tests in Accordance with SAE J1074
Sound Level dBA at 1 Meter

Engine	Right	Left	Front	Top
8V-71N	100.4	99.7	100.4	97.7
8V-71TA[2]	99.8	99.0	101.5	97.0
6V-92TA	98.5	99.0	99.0	95.0

OPERATING PARAMETERS OF TEST

Engine	RPM	BHP[3]	Injectors	Timing
8V-71N	2,100	304	N65	1.460
8V-71TA[2]	2,100	334	7A75	1.460
6V-92TA	2,100	335	9B90	1.470

[1] Data supplied courtesy of Detroit Diesel Allison Division of General Motors Corporation.

[2] TV-7101 turbocharger

[3] Adjustments to sound levels must be made for differences in loading before fair comparisons.

The front of the engine is situated to the left side of the bus where it looks out through the radiator opening. Adding to its importance, the front end tends to be as loud or louder than the other sides of the engine.

The turbocharged engines shown on Table 5 are operating at 10% higher output levels than is their naturally-aspirated companion. This results in about a 1½ dBA handicap. After making allowances for this, it appears that the TV-7101 turbocharger's quieting effect on front end sound was less than ½ dBA. However, the new generation turbocharged 6V-92TA engine should be almost 3 dBA quieter than its 8V-71N predecessor.

The relatively lower sound emission of the turbocharged 6V-92TA was demonstrated when one of Tri-Met's 1976 AMG buses was retrofitted with one of these engines and compared to a representative counterpart having its original 8V-71N. The AMG bus was chosen for this project because of its larger engine compartment. Source levels were identified by separately determining the exhaust and tire sounds and deducting them from the overall level (without the fan turning) to arrive at engine compartment sound. Table 6 shows that engine sound from the 6V-92TA powered bus was 3 dBA less on the left side and 5 dBA less on the right side. Both overall exterior and interior ratings reflected this improvement. Moreover, the 6V-92TA bus climbed a sustained 4% grade at 45 mph compared to a 42 mph baseline, indicating at least a 10% higher power output.

TABLE 6.

QUIETING EFFECT OF INSTALLED 6V-92TA ENGINE
Fan Not Turning
1976 AMG Model 10245B-8
Portland International Raceway Dragstrip
October 1979

		Baseline Bus 1007 8V-71N[1]	Turbocharged Bus 1027 6V-92TA[2]
Engine Compt.	Left	83	80
	Right	82	77
Exhaust	Left	68	74½
	Right	67	70
Overall Exterior	Left	83	81
	Right	82	77 3/4
Interior	Rear	82	80½
	Front	74	71

[1] 50 mm injectors
[2] 75 mm injectors

Figure 11 shows that a tradeoff can be made of performance for engine noise reduction. This was demonstrated during a series of tests made to restore an antinoise-treated bus to a driver-acceptance level of acceleration performance. The treatment package included a retrofitted 8V-71TAC engine and a more extensive absorptive lining of the engine compartment. Complaints had been received during a 10-month field trial. Baseline was a 200-foot full throttle acceleration time of 8.7 seconds. A 10% increase in injector size and various throttle delay settings[1] were tried. The goal was to get back to a time of 9½ seconds without jeopardizing noise reduction gains unduly.

[1] A device in the fuel system that prevents the puff of smoke that comes when the throttle is first depressed.

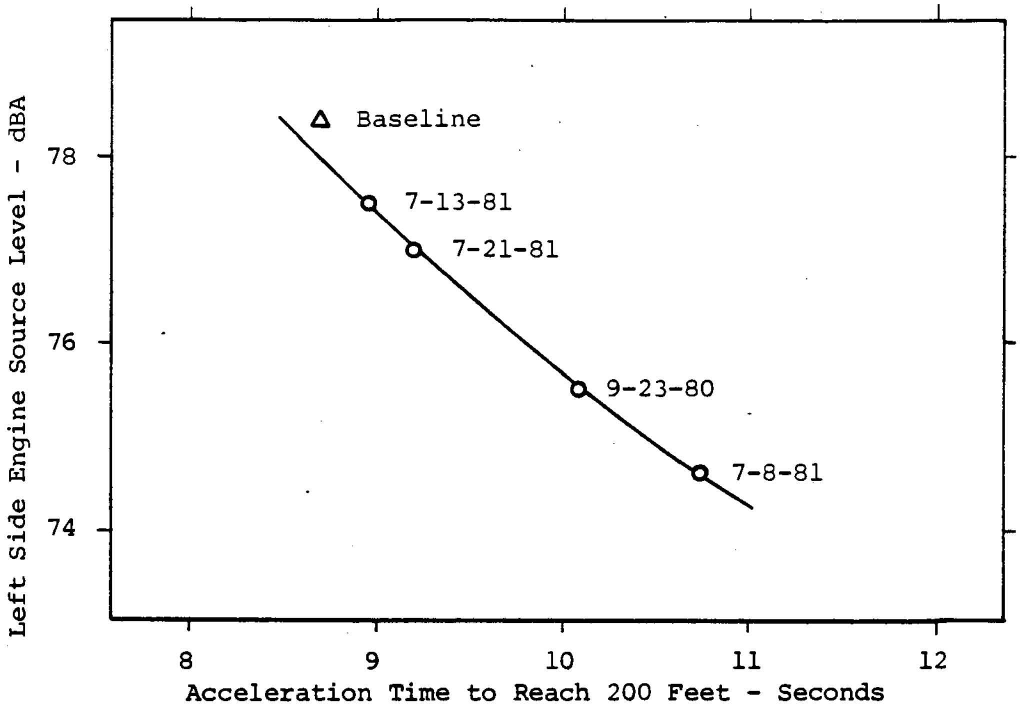


FIGURE 11.

TRADEOFF OF ACCELERATION PERFORMANCE
FOR ENGINE NOISE
Bus 189 (1973 GMC Model T8H-5307A)
Portland International Raceway Dragstrip

	Condition	Engine	Engine Compt. Lining
Triangle Circles	Baseline Antinoise Treatment	8V-71N 8V71-TAC	Original Equipment Insul-Quilt™

Figure 11 confirms the laboratory finding that only a small reduction in engine sound can be expected if the 8V-71N engine is changed to an 8V-71TA engine.

A Seattle METRO bus (1976 AMB Model 10240B-8) that had had a recent engine conversion afforded an opportunity to noise rate a Cummins turbocharged vee engine. Table 7 compares the salient features of this bus with Tri-Met's counterparts.

TABLE 7.

NOISE RATING OF CUMMINS VT-903 BUS

		Seattle METRO	Portland TRI-MET
Engine Features	Engine make & family	Cummins VT-903	Detroit Diesel 8V-71N
	Engine output rating	275 bhp @ 2,100 rpm	218 bhp @ 2,000 rpm
	Fuel	#2 diesel	#1 diesel
	Strokes per cycle	4	2
	Aspiration	Turbocharged	Natural with Roots blower
	Smoke avoidance	Aneroid control	Throttle delay
Bus Features	Transmission	Allison V730	Allison V730
	Fan drive	Facet automatic	Facet automatic
	Radiator shutter	Back mounted	None
	Bellypan	AMG optional	None
Test Results	[1]Acceleration Distance	236 feet	187 feet
	[2]EPA Noise Rating	78 dBA	82 dBA

[1] Distance from standing start to first upshift of automatic transmission during full throttle acceleration run on flat.

[2] Except that fan clutch was left "as-is" (off, in this case) instead of being artificially forced on.

Several reasons probably account for the significantly lower noise rating of the METRO bus.

1. Tradeoff of acceleration performance. The METRO bus had a 26% longer first upshift distance (same transmission, same bus body) despite the 25% higher engine output rating. The aneroid control[1] was probably set very "low" to avoid visible exhaust emissions, thus temporarily and severely derating the engine.

2. Turbocharged 4-stroke cycle engine vs. naturally-aspirated 2-stroke cycle engine.

3. Louvered sheet metal engine compartment bellypan.

[1] A device in the fuel delivery system that curtails fuel flow until there is sufficient airbox pressure to avoid exhaust smoke during acceleration maneuvers.

The faster an engine is allowed to go, the more power it can produce....and the more noise it will make. Automotive diesel engines are equipped with centrifugal-action overspeed governors. These prohibit fuel ingestion while the engine is above a preset and adjustable value. Inexpensive engine hardware, called "2-speed governors", will allow the driver to remotely switch the overspeed governor from one value to another. One position might be "Normal", providing for the usual 2,100 rpm setting that is the optimal compromise between maximum available engine output and engine life and maintenance cost. The second position might be "Quiet", recognizing that the governor could be temporarily downset while operating in noise-sensitive parts of the city, such as a transit mall. If an across-the-board tradeoff of acceleration performance for a measure of noise reduction is unacceptable, perhaps a controllable mechanized tradeoff during selected portions of a bus route would be acceptable.

Pursuing this concept, tests were performed to determine the correlation of governor setting, noise rating, and performance. Buses utilized are described by Table 8.

TABLE 8.

BUSES USED FOR 2-SPEED GOVERNOR TESTS

	Bus 537	Bus 635	Bus 304
Year built	1964	1971	1972
Make	GMC	Flxible®	Flxible®
Model	TDH-4519	111DD-D51	111DC-D061
Engine family	6V-71N	6V-71N	8V-71N
Transmission	VH	VH-9	VS2-8
Stall speed	1,550 rpm	1,550 rpm	1,250 rpm
[1]Upshift speed	2,100 rpm	2,100 rpm	1,750 rpm

[1] These transmissions are shifted by a signal generated by road speed. The coincidental engine speeds listed occur during full throttle acceleration on a flat.

Settings of 2,150, 1,900, 1,750, and 1,600 rpm were tried. Downsets below 1,600 rpm were not possible without governor redesign.

Figure 12 displays promising results when the bus is measured by the EPA upshift method. Reductions up to 5 dBA can result from a 1 second delay in 100-foot acceleration time. However, it is more realistic in this case to go by a "pullaway" noise rating. Figure 13 shows that downsets below 1,600 rpm would be necessary if pullaway noise ratings are to be improved by the 2-speed governor technique. This is not surprising when one realizes that stall speeds are below 1,600 rpm. The engine does not enter the 1,600+ rpm zone while the bus is being rated by the pullaway method, and therefore is not affected by available downsets.

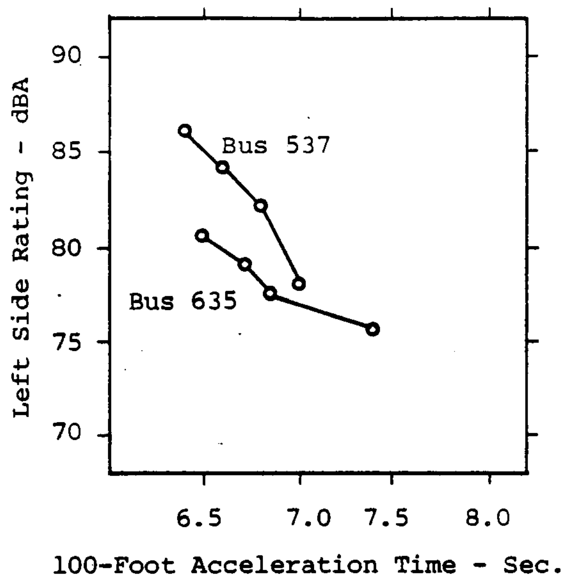


FIGURE 12.

EFFECT OF OVERSPEED GOVERNOR DOWNSETTING
ON EPA UPSHIFT NOISE RATING
Portland International Airport, July 1975

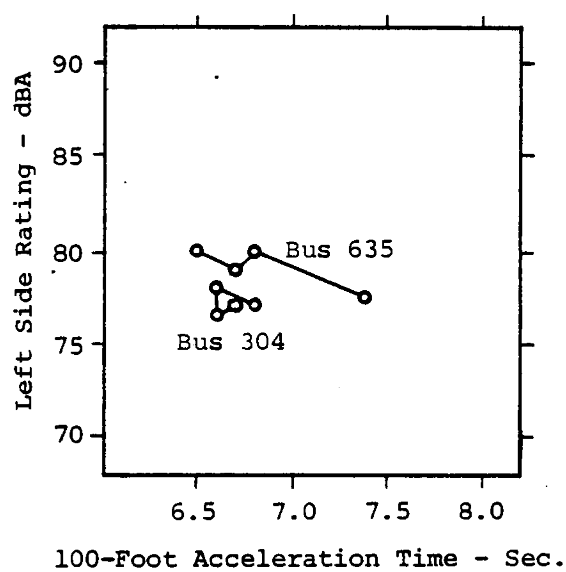


FIGURE 13.

EFFECT OF OVERSPEED GOVERNOR DOWNSETTING
ON PULLAWAY NOISE RATING
Portland International Airport, July 1975

Drivers can control bus noise to some degree simply by a judicious use of the throttle while accelerating. To determine what potential lies in the direction of driver training, a series of noise rating tests were made with a typical bus where the driver was asked to use various degrees of part throttle, a technique that is wholly discretionary...not subject to mechanical devices. Acceleration times and sound levels were measured as the bus passed the 100-foot mark, simulating what a downtown Portland pedestrian is exposed to standing on a street corner as a bus goes past from a midblock bus stop. A senior driver supervisor judged what constituted acceptable acceleration performance. Results shown on Figure 14 indicate that passby noise can be reduced by about 3 dBA from maximum if drivers would take it easy on the throttle pedal.

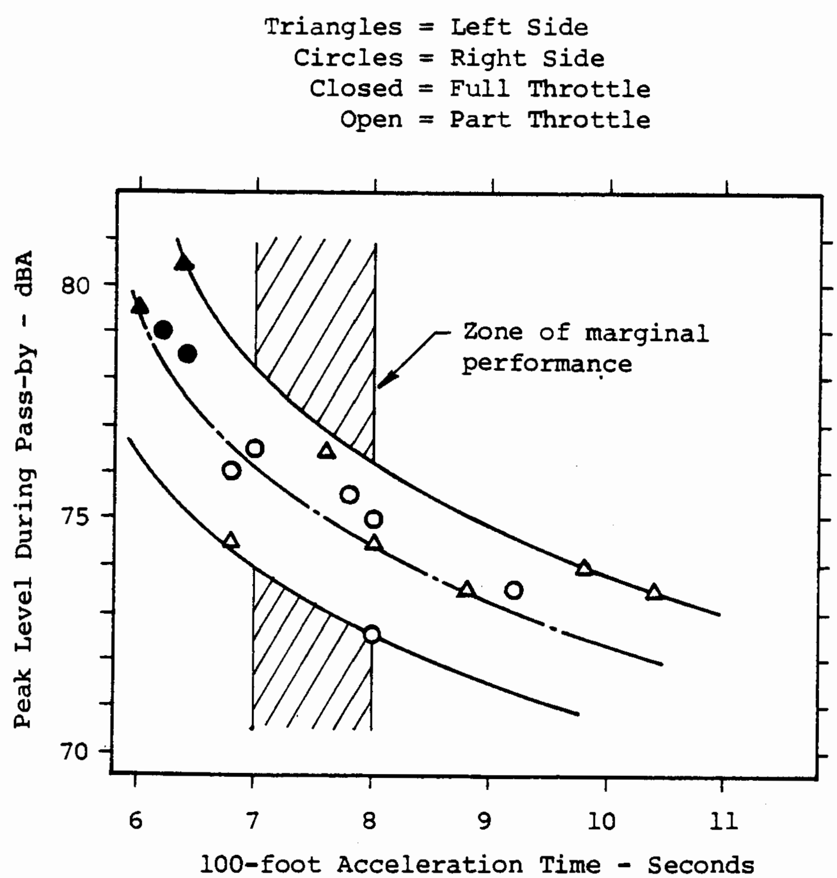


FIGURE 14.

NOISE CONTROL ACCOMPLISHED BY THROTTLE MODULATION

Bus 304
Fan Drive Normal
50-foot Microphone Distance
Portland International Airport, July 1975

An experiment was conducted with a mechanical device that would approximate what the driver does when he eases up on the throttle to reduce noise. This contrivance features a plunger that causes interference with the engine's throttle linkage when a switch is voluntarily thrown by the driver. The action is the same as if a stop were to be placed under the throttle pedal. A 2 dBA noise reduction did indeed result when this throttle stop was tested, but at the expense of a 30% increase in first upshift acceleration distance as Table 9 shows.

TABLE 9.

THROTTLE STOP EXPERIMENT
Bus 330
1972 Flxible® Model 111DC-D061
8V-71N Engine with 50 mm Injectors
Fan Drive Normal
Portland International Raceway, April 1980

	*Acceleration Distance - ft	Noise Rating - dBA	
		Left	Right
Baseline	150.1	79½	77½
Throttle Stop	197.7	78	75½

*Distance to the first upshift which occurs at about 24 mph.

Sometimes one encounters the notion that a "tune up" will lessen a bus's noise. It might sound "better", but would it be quieter? A bus was found that needed a tune up badly. It was noise rated before and after. The tune up consisted of adjusting the running clearances for the fuel injectors and the exhaust valves. Injector rack position was adjusted. No discernible change in noise rating accompanied these adjustments.

Detroit Diesel's standard valve cover prior to around 1976 was a deep rectangular shape made of pressed thin-walled steel. It was sealed by an ordinary gasket, but it tended to leak oil because of occasional overzealous bolt tightening. A stiffer cast aluminum valve cover replaced the older design and solved the problem. This valve cover is vibration-isolated by means of a thick rubber gasket and rubber hold-down bolt grommets. Tri-Met tested its reputation for noise reduction and found a consistent, but barely discernible effect, insufficient to make a difference to overall noise rating as shown by Table 10.

TABLE 10.

EFFECT OF ALUMINUM VALVE COVERS
Bus 357
1972 Flxible® Model 111DC-D061
8V-71N Engine with 50 mm Injectors
Rear Door Open
Steady 2,150 rpm High Idle
Bus Parked on Smooth Wet Cement Concrete
Microphone on Centerline to Rear of Bus
Fan Drive Normal
S.E. Powell Substation, September 1978

	Sound Level - dBA	
Distance -Feet	Steel Valve Covers	Aluminum Valve Covers
12½	95½	95
25	89½	89
50	84½	84

Even though bellypans (removable engine compartment floors) are known to reduce bus noise, Tri-Met's maintenance personnel are reticent to install them because of their susceptibility to damage, their interference with engine access, and their partial obstruction to cooling system air flow. As a compromise, an experiment was carried out with a heavy rubber skirt that hung down from the engine compartment's sides and rear edge to within a few inches of the ground. Table 11 shows there was no change to the bus's noise rating.

TABLE 11.

RUBBER SKIRT EXPERIMENT
Bus 357
1972 Flxible® Model 111DC-D061
8V-71N Engine with 50 mm Injectors
Fan Drive Normal
Portland International Raceway, October 1978

	Noise Rating - dBA	
	Left	Right
Without Skirt	79	77
With Skirt	79	77

An acoustically-effective bellypan was simulated by temporarily suspending a sheet of Insul-Quilt™ [1] beneath the engine compartment on a bus that had already received a Tri-Met antinoise treatment.[2] The only egress for the cooling system air flow was out through the hole in the firewall for the main drive line and through the numerous small holes and leakages. Thus, means for providing adequate cooling system performance would have to be found before such a bellypan could be made practical.

Results are given in Table 12. A very low exterior noise rating of 70½ dBA was achieved for the right side with the fan drive operating normally. On the other hand, the left side with the fan turning at maximum speed was not materially helped. No change at all to interior noise ratings resulted.

TABLE 12.

BELLYPAN EXPERIMENT
Bus 341
1972 Flxible® Model 111DC-D061
8V-71N Engine with 50 mm Injectors
TV-7101 Turbocharger
[3]Insul-Quilt™ Engine Compartment Lining
Donaldson 10 x 15 Oval Muffler, 5" x 5" Pipes
Portland International Raceway, April 1979

		Noise Ratings - dBA			
		Exterior		Interior	
		Left	Right	Rear	Front
Fan Normal	Without Bellypan	76	73	80	73
	With Bellypan	74½	70½	80	73
	Improvement	1½	2½	0	0
Fan On	Without Bellypan	79½	76	80	73
	With Bellypan	79	72½	80	73
	Improvement	½	3½	0	0

1 A blanket consisting of a 1½" layer of dense glass fiber batting on either side of a 10 oz/ft^2 lead septum sandwiched between acoustically porous woven nylon covers. NRC = .70 STC = 28

2 The acoustic lining for the rear door consisted of a continuous sheet of Insul-Quilt™. Field testing later discovered several points of chafing against the engine, resulting in holes and tears. The final version was made of patches recessed into the door cavities. They added up to only 60% of the original door lining. This may be the reason why the noise ratings on Table 12 are slightly better than those for the service-proven kit.

3 Testing of this material does not imply an endorsement. Similar results can be expected with equivalent products.

The engine is the dominant source of interior sound at road speeds of 25 mph and below. Levels at the back of the typical bus (where the engine is) are usually about 5 to 7 dBA higher than in the front, depending on how well the rear seat engine-access hatch is fitted. Turning the fan on and off seems to make little or no difference to interior noise...neither does changing exhaust mufflers.

The question is: how much of the engine sound entering the bus interior is structureborne and how much is airborne? To shed light on this question, an experiment with engine mounts was carried out.

The engine-transmission mass is resiliently mounted on a cradle. This assembly is attached to the bus body at a pair of front mounts and a pair of rear hangers, each pair taking about half the weight of the assembly. The front mounts consist of rubber doughnuts taking the vertical and horizontal loads. Being single-stage and relatively stiff, they probably transmit a high percentage of the impinging vibration. The rear hangers are simply metal straps in tension, having hardly any resilience at all. They probably transmit all impinging vibration.

The experiment was to prop the rear of the engine-transmission cradle with a pair of wooden blocks resting on the ground. The rear hangers were detached. *If* the engine were the dominant source of sound around the rear seat, and *if* all of it were structureborne, and *if* half of the sound energy were entering the bus body at the front cradle mounts, then disconnecting the rear mounts should reduce the interior sound by 3 dBA. If the front mounts were not transmitting all their input vibration, the reduction of sound inside the bus might be 1 or 2 dBA better than the 50-50 3 dBA.

Figure 15 displays how much the interior sound was decreased by disconnecting the rear hangers while the bus was parked with the engine revving. The decrease was about 3½ dBA at all but the highest engine speeds. Table 13 compares these idling condition interior rear seat levels with the EPA interior rating for this bus at the same engine speed. The EPA rating is louder than the idling condition because of the addition of power-sensitive engine noise (calculates to be ≈ 2 dBA) and, possibly, to other contributive sources (like the fan). The increase turns out to be 3 dBA.

The conclusion one must reach is that practically all sound penetrating the rear of the bus is structureborne engine noise. Prospects for dramatic interior noise reduction through improved engine mount design are excellent. One can expect to gain at least 6 dBA.

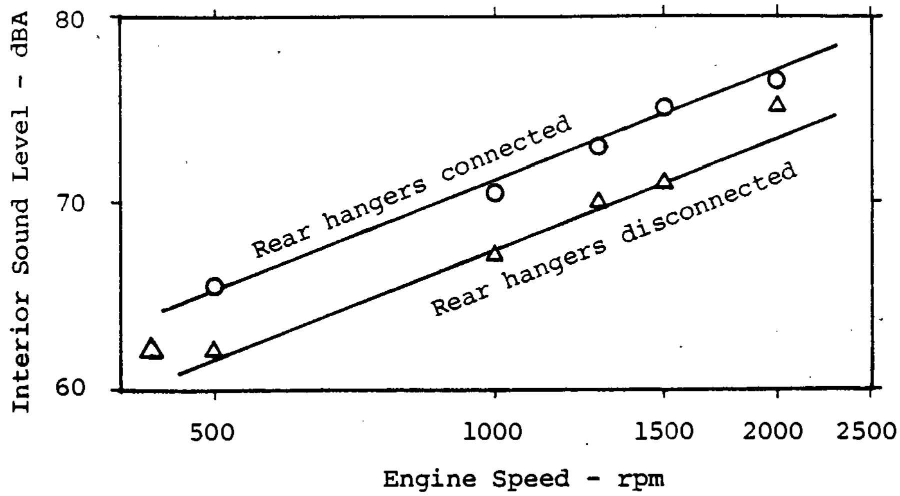


FIGURE 15.

INTERIOR SOUND AT REAR SEAT

Bus 807
1973 GMC Model T8-H5307A
Parked and Idling
Fan Off
Portland International Raceway, May 1980

TABLE 13.

ENGINE MOUNT EXPERIMENT
Bus 807
1973 GMC Model T8-H5307A
Baseline Condition
8V-71N Engine with 50 mm Injectors

		Interior Sound Level at Rear Seat - dBA
Noise Rating, Fan Normal		78
No-load Engine Idling at 1,750 rpm Fan off	Rear Hangers Connected	76
	Rear Hangers Disconnected	72½

2.3.2 Fan Sound

The typical transit motorbus cooling system fan has 6 or 8 equally spaced rectangular sheet metal blades riveted to a sheet metal hub. Its outside diameter is 26 to 30 inches. In the trade, it is called a "blower", even though it *draws* air in through the radiator. This is because transit motorbuses drive the rear axle from behind rather than from the front. The engine therefore turns clockwise as viewed from the rear.

Mounted on the engine's front, the typical bus fan is driven through a hydrodynamic coupling that modulates the fan's speed in such a way as to keep the radiator bottom tank in a preset temperature range. At maximum clutching, the fan's speed is 95% of the engine's speed.

Many refinements to the simple, ordinary sheet metal fan for heavy duty trucks and buses have been offered over the years, but none have enjoyed more than lukewarm acceptance. Fiberglas has been tried. Staggered, flared blades have been tried. Shrouded tips have been tried. Cast air foil cross-sections have been tried. The market keeps coming back to the conventional sheet metal fan. Its price is low, and it works surprisingly well. But, it can be noisy...potentially as loud as the engine.

Fan sound covers a broad range of frequencies. A major contribution stems from the blade passage rate and from the second and third multiples of this frequency. A lesser contribution is from a band of higher frequency sound associated with turbulence. Fan sound level appears to depend on tip speed more than any other parameter.

Fan sound is very sensitive to fan speed as Figure 16 shows. Fan-off levels were logarithmically subtracted from fan-on levels to determine fan sound with a typical bus parked where first upshift (and peak noise) occurs during a noise rating test. Cutting fan speed in half reduces fan sound by 15 dBA.

At speeds above 1,300 rpm, the speed sensitivity is very close to the classic 60 log N. That is,

$$L_{fan} = C + 60 \log D + 60 \log N + 20 \log S$$

where:

C = a constant for fan sound propagation
D = fan diameter
N = fan speed
S = distance to fan

Several other fan sound tests were made during the course of the program using other buses and testing various engine compartment lining treatments. Always the fan's speed sensivity was found to be near 60 log N.

With a modulating speed fan drive and a reasonably clean radiator core, fan noise is not a problem. Consider, for example, the data from Table 14. With the automatic cooling system functioning normally, the fan speed for a typical motorbus climbing a sustained grade at nearly full throttle was only 700 rpm. That was all the fan had to do to keep the engine at the proper temperature on that day.

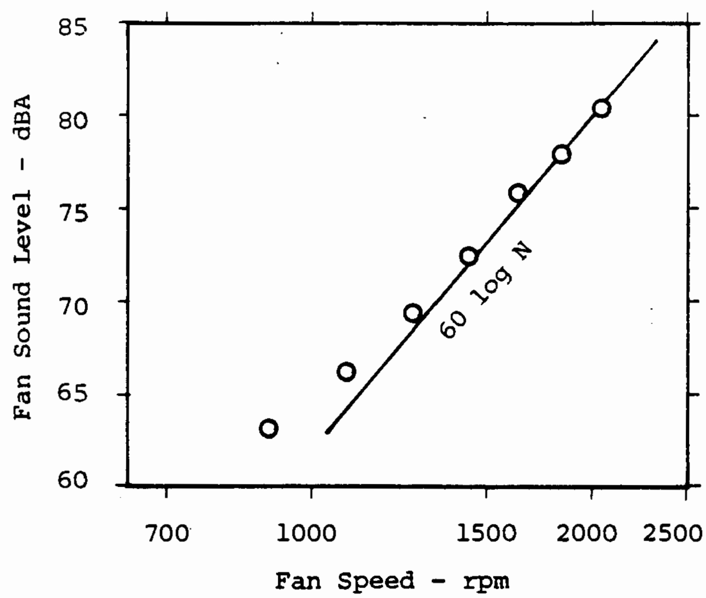


FIGURE 16.

SPEED SENSITIVITY OF FAN SOUND

Bus 324
1972 Flxible® Model 111D-D061
Schwitzer L71-911024 28" 8-blade Fan
Parked with Rear of Bus 30' Beyond Microphone
Portland International Raceway, October 1978

AT 700 rpm, the fan's contribution to the bus's left side noise rating would be about 60 dBA...completely masked out by the other engine compartment sounds. Granted, there are other times on hotter days when the fan must furn faster and make more noise. But, conditions must approach the severest for fan noise to become a significant problem. For instance, if the conditions of Table 14 were repeated, but with the ambient air temperature at 100 °F and 50% relative humidity, the left side fan source contribution would rise to about 68 dBA, still not enough to be of consequence.

Consider the effect of either not having a modulated speed fan drive or of having one rendered inoperative. During Table 14 conditions, the left side fan source contribution would become 75 dBA...definitely contributive.

TABLE 14

TYPICAL COOLING SYSTEM PERFORMANCE
Bus 357
1972 Flxible® Model 111D-1061
8V-71N Engine with 71C5 Injectors, #1 Diesel
4% Grade for 3 Miles
Normally Dirty Radiator Core
GMC Fan Drive
Canyon Road, November 1978

Road Speed for Test	40 mph
Full Throttle Road Speed	45 mph
Engine Speed	1,700 rpm
Fan Speed	700 rpm
Ambient Air Temperature	60 °F
Relative Humidity	72 %
Top Tank Temperature	180 °F

The surest way to reduce fan noise is to use a larger diameter fan and a larger radiator core to go with it. Unless there is to be major alteration, this must be done by the bus manufacturer. Prospects for noise control with this approach are very good. Suppose, for example, that the bus in Table 12 were climbing the same grade on a 100 °F, 50% relative humidity day with a dirtier radiator core causing the fan to turn at its maximum speed (1,615 rpm). The fan's left side contribution would be 75 dBA. Keeping everything constant, let the fan's diameter increase by 20% to 34 inches. Let the radiator core's frontal area also increase by the same proportion. Since the core has more area, its rows of tubes can be reduced from 4 to 3, keeping its cost the same and making it more efficient and less sensitive to dirt accumulation. Analysis indicates that the fan's tip speed would reduce to 6,000 fpm from 12,000 fpm, resulting in a sound reduction of 10 to 15 dBA.

As a noise control experiment, seven 4-inch wide, ½-inch thick fabric-covered fiberboard louvers were placed between the radiator grille and the radiator core in such a way as to cut off line-of-sight sound radiating outward through the core. No discernible noise reduction resulted. There was a significant reduction in air flow. This approach was abandoned.

2.3.3 Exhaust Sound

On our typical transit motorbus, the engine's exhaust is carried by a vertical 4-inch pipe to the upper left rear corner of the body where it is directed upward from an almost concealed terminus. Tests have revealed that the sound from this outlet radiates as if from an omnidirectional point source.

An exhaust experiment was conducted using a typical bus. The terminus was extended by clamping on an additional 4-foot vertical stack. To this was attached a horizontal arm carrying a microphone. The arm was swung at 45° to the bus centerline, placing the microphone over the bus roof where it was shielded from the engine compartment sound. The bus was operated just as when making a standard noise rating test. Readings were taken at the moment of first upshift. Wind noise was determined by coasting at test speed (86.5 dBA at 24 mph). Raw data was corrected for background sound (wind plus an estimated engine sound based on a source level of 79 dBA at 56 feet and 4 dBA shielding). Figure 17 shows that as the microphone was moved from ½ foot to 3½ feet away from the terminus center, the exhaust sound level decreased by the classic 20 log D distance rule. This indicates spherical divergence. Exhaust source levels can be measured at a distance of one foot without concern for error due to either "near field" effect or background sound. (If the microphone is placed too close to a source, sound level does not vary with distance in a regular way. This is called the "near field" effect.)

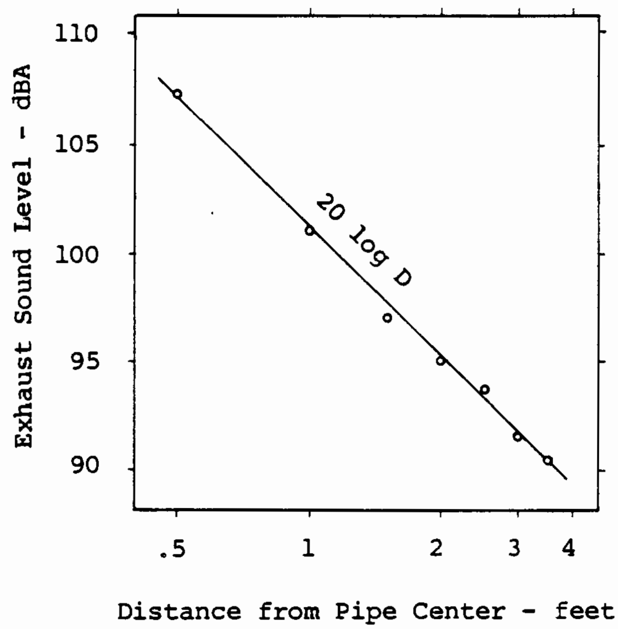


FIGURE 17.

DIVERGENCE OF EXHAUST SOUND

Bus 324
1972 Flxible® Model 111D-D061
8V-71N Engine with 71C5 Injectors
Nelson T-12023-F Exhaust Muffler
Portland International Raceway, October 1978

The same test was repeated, but with the microphone set at a constant distance of one foot and with the arm swung in 8 even arcs around a circle. Figure 18 shows the exhaust sound level to be uniform in all directions.

Sound Levels at 1 Foot from the Exhaust Terminus

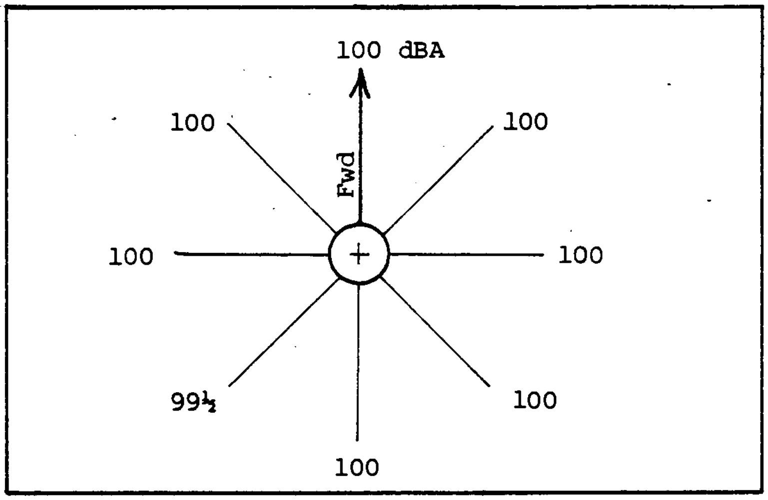


FIGURE 18.

DIRECTIVITY OF EXHAUST SOUND

Bus 324
1972 Flxible® Model 111D-D061
8V-71N Engine with 71C5 Injectors
Nelson T-12023-F Exhaust Muffler
Portland International Raceway, October 1978

Exhaust sound is sensitive to both speed and load. A trend of increasing exhaust sound level as the engine is revved up under parked-and-idle conditions can be seen in Figure 19.

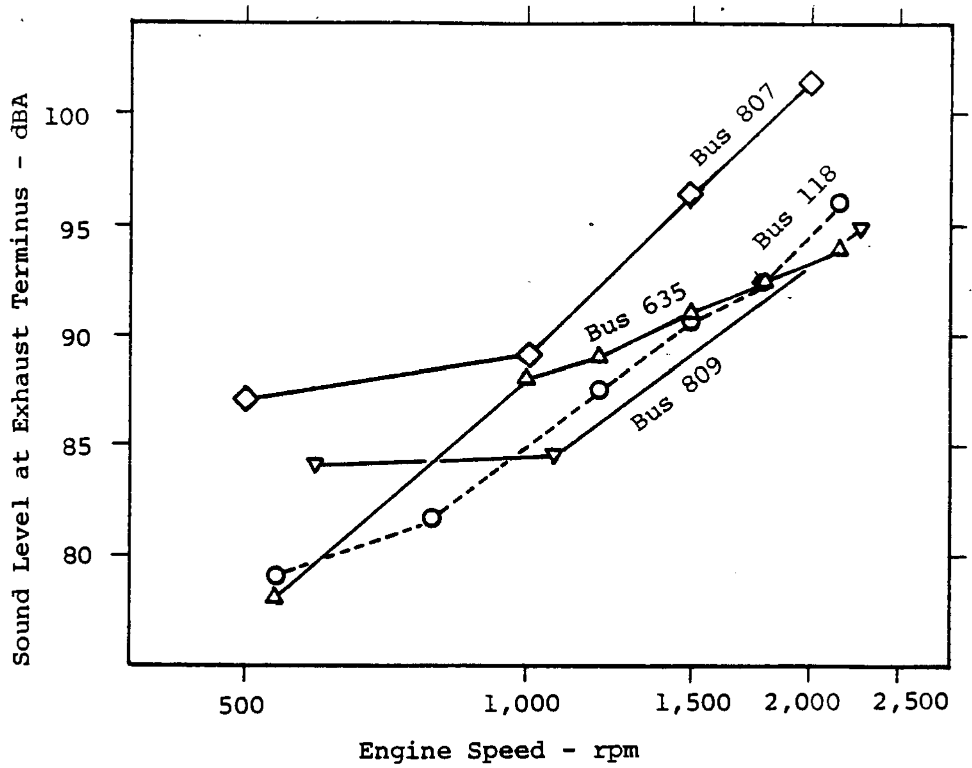


FIGURE 19.

EXHAUST SOUND AT NO-LOAD IDLE

Bus 635: 1972 Flxible® with 6V-71N Engine
Bus 807: 1973 GMC with 8V-71N Engine
Bus 809: 1973 GMC with 8V-71TAC Engine
Bus 118: 1974 Flxible® with 8V-71N Engine

1-Foot Microphone Distance
Parked and Idling

Due to engine friction, engine load increases somewhat as speed increases during a rev up test. The effect of additional engine loading can be further explored by stalling the engine at full throttle against the torque converter with the bus in gear and with the brakes set. A typical set of data is presented by Table 15.

TABLE 15.

LOAD AND SPEED SENSIVITY OF EXHAUST SOUND
Bus 807
1973 GMC Model T8-H5307A
8V-71N Engine with 71C5 Injectors, #1 Diesel
Orginal Equipment Muffler
1-Foot Microphone Distance
Portland International Raceway, May 1980

Engine Loading	dBA	RPM
No-load Idle	89	1,000
No-load Idle	96½	1,500
No-load Idle	101½	2,000
Full Throttle	103½	1,250
Full Throttle	106½	1,750

Analysis of this and of other tests on the 8V-71N engine gives the following equation for exhaust sound. Its accuracy has been ± 1 dBA at speeds above 1,000 rpm.

$$L = C + 18.5 \log THP + 13.6 \log N + 20 \log D$$

recall:

$THP = BHP + FHP$
$FHP = .0205\ N + (4.5 \times 10^{-9})N^3$...for the 8V-71N engine

where:

L = Exhaust source level.
C = A constant characterizing a certain engine and exhaust system.
THP = Total horsepower.
BHP = Brake horsepower.
FHP = Friction horsepower.
N = Engine speed - rpm.
D = Distance from exhaust terminus center - feet.

During the course of the program at Tri-Met, a number of exhaust source levels were measured. A summary is given on Table 16.

TABLE 16.

EXHAUST SOURCE LEVELS
Full Throttle Engine Loading
1-Foot Microphone Distance

Date of Test	Bus	Engine	Muffler	Make	RPM	dBA	[1]"Hg
Jun 1975	118	[2]8V-71N	[3]OEM	1974 Flx	1,825	99	
Jun 1975	304	8V-71N	Nelson T-120323-F	1972 Flx	1,750	96	?
Jun 1975	635	[2]6V-71N	OEM	1971 Flx	2,100	106	?
Oct 1978	324	8V-71N	Nelson T-120323-F	1972 Flx	1,800	101	
Oct 1978	357	8V-71N	Nelson T-120323-F	1972 Flx	1,740	101½	5.1
Mar 1979	341	8V-71T	[4]Donaldson 11200	1972 Flx	1,800	107	1.7
Mar 1979	341	8V-71T	Donaldson 11200 [5]+Donaldson 16025	1972 Flx	1,800	104½	?
Aug 1979	341	8V-71T	[6]Donaldson 11180	1972 Flx	1,800	100½	4.6
Aug 1979	341	8V-71T	Donaldson 11180 [7]+Donaldson K8 567410410	1972 Flx	1,800	99	4.9
Oct 1979	1007	8V-71T	OEM	1976 AMG	1,850	103	?
Oct 1979	1027	6V-92TAC	Donaldson 11200	1976 AMG	1,850	109½	?
Oct 1979	341	8V-71T	[8]Donaldson 5080B79	1972 Flx	1,800	100	2.1
Oct 1979	341	8V-71T	Donaldson 5080B79 +Donaldson 16025	1972 Flx	1,800	98	2.1
May 1980	807	8V-71N	OEM	1973 GMC	1,750	106½	6.1
Aug 1980	809	8V-71TAC	Donaldson 5080B79	1973 GMC	1,750	99	3.5

[1] See Section on performance benchmarks for a disussion on back pressure.
[2] with 71C5 injectors and #1 diesel.
[3] Original equipment.
[4] 10" x 15" oval x 26" long...5" pipes in and out.
[5] "Super Stack" for 5" pipe.
[6] 10" x 15" oval x 26" long...4" pipes in and out.
[7] "Super Stack" for 4" pipe.
[8] 10" x 15" oval x 26" long...4" pipe in, 5" pipe out.

The compact 10" x 15" oval Donaldson muffler did a less than desirable job of exhaust sound attenuation for the turbocharged Detroit Diesel engines when in the 5" pipe configuration. It was more than adequate when the pipe size was reduced to 4" diameter, but the back pressure exceeded the 3 "Hg limit imposed by the engine manufacturer. A satisfactory compromise was found with the 4" pipe in and 5" pipe out configuration without the need for a supplemental muffler such as the Donaldson "Super Stack".

2.3.4 Tire Sound

At urban traffic speeds, tire sound from a typical transit motorbus is a minor source. An average source level at 25 mph would be 65 dBA...at 35 mph: 68½ dBA. At highway speeds, tire sound can exceed 75 dBA and become significant.

The typical transit bus has 6 tires, 2 on the front wheels and 4 on the dual-tired rear wheels. Rib-type tread design is quieter than the more traction-aggressive cross-lug design. In the interest of cost savings through uniformity, transit motorbuses commonly use rib-type tread at all wheel positions. It is the intercity highway truck-tractor that needs the cross-lug tires on its driving wheels. Radial tires with rib-type treads can be slightly quieter than their bias-ply counterparts. Bias-ply tires are more popular for transit buses because of their sturdier sidewalls ...less susceptible to curb damage.

The tire source level is established by coasting the bus past the microphone at various speeds with the engine off and the transmission in neutral. The peak sound level is noted. Figure 20 shows typical test results. Speed sensivity is 27 log MPH.

Lumped with tire sound is aerodynamic sound and gear sounds from the rear axle and transmission. Gear whine tends to be variable from bus to bus, and, because of backside tooth loading, is probably quieter during passbys under power.

The equation for the best-fit linear curve from Figure 20 is:

$$L_T \text{ (peak)} = 27.5 + 26.6 \log \text{MPH}$$

L_T (peak) is the total sound made by the bus as it coasts by. According to analysis, the center of the bus is very close to the microphone's perpendicular when peak sound occurs. We shall want to know what the tire source contribution is to the overall noise rating. The overall peak occurs when the rear of the bus is 30 feet past the microphone where the received tire sound has fallen off somewhat. A close approximation is to assume that all tire sound is coming from the center of the bus and obeys the 20 log D distance rule. For instance, the center of a 35' 8" bus is 47.8 feet past the microphone at the moment of noise rating and is 69 feet from it. The tire source contribution by this approximation is 2.8 dBA less than its peak coastby level at test speed. Detailed analysis gives the decrement as 2.6 dBA.

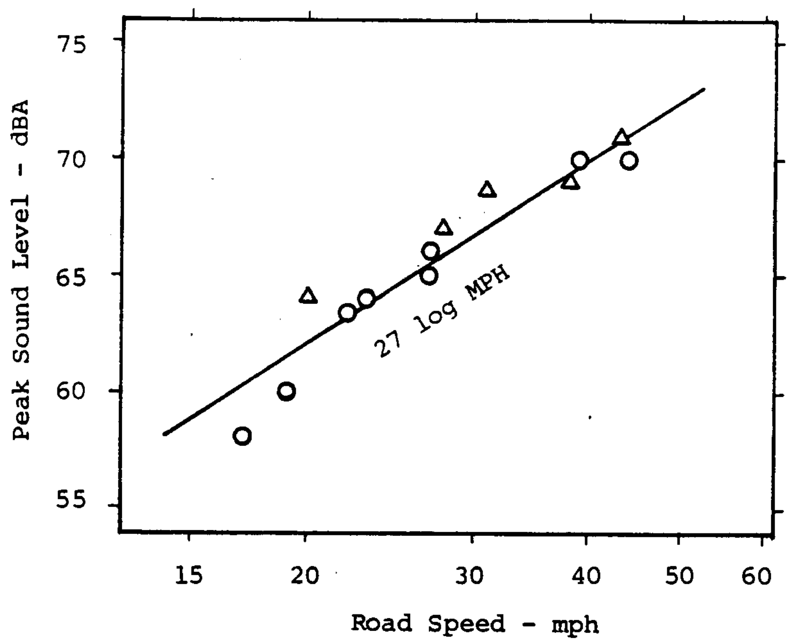
For the analysis, these assumptions are made:

1. Each wheel is radiating omnidirectional sound from the tire-pavement interface.
2. $L_2 = L_1 - 3$
3. $L_3 = L_1 + 1$
4. $L_4 = L_1 - 2$

Where, in the direction of the microphone at one foot:

L_1 = Sound level from near side front wheel.
L_2 = Sound level from far side front wheel.
L_3 = Sound level from near side rear wheel.
L_4 = Sound level from far side rear wheel.

Circles: Bus 324
Triangles: Bus 807



	Bus 324	Bus 807
Make	Firestone	Firestone
Carcass	Radial	Bias-ply
Fabric	Steel	Nylon
Tread	Rib-type	Rib-type
Size	12.75 R 22.5	12.5 x 22.5
psi-front	100	95
psi-rear	100	90
Test date	Oct 1978	Apr 1980

FIGURE 20.

COASTBY SOUND

50-foot Microphone Distance
Engine Off, Transmission in Neutral
Ordinary Dry Asphalt Concrete Pavement
Tubeless Tires, Partly Worn
Portland International Raceway

Bus length	35' 8"
Wheelbase	225"
Rear overhang	100"
Front axle track	85¼"
Rear axle track	76½"

Solutions are:

1. The left front tire level at one foot is L_T(peak) + 28.6 dBA.
2. The center of the bus is 1½ feet uprun from the microphone's perpendicular when peak coastby sound occurs.
3. The total tire sound when the rear of the bus is 30 feet past the microphone is 2.6 dBA less than peak...a source contribution of 61½ dBA at a first upshift speed of 24 mph.

There isn't much that can be done...or needs to be done...about transit bus tire noise.

2.3.5 Air Dryer Sound

The air dryer is a safety device, yet it sounds dangerous. Even though air dryer hiss is not counted in standard bus noise ratings[1], it is annoying. It also adds to the ambient Leq where buses are operating, especially around bus stops. There is no excuse for subjecting the public to this noise. It can be eliminated easily and harmlessly by simply attaching a long tube to its exhaust port.

All buses in the Tri-Met fleet have air dryers. Most have the Bendix AD-2. Some have the older Bendix AD-1. Still others have the Graham-White 918 007. All work in essentially the same manner. Located just downstream from the air compressor, the dryer removes condensate from the newly compressed air before the air reaches the valves, chambers, and resevoirs of the air brake system where freezing might cause blockage. Each time the compressor replenishes the air storage tanks, the same signal that unloads the compressor opens the dryer's sump valve, allowing the pent up air within (about 1 gallon at 120 psi) to blow the collected condensate down onto the street below. This causes a loud, prolonged jet noise. Often this will occur about a half-block after the bus pulls away from a boarding station. Since it sounds like something dramatic is happening with the airbrakes, it scares some people, creating an unnecessary sense of alarm and annoyance. Figure 21 depicts an effective and proven way to eliminate dryer hiss.

LEGEND
1 Bendix air dryer
2 1-1/2" Hose clamp
3 3/4" x 2-1/2" Heater hose
4 3/8" NPT Coupling
5 3/8" Compression elbow
6 3/8" OD x 14' Copper tube

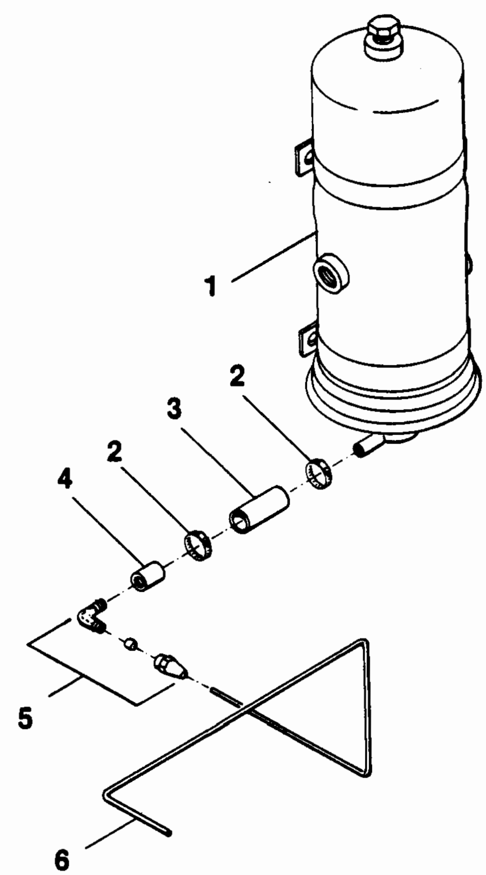


FIGURE 21.

AIR DRYER SILENCING

[1] It is loud enough to spoil an ordinary noise rating test run, however.

Two means of silencing were evaluated. The first was the run of 3/8" copper tube as illustrated in Figure 21. The second was directly attaching a small off-the-shelf muffler designed for this purpose. It was an Automatic Valve 84A-5. This consisted of a 1¼" diameter, 4" long slotted aluminum cylinder filled with woven steel mesh. Both treatments work by restricting air flow, forcing it to bleed out more slowly and thus reducing jet noise.

A typical bus was undergoing testing of an engine compartment lining kit. It was used to test the two air dryer treatments. It was parked on the test track with its rear end 30 feet beyond the microphone. Its engine was idled to give the lowest possible background sound. The sequence leading to dryer hiss was triggered by repeated brake applications. Table 17 gives results.

TABLE 17.

AIR DRYER HISS TEST
Bus 324
1972 Flxible® Model 111D-D061
Insul-Quilt™ Engine Compartment Lining Kit
Fan Held Still
500 rpm Engine Idle
Parked - Rear 30 feet Beyond Microphone
Levels Same for Both Sides
Ambient 50 - 55 dBA
Portland International Raceway, March 1979

	Sound Level - dBA			
	Background	+Dryer Hiss	Source Level	Attenuation
No treatment	62 - 63	74½	74	
Small muffler	62 - 63	67	65	9
Long thin tube	65½ - 66	65½ - 66	<56	>18

The dryer was about 64 feet from the microphone when its source level of 74 dBA was measured. A pedestrian 6 feet away would hear 95 dBA.

The small muffler helped, but dryer hiss was still clearly audible. With the long 3/8" tube, the dryer hiss was faintly audible, but no increase in level could be seen on the meter.

The 3/8" tube was field tested for 10 months (50,500 miles, 3,600 engine hours) with the 3/8" tube terminated at a fitting into the exhaust pipe upstream of the muffler. It was thought that the condensate and other debris would be incinerated by the hot exhaust gases. Inspection after the field test found an ash deposit blocking the passage through the fitting. Terminating the 3/8" tube at the firewall and just letting it empty into the noisy engine compartment solved the problem.

The same treatment of exhausting through a long thin tube should be effective in reducing the other hissing sounds from a bus such as when the brakes are released and the doors are operated. Such treatment was not attempted or field tested during the Tri-Met program. The effect of increasing brake release time would need to be investigated.

2.3.6 Other Exterior Sounds

Exhaust shell sound makes a minor contribution to the overall exterior noise rating. The typical transit motorbus muffler is a large cylindrical shape fabricated of thin steel. It is slung in a floorless compartment centered beneath the bus just forward of the engine compartment. In and out pipes go through a large hole in the firewall from the engine and to the stack. Both pipe joints are on the rear face of the muffler.

Sound radiates from the muffler body and from the runs of pipe. No flexible pipe sections are provided to accommodate intermotion between the engine and muffler, so this possible source of shell noise is removed. On the other hand, pipe joints at the muffler and engine manifolds often wear or break open, letting out loud sound.

The sound radiates from the exhaust piping within the engine compartment, but these radiations are lost with the other sounds there. When the muffler is located in a separate place, it can be considered a separate source.

An experiment with a typical bus was done to evaluate exhaust shell sound coming from the muffler compartment. This was done by making normal 50-foot noise rating passbys with the fan held still and an absorptive barrier temporarily forming a floor to the muffler compartment. The barrier nullifies exhaust shell radiations as well as that portion of engine compartment sound that travels out through the hole in the firewall for the exhaust pipes. The exhaust terminus and tire source contributions had been previously measured.

Raw data were:

	Noise Rating - dBA	
Muffler Cover	Left Side	Right Side
Off	80½	75½
On	79	74½

Table 18 and Figure 22 together perform a source analysis arriving as an exhaust shell contribution of 68 dBA to both right and left side baseline ratings. The following assumptions are made:

1. The sound from the muffler compartment is the same on both sides of the bus.
2. The engine sound energy reaching the exterior measurement station from the firewall hole is one-twentieth of the sound energy from beneath the engine compartment. This is the ratio of these path areas.
3. Due to the transmission's barrier effect, the engine compartment sound on the right side of the bus is 2 dBA less than on the left side...not counting sound emitted by the radiator grille.

The analysis shows how important the radiator opening is to the noise rating of a typical bus.

TABLE 18.

SOURCE ANALYSIS OF EXHAUST SHELL TEST
Bus 807
1973 GMC Model T8-H5307A
Fan Held Still
Muffler Compartment Cover Experiment
Portland International Raceway, April 1980

	Noise Rating - dBA			
	Cover Off		Cover On	
Source	Left	Right	Left	Right
Engine	79.7	73.7	78.2	73.5
Shell	68.2	68.2	nil	nil
Exhaust	71	67	71	67
Tires	58	58	58	58
	80½	75½	79	74½

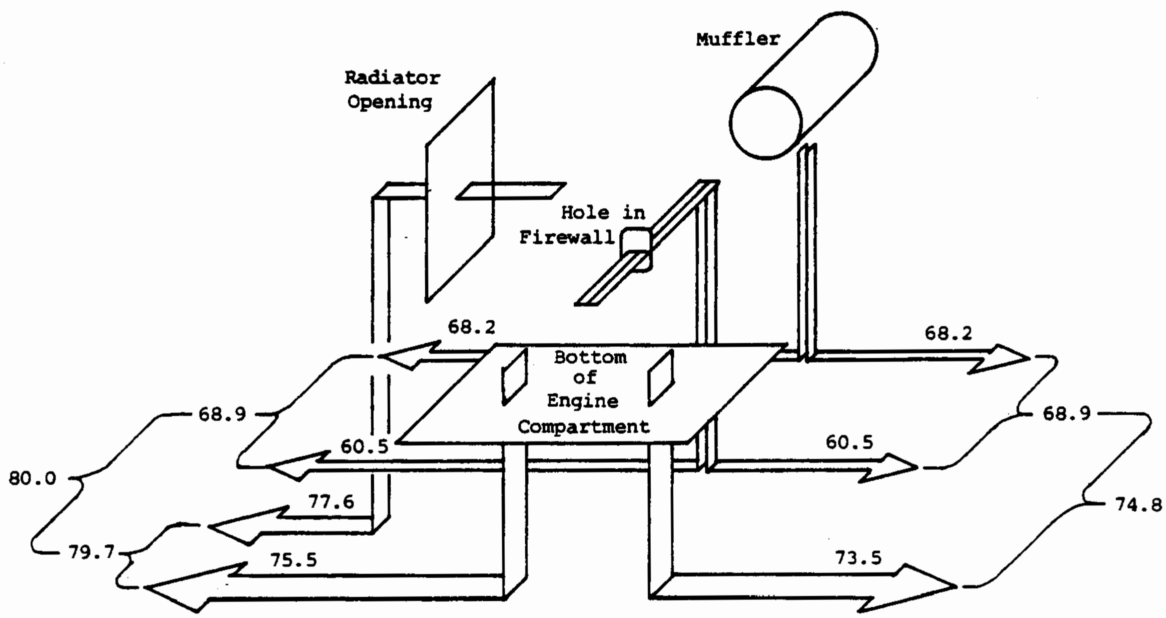


FIGURE 22.

SOUND PATHS AND SOURCES FOR EXHAUST SHELL TEST
Bus 807
1973 GMC Model T8-H5307A
Fan Held Still
Muffler Compartment Cover Experiment
Portland International Raceway, April 1980

Air intake sound on a typical transit motorbus would radiate from the screened inlet on the right rear corner of the bus body, on a level with the rear window. It is so weak as to be completely masked by other bus sounds, even after they have been given effective treatment. An experiment was conducted using a 1972 Flxible® Model 111D-D061 which was in the process of receiving an antinoise kit. An absorptive baffle was attached over the inlet which directed its sound completely away from the right side of the bus. Right side noise ratings were 73 dBA with and without the baffle, indicating that the air intake source contribution must be less than 63 dBA on the right side and even less than that on the left side. The remaining sources were relatively low: The orginal equipment muffler brought the exhaust terminus contribution down to 62 dBA, a simulated absorptive bellypan blocked a large amount of engine sound, and the fan drive was in normal mode during the sub-50°F winter weather.

These test results were supplemented by personal inspection. Only a faint sound could be heard above background when placing the ear directly over the intake inlet with the bus parked and the engine revving.

If the intake manifold is removed from an 8V-71N engine, one hears a loud whine from the Rootes blower as the engine is revved. Evidently, the pleated paper element in the typical air cleaner together with the rest of the air intake system provides all but complete silencing of this source of sound.

2.4 TECHNIQUES FOR DETERMINING SOURCE CONTRIBUTIONS

Once the noise rating of a bus is established by following the standard procedure, one wants to know how much noise is coming from the several sources. At the moment when the overall sound is greatest, what is the sound level of each source arriving at the microphone? At this moment, the rear of the typical bus is 30 feet downrun from the mirophone's perpendicular.

2.4.1 Tires

The sound created at the tire-pavement contact patch together with aerodynamic sound and drive gear sound make up what we call here as "tire" sound. A series of coastby runs with the engine off and the transmission in neutral establishes the relationship of peak sound level with road speed. The peak level can then be determined for the road speed when first upshift occurs. The tire source contribution will be somewhat less than the peak during coastby because the tires are further away. A close estimate is obtained from:

$$L(\text{tire}) = L(\text{peak}) - 20 \log \frac{\sqrt{\left(\frac{OAL}{2} + 30\right)^2 + 50^2}}{50}$$

where:

L(tire) = Tire source contribution to overall noise rating - dBA
L(peak) = Peak sound level from coastby testing at road speed when overall noise rating is taken - dBA
OAL = Overall length - feet

Only one side needs to be tested since tire sound is the same for both sides.

2.4.2 Exhaust Shell

Sound radiating from the exhaust muffler and associated piping is determined by wrapping or covering the muffler and logarithmically subtracting the overall noise rating obtained with this treatment from the overall noise rating obtained without this treatment. The equation is:

$$L2 = 10 \log \left(10^{\frac{LT}{10}} - 10^{\frac{L1}{10}}\right)$$

where:

L2 = Sound level of a source - dBA
L1 = Combined sound level of all other sources - dBA
LT = Overall sound level - dBA

Since exhaust shell radiation is normally low, it would give more accurate results if the baseline sound is kept as low as is convenient...such as turning off the fan, for instance.

Both sides need to be tested since shell noise is not necessarily omnidirectional.

2.4.3 Exhaust

By measuring the exhaust source at 1 foot from the terminus center under the same operating conditions as when the overall rating is obtained, the other more intense sounds, such as from the engine, fall back to relatively low background levels. This entails onboard sound instruments and a means of mounting the microphone close to the terminus. Precautions are taken to shield the microphone from engine sound. A clamped pipe extension is usually necessary. Wind noise is measured by engine-off coasting at the rating test speed. This value is logarithmically subtracted from the raw value at 1 foot. An estimate for engine background sound is also subtracted from the 1-foot raw value. The remainder is the exhaust sound level at 1 foot. A correction is applied to theoretically move the receiving point out to the position of the microphone at the moment when the overall rating is taken. Figure 23 shows the geometry of the situation for a typical bus.

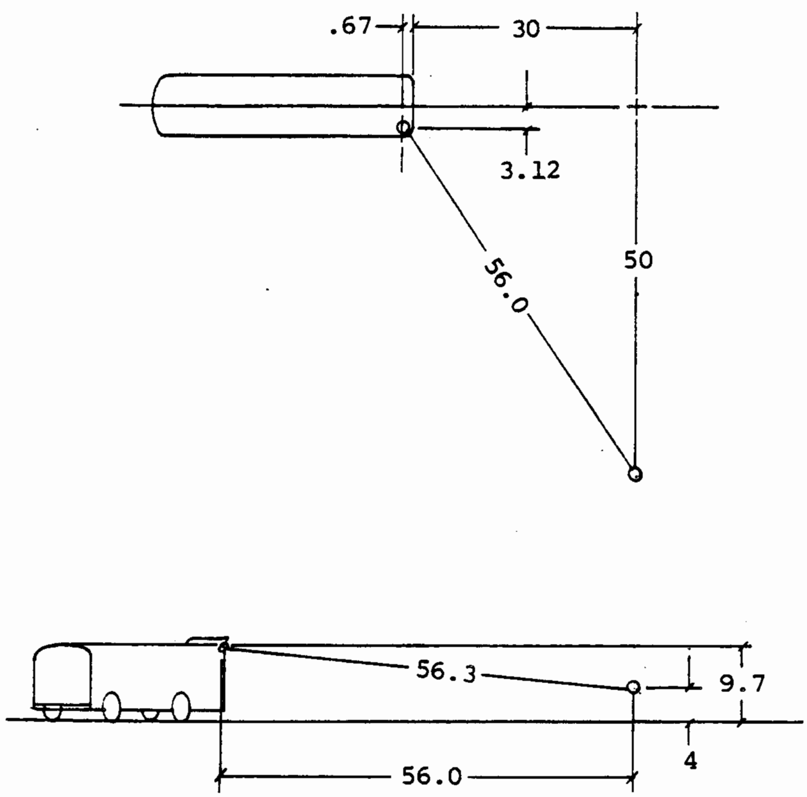


FIGURE 23.

TYPICAL DISTANCE CORRECTION FOR EXHAUST SOUND
1973 GMC Model T8-H5307A
All Distances in Feet

	Left Side	Right Side
Microphone Distance *Distance Correction	56.3 ft -35 dBA	61.6 ft -36 dBA

*20 log D

A shielding adjustment is made for right side exhaust ratings if the terminus is partly hidden from the microphone by the bus body. Table 19 shows how the foregoing considerations are brought together in the derivation of a typical bus's exhaust source contribution.

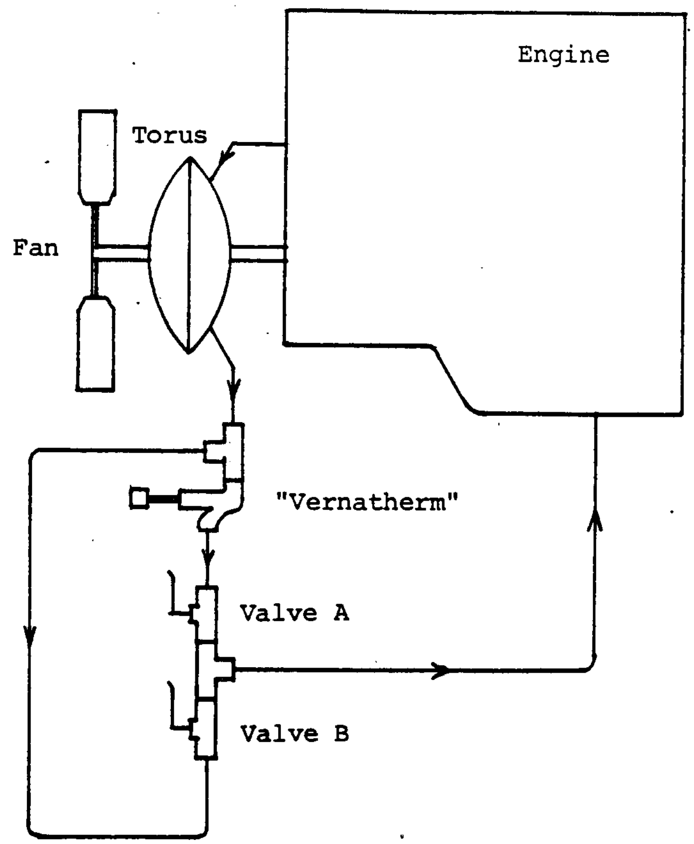
TABLE 19.

TYPICAL DERIVATION OF EXHAUST CONTRIBUTION
1973 GMC Model T8-H5307A

	Sound Level - dBA	
	Left Side	Right Side
Raw Data at 1 Foot Engine Background Wind	106½ -89 -86	106½ -89 -86
Exhaust at 1 Foot	106.4	106.4
Distance Correction Shielding	-35 -	-36 -3
Exhaust Contribution	71	67

2.4.4 Fan

The fan's contribution to the overall rating is determined by logarithmically subtracting the rating with the fan turned off from the rating with the fan turned on. In the case of the GMC hydrodynamic fan drive, the thermostatic control valve may be overridden as shown by the diagram in Figure 24. If the installation of this plumbing is not justified, the fan may be turned off in the field by temporarily bypassing the control valve. Blocking the control valve outlet fitting with a brake shoe rivet (old mechanics trick) will cause the fan to turn at full speed continuously.



	FAN ON	FAN NORMAL	FAN OFF
A	CLOSED	OPEN	EITHER
B	CLOSED	CLOSED	OPEN

FIGURE 24.

PLUMBING FOR GMC FAN CONTROL

2.4.5 Engine

All the sounds remaining come from the engine compartment and are grouped together as the "engine" source. This includes transmission gear sound and engine accessories that haven't been isolated by muffling or by on-off tests. The "engine" source is what's left over after logarithmically subtracting all the other source contributions from the overall rating.

2.5 "NEW LOOK" RETROFIT KITS

Tri-Met applied similar noise treatments to two representative buses, a 1972 Flxible® and a 1973 GMC. They were extensively field tested. The treatment was taken up to the level where a bellypan would be the next step. Essentially, the engine was turbocharged and the engine compartment was given an acoustically absorptive lining. Figure 25 is an illustration. Table 20 summarizes. More details are found in References 1 and 2. Tables 21 and 22 provide source analysis and performance benchmarks. Appendix B presents a broader perspective of motorbus noise treatment.

Results of the treatments were modest and uniform. Both buses were brought to reasonably low noise levels without unacceptable side-effects. Field tests were uneventful except that, in both cases, the absorptive lining wore badly whenever there was the slightest chafing. Otherwise, it lasted well.

TABLE 20.

SUMMARY OF TREATED BUS NOISE RATINGS
"New Look" Motorbuses

	1972 Flxible®	1973 GMC
EXTERIOR Ave R & L Fan as-is	75½	75½
Improvement	4	3½
INTERIOR Ave F & R	76½	72½
Improvement	3	3

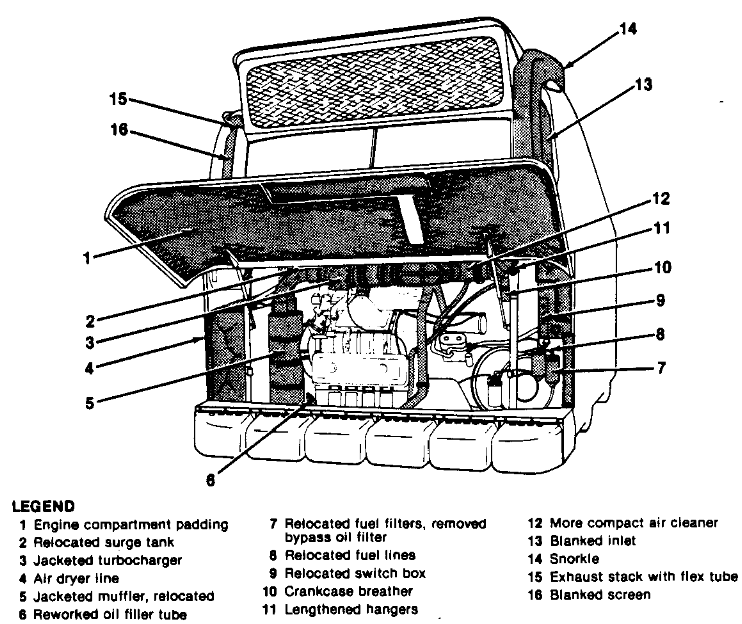


FIGURE 25.

NOISE TREATED MOTORBUS
1973 GMC "New Look"

TABLE 21.

NOISE SOURCE ANALYSIS OF TREATED MOTORBUSES

1972 Flxible®

BASELINE

EXTERIOR			LEFT	RIGHT
	SOURCE	ENGINE	79	77½
		SHELL	73	70
		EXHAUST	66	65
		TIRES	58	58
		FAN ON	77	76½
		FAN AS-IS	71	70½
	OVERALL	FAN ON	*82	81
		FAN AS-IS	80	79
		FAN OFF	80	78½
INTERIOR			**REAR**	**FRONT**
		FAN ON	*83½	77½
		FAN AS-IS	83	76

TREATED

EXTERIOR			LEFT	RIGHT
	SOURCE	ENGINE	74½	73½
		SHELL	NIL	NIL
		EXHAUST	65	64
		TIRES	58	58
		FAN ON	77½	73
		FAN AS-IS	71	65
	OVERALL	FAN ON	*79½	76½
		FAN AS-IS	76½	74½
		FAN OFF	75	74
INTERIOR			**REAR**	**FRONT**
		FAN ON	*80½	74½
		FAN AS-IS	80	73

1973 GMC

BASELINE

EXTERIOR			LEFT	RIGHT
	SOURCE	ENGINE	78	73½
		SHELL	72	65
		EXHAUST	71	67
		TIRES	58	58
		FAN ON	80½	77½
		FAN AS-IS	77½	73
	OVERALL	FAN ON	*83½	79½
		FAN AS-IS	82	77
		FAN OFF	80	75
INTERIOR			**REAR**	**FRONT**
		FAN ON	*79	72½
		FAN AS-IS	78	72½

TREATED

EXTERIOR			LEFT	RIGHT
	SOURCE	ENGINE	77	72
		SHELL	NIL	NIL
		EXHAUST	63½	59½
		TIRES	58	58
		FAN ON	77	74½
		FAN AS-IS	70	62½
	OVERALL	FAN ON	*80	76½
		FAN AS-IS	78	73
		FAN OFF	77	72½
INTERIOR			**REAR**	**FRONT**
		FAN ON	*76	69
		FAN AS-IS	75½	69

*Comparable to EPA Rating

TABLE 22.

PERFORMANCE BENCHMARKS OF NOISE TREATED BUSES

1972 Flxible®

	UNITS	BASELINE	TREATED
POWER OUTPUT			
Terminal Speed on 4% Grade	mph	45½	46
Acceleration on Flat [1]200-foot Time	sec	8.7	8.8
[2]Engine Dynamometer Full throttle, 2,000 rpm	BHP	218	220
FUEL MILEAGE			
Average Mission	mpg	[3]4.22	[4]4.40
COOLING SYSTEM			
Full Speed on 4% Grade	[5]°F ATB	134°	127½°
Fan Delivery, 1,500 rpm	CFM	10,500	12,000
ASPIRATION (Full Power, 2,000 rpm)			
Exhaust Back Pressure	"Hg	5.6	2.1
Exhaust Temperature	°F	810°	560°
Intake Restriction	"H_2O	7	11

1973 GMC

	UNITS	BASELINE	TREATED
POWER OUTPUT			
Terminal Speed on 4% Grade	mph	44	48½
Acceleration on Flat [1]200-foot Time	sec	8.7	9.2
[2]Engine Dynamometer Full throttle, 2,000 rpm	BHP	218	
FUEL MILEAGE			
Average Mission	mpg	[3]4.16	[4]4.26
COOLING SYSTEM			
[5]ATB on 4% Grade	°F	132	130
Fan Delivery, 1,500 rpm	CFM	9,600	9,600
ASPIRATION (Full Power, 2,000 rpm)			
Exhaust Back Pressure	"Hg	6.2	3.5
Exhaust Temperature	°F	895	600
Intake Restriction	"H_2O	12	15

2.6 ENVIRONMENTAL ASPECTS

2.6.1 Guidelines and Standards

To avoid possibilities of hearing loss, the Leq(24) should be kept below 70 dBA. To avoid residential annoyance problems, the Ldn should be kept below 55 dBA. Higher levels than these can be acceptable, but not without cost or compromise.

The most commonly accepted guide (not a standard because of not being subjected to economic feasibility) for tolerable environmental noise is contained in the 1974 EPA "Levels" Document.[1] Table 23 is taken from this reference.

TABLE 23.

SUMMARY OF NOISE LEVELS IDENTIFIED
AS REQUISITE TO PROTECT
PUBLIC HEALTH AND WELFARE
WITH AN ADEQUATE MARGIN OF SAFETY

Effect	Level	Area
Hearing Loss	Leq(24) $\leq$ 70 dBA	All areas
Outdoor activity interference and annoyance	$L_{dn} \leq$ 55 dBA	Outdoors in residential areas and farms and other outdoor areas where people spend widely varying amounts of time and other places in which quiet is a basis for use.
	Leq(24) $\leq$ 55 dBA	Outdoor areas where people spend limited amounts of time, such as school yards, playgrounds, etc.
Indoor activity interference and annoyance	$L_{dn} \leq$ 45 dBA	Indoor residential areas.
	Leq(24) $\leq$ 45 dBA	Other indoor areas with human activities such as school, etc.

A more recent consensus among federal agencies is found in the 1980 Guidelines for Land Use Planning.[2] This reference concludes that:

1. Residential land use is compatible without restrictions with an Ldn of 55 dBA or less. Where the Ldn is between 55 dBA and 65 dBA, this land use is somewhere between "all right" and "discouraged", but is still deemed compatible considering cost and technical feasibility factors.
2. Use of land for general downtown purposes that is compatible without restrictions ranges from where the Ldn is 55 dBA for parks, hotels, and schools to 70 dBA for office buildings and stores. Ldn's above 80 dBA are deemed incompatible with most downtown land uses.

[1] Reference 4.
[2] Reference 5.

2.6.2 Transit Mall Sound

The sound a bus makes while maneuvering for a bus noise rating test is one thing, but the sound it actually makes in the real world is another. Its real world sound depends more on how it is driven than on how loud it can be forced to be. Take, for example, a Transit Mall. Here, in the absence of automobiles, one bus after another will approach a bus stop under power; slow to a stop, usually with audible brake squeal; wait, with engine idling while passengers get off and get on; then accelerate away with concomitant air hissing sounds as brakes release and air dryers exhaust. Other times, a bus will drift by a bus stop and the next intersection quickly and with little sound if there are no boarders there and the traffic light is green.

2.6.2.1 Levels on the Mall - A station was set up to monitor sound at the heart of the Portland Transit Mall, a buses-only one-way couplet ten blocks long through the central business district. Sound was sampled before, during, and after the Mall's construction. During construction, for a time, all buses were channeled two-way onto the Sixth Avenue corridor. As shown in Table 24, the sound on the Mall did not turn out to be appreciably louder than the previous traffic mix.

TABLE 24.

IMPACT OF THE MALL ON AMBIENT SOUND
Meier & Frank Department Store
Third Floor Midblock Sixth Avenue

	Average Traffic	Leq(0700-1800)
Before (1975)	General Mix	71.3 dBA
During (1976)	152 Buses/hr	74.0 dBA
After (1982)	92 Buses/hr	72.3 dBA

The character of the sound is what changed. Instead of a steady stream of mostly automobile sounds, one hears relatively loud episodes of sound as individual buses pass against a backdrop of relatively quiet background sound.

At the Meier & Frank station, a correlation was obtained between bus passage rate and sound level. Figure 26 shows the fluctuation of sound levels and bus rates during an ordinary weekday. When buses passed more frequently during the morning and evening rush hours, sound levels also increased.

The least squares best fit first order equation correlating Leq with bus rate for the 20 minute samples from Figure 27 is:

$$\mathrm{Leq} = 58.1 + 7.25 \log R$$

Mall sound is not very sensitive to bus rate. Doubling buses per hour increases Leq by about 2 dBA. Saturation of the Portland Mall occurs at about the rate of 100 buses per hour average 0700 to 1800. The bottleneck is access on and off the Mall, not the Mall itself.

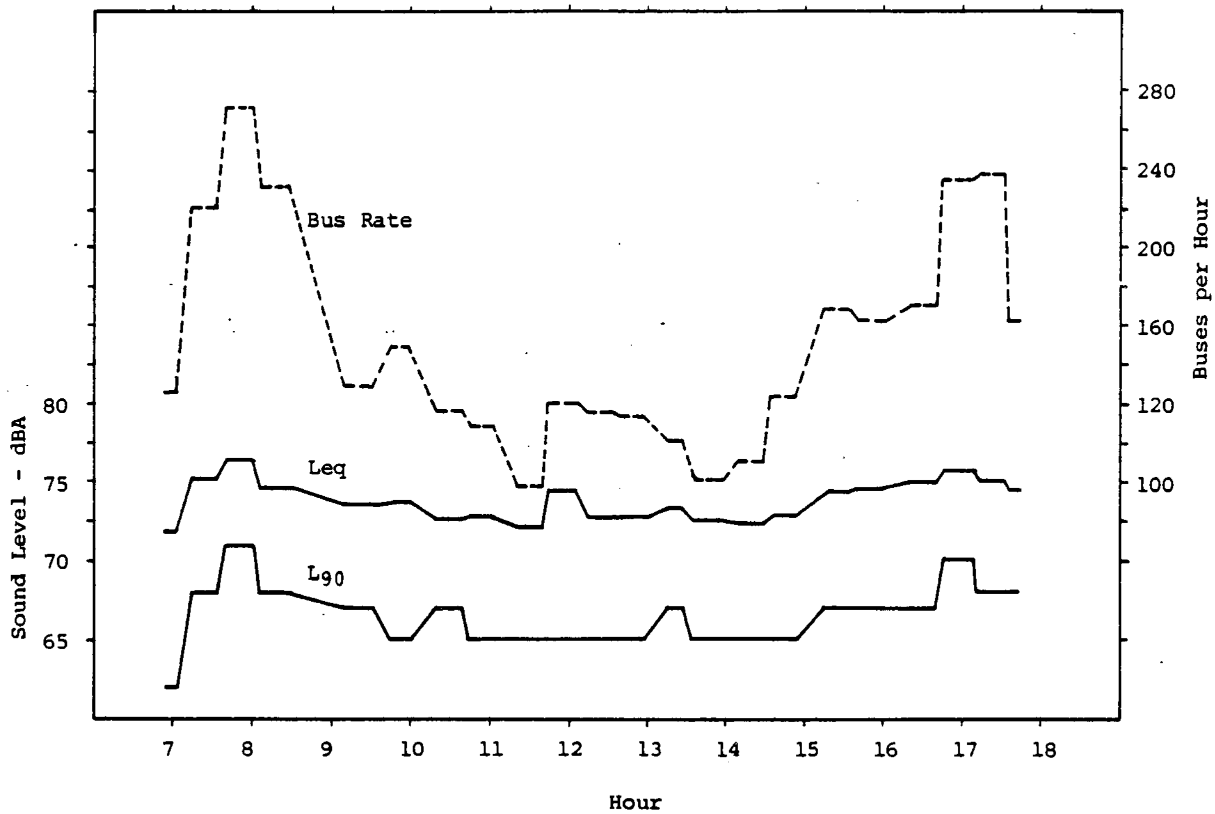


FIGURE 26.

AMBIENT SOUND AT PORTLAND TRANSIT MALL

Meier & Frank Department Store
Third Floor Fire Escape
Midblock Over Bus Stop
Construction Phase
Wednesday 3-3-76

Measurement Time Window	413 Minutes
Total Bus Count	1,050 Buses
Average Leq	74.0 dBA
Average L_{90}	67.0 dBA
Average Leq(hr) per bus	51.2 dBA

2.6.2.2 <u>Canyon Effect</u> - Manmade canyons are formed when downtown streets are lined with tall buildings. The sound from a passing bus might be less than it would be if the buildings and their echoes weren't there. Some call the difference "canyon effect", and some fear it is more dramatic than it really is.

The noise rating of a typical motorbus was determined by tests on an apron at the Portland International Airport. The bus was then taken downtown and retested on Fifth Avenue, in the "canyon" there. Each measurement was made with the microphone placed 50 feet from the centerline of the lane of travel, with the bus accelerating at full throttle, and with the rear of the bus 20 feet beyond the microphone's perpendicular at the moment when 10 mph was reached. The results listed on Table 25 indicate that the canyon effect on peak sound levels as buses pass by streetlevel stations is in the range of +1 to +2 dBA.

TABLE 25.

CANYON EFFECT AT STREET LEVEL
Full Throttle Passby, Bus 304
50-foot Distance, 10 mph
Background 60 to 65 dBA
Early Evening, 7-16-75

Station	1	2	3
Side Toward Microphone	Right	Left	Left
Peak Level Downtown	73	79½	79
Peak Level at Airport	77½	78	78
Canyon Effect	-3½	+1½	+1

Station 1.

On the grass slope directly in front of the Pioneer Post Office between S.W. Morrison and Yamhill. More people wait for buses at this place than at any other stop in the city. Across the street stands an unbroken wall of tall buildings.

Station 2.

Across the street from Station 1, midblock. One sees the old 3-story post office set back 100 feet from the sidewalk.

Station 3.

Midblock in front of the Aus Building between S.W. Alder and Washington, 2 blocks north of Stations 1 and 2. Tall buildings crowd up to the sidewalk on either side of Fifth Avenue, creating as extensive a "canyon" as anywhere in the city.

As one moves away from a source of sound in a partially reverberant chamber (a downtown "canyon" is one), the received sound level diminishes with distance, but not so abruptly as it would in an echo-free space. At a certain distance, the received level stays nearly constant despite any further increase in distance.

The Meier & Frank Department Store is in a downtown canyon. Simultaneous sound measurements were made on the sidewalk midblock and at several floors outside the building directly overhead during heavy bus traffic. Table 26 indicates that sound levels did not diminish with distance as much as they would in an anechoic space.

TABLE 26.

CANYON EFFECT ON A TALL BUILDING
Meier & Frank Department Store
S.W. Sixth Avenue
Midblock Over Bus Stop
10-22-75 and 3-3-76

Station	ΔL - dBA
Sidewalk	datum
1st Floor	-3
3rd Floor	-3
6th Floor	-4

Phalanxes of tall buildings downtown probably have more beneficial effect upon noise than detrimental. While their echoes do reinforce radiations from sound sources in their own "canyons" to a slight extent, they act as formidable barriers to sounds emanating from sources on other streets. One hardly hears even loud sounds coming from the other side of the block.

2.6.2.3 Predicting Mall Sound - When a bus passes by a station, a dose of its sound is received. This dose may be found by logarithmically subtracting the background level (taken to be the L_{90}) from the overall Leq and may be expressed as an hourly Leq. If a group of buses happens to be included in a sample, the average sound per bus is obtained by dividing the total bus sound by the number of buses. Performing this analysis for the 20 minute samples from Figure 27 gives the average Leq(hour) per bus as 51.2 dBA. Making the assumption that two average buses make twice as much sound as one average bus, regardless of bus passage, results in the equation:

$$\text{Leq(hour) per bus} = 10 \log \left(10^{\frac{Leq}{10}} - 10^{\frac{L_{90}}{10}} \right) = 10 \log R + 51.2$$

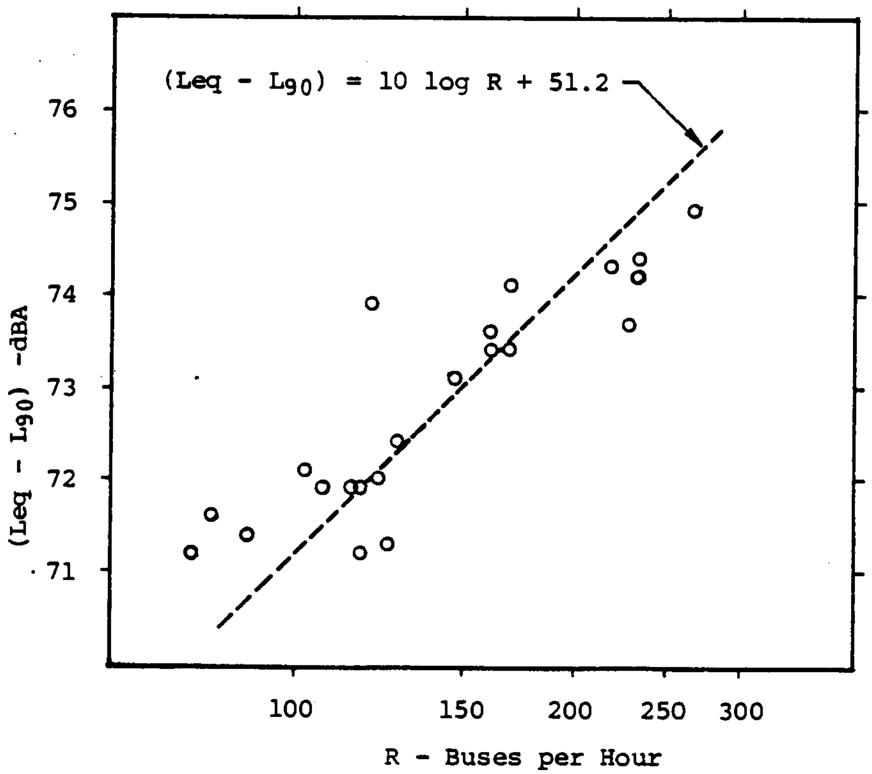


FIGURE 27.

CORRELATION OF BUS SOUND WITH BUS RATE ON MALL
Meier & Frank Department Store
Third Floor Fire Escape
Midblock Over Bus Stop
Construction Phase
Wednesday 3-3-76

This equation fits the observed data pattern in Figure 27 very well.

Six years later, looking for a method of predicting ambient sound caused by urban traffic, the Meier & Frank station was again used to measure bus sounds. This time, a deliberate attempt was made to capture individual bus passes. There was no correlation between the peak sound level observed during a bus pass and the average sound, the Leq(hour) per bus. Individual bus sounds ranged from 42 dBA to 55 dBA, but the average of a group of about 15 matched the average of much larger groups extremely well as Table 27 shows.

TABLE 27.

AVERAGE BUS SOUNDS AT THE MEIER AND FRANK STATION
Meier & Frank Department Store
Third Floor Fire Escape
S.W. Sixth Avenue
Midblock over Bus Stop

Sounding	Leq(hour) per Bus	Buses in Sample
3-3-76	51.2	1,050
9-21-82	51.4	14
9-24-82	50.6	29
	51.2	1,093

Ambient sound at a particular station where urban traffic is the dominant source may be predicted by following these steps:

1. Determine the average Leq(hour) per vehicle for each class. Sorts into bus and non-bus, where transit malls are the issue, are sufficient. Only those vehicles that pass directly by the station need be considered. All others are relegated to background.
2. Forecast background sound. Usually background sound will be the same in the future as it is in the present. The L_{90} adequately represents background sound.
3. Forecast vehicle volumes and calculate sound produced by each class.
4. Determine spurious loud sounds from emergency vehicle sirens, aircraft flyovers, explosions, sonic booms, bells, etc. Represent these by an average Leq(hour).
5. Total the forecasted Leq's with and without spurious sound.

This method requires some field measurements, but it does take into account the exact traffic situation and sound propagation peculiarities of each site which are normally difficult to predict.

2.6.3 Contribution of Buses to Street Sound

This question is sometimes asked: How much less noise would there be if diesel motorbuses were silenced? The answer depends greatly on the background sound level.

In a quiet residential neighborhood, a bus idling at a route turnaround can be the chief source of sound. If its engine were to be turned off, nearby ambient levels might well appreciably subside. In this case, if bus noise control treatments were applied, their improvements might well be deemed cost-effective. The resident of a house on a neighborhood street which happens to be used by buses deploying from a nearby bus yard in the early morning is quite likely to be relieved if the buses were to use another route instead.

On the other hand, Table 28 shows that no one would notice the difference in noise level if all buses were removed from bus routes along busy arterials. They are outweighed by all the other traffic. Sound measurements were taken at sidewalk stations on the three busiest Tri-Met bus routes during rush hour. Sound levels without buses were estimated by scissoring out the time segments when buses were passing. Incidentally, peak levels caused by "lunatic fringe" automobiles were consistently greater than those caused by typical buses. Examples were cars modified for high performance including low restriction exhaust mufflers being driven at excessive rates of acceleration, older vehicles with deteriorated exhaust mufflers, and heavy diesel trucks.

TABLE 28.

BUSY STREET NOISE WITH AND WITHOUT MOTORBUSES
Bus Routes Along Arterials
Evening Rush Hour Traffic
Sidewalk Stations
July 1976

		S.E. 30th & Hawthorn	N.E. 23rd & Ainsworth	N.W. 23rd & Everett
Leq	With Buses Without Buses	72.8 71.4	72.6 72.5	69.3 68.2
	Δ	1.4	.1	1.1
Buses per hour Bus Leq(hour) Leq(hour) per bus		24 67.2 53.4	12 ≈62 ≈51	15 62.8 51.0

2.6.4 Busyard Sound Barrier

Ranging in height from 6 to 15 feet above ground, the noise barrier at Tri-Met's Powell busyard has protected the immediate neighborhood of houses from increased ambient sound levels due to bus activity within the yard. The result is a uniform and acceptable Ldn of 54 to 56 dBA. Within the yard, just inside the barrier, an Ldn of nearly 70 dBA was measured. Sound levels during early morning hours at houses bordering the yard would be up to at least 13 dBA higher were it not for the barrier.

When the need for expansion was felt in 1977, Tri-Met elected to shorten deployment distances by establishing an additional depot rather than enlarge the existing one. Time and fuel were saved and traffic was reduced.

The location chosen for the new yard was the 15½ acre quadrant of the intersection of S.E. Powell Boulevard with the I-205 freeway. Figure 28 shows the site. Clusters of homes are situated to the south and southeast of the yard. The worst case is a one-story house whose north windows are barely 100 feet from the traffic lane within the yard. Oregon law prohibits new noise sources from raising neighbor's sound levels by more than 10 dBA at any time. Accordingly, noise reduction measures were taken, including the installation of a sound barrier around the southeastern flank of the property.

A barrier works by forcing sound to reach the receiver only by diffraction over the top. The greater the difference between the line-of-sight path from the source to the receiver and the over-the-top path, the greater is the barrier's attenuation. Thus, the worst place for a barrier is halfway between a source and its receiver. It becomes more effective as it is moved toward either the source or the receiver. For a given situation, higher frequencies are more attenuated than are low frequencies.

The Powell busyard barrier rises from a height of 6 feet above ground where its need is least, to 15 feet where neighbors are closest. To the outsider, it appears as a 6-foot high wooden fence made of horizontally arranged 3" x 12" timbers meandering over a planted berm ranging up to 9 feet high. To the insider looking southward across the cement concrete parking lot, it has the same appearance. Looking eastward, the 6-foot wooden fence is seen atop a concrete retaining wall that tapers from 9 feet high at the south end to a 6 foot height at the north end. Cracks between the timbers constitute less than 1% open area of the 6-foot fence. Here and there a tree or large shrub appears above the fence, but, in the main, the top of the barrier presents the clean profile conducive to sound attenuation.

In addition to the barrier, noise abatement is enhanced by locating the busy places...the garage, the wash rack, the fuel station, the entrance driveway... at the north end of the yard, away from the residences to the south.

Traffic and parking patterns take advantage of the characteristic that buses radiate less sound from their right side than they do their left. While parked, the buses present their right sides to the southern neighbors. While making a circuit of the yard, they move counterclockwise. This means they are travelling from west to east as they pass along the the noise-sensitive southern edge of the yard, also with their right sides toward the neighbors.

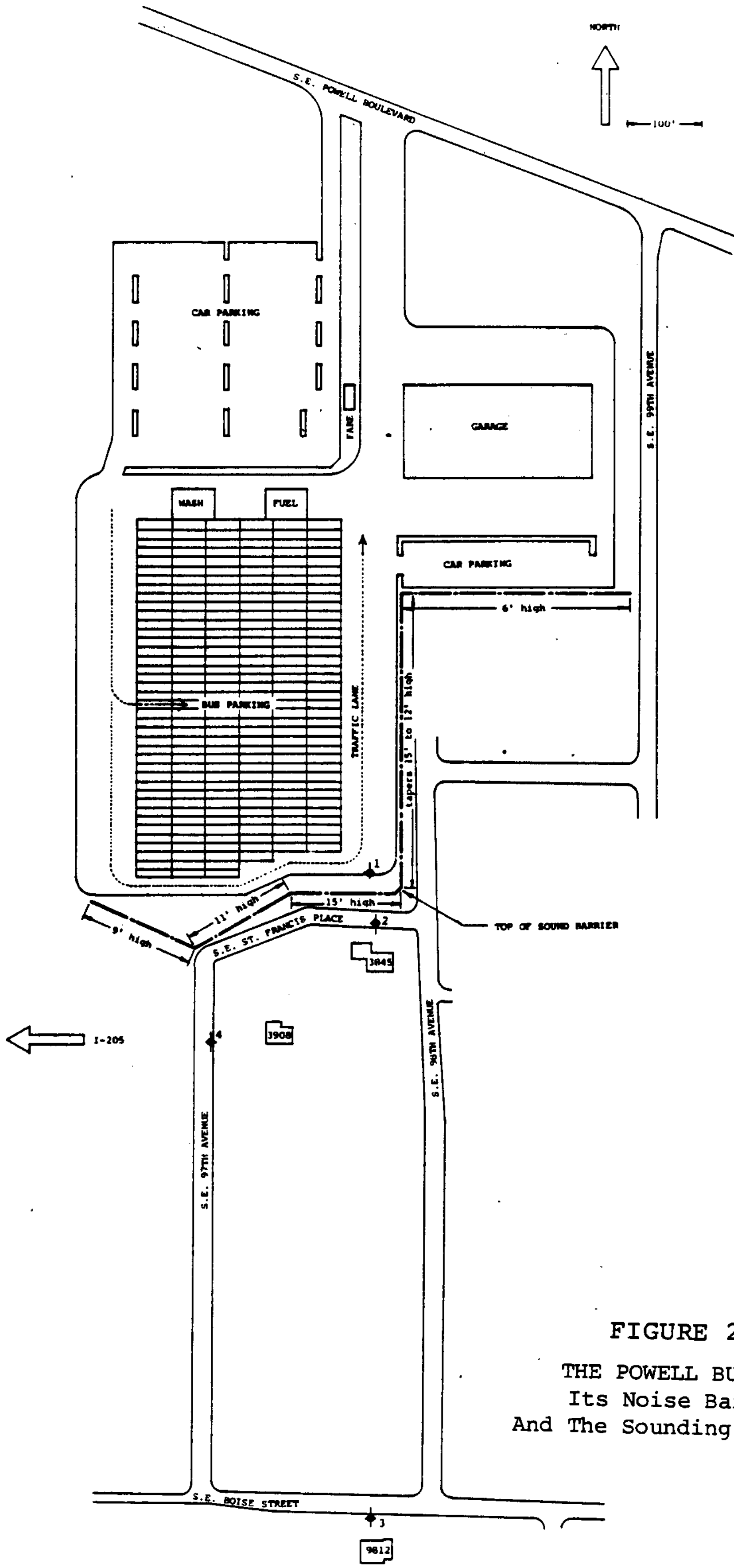


FIGURE 28.

THE POWELL BUSYARD
Its Noise Barrier
And The Sounding Stations

The bus yard is like a beehive. Some buses are making loud noises, some low noises, and some none at all. Some are stationary. Some are moving. Some are close to the critical south end, and some are distant. There is a rhythm of activity that repeats during a 24-hour cycle, but no two repeated hours are ever exactly alike. Anything can happen.

About 200 buses are assigned to the Powell busyard. During weekdays, there are 256 arrivals and an equal number of departures for scheduled routes. The bulk of the fleet deploys between 0400 and 0800 and again between 1400 and 1700, returning about 3 hours later.

The barrier's acoustic effectiveness was measured in August 1982 at 3 stations outside the yard and 1 station inside.

Station 1.
Just inside the yard, 23 feet on the other side of the barrier from Station 2.

Station 2.
Worst-case house, 40 feet on the other side of the barrier from Station 1.

Station 3.
Too far (890 feet from the barrier) to receive contributive yard noise, yet equidistant from the sunken I-205 freeway (200 yards) as Station 2.

Station 4.
213 feet from the barrier. Included to cover the possibility that the barrier might protect houses deep within its shadow, but allow yard noise to impact houses at intermediate distances.

Operations are reduced over the weekend. Because most of the buses have been idled for one or two days, they are all inspected every Sunday evening. A mechanic, looking for leaks and other malfunctions, goes through the parked mass, starting all the engines. After about an hour, most of the buses are at a low idle. A deep-throated rumbling sound pervades the yard. This steady sound was measured on both sides of the barrier. Results:

	dBA
Station 1	63
Station 2	50
Difference	13

Levels at the 3 neighborhood stations are the result of busyard noise reaching those locations plus distant background sound and local sound events. During the daylight hours there is enough local noise at each station to mask out most, if not all, of the busyard noise. However, the neighborhood settles down from midnight to 0500 and the traffic on the incompleted I-205 freeway becomes low. It is then that busyard noise would be most easily heard and also would be most annoying. An equivalent sound level for this part of the night was measured along with the usual Ld and Ln levels. Results are listed in Table 29.

TABLE 29.

SOUND LEVELS AROUND POWELL BUSYARD
On Both Sides of the Noise Barrier
August 1982

Station	1	2	3	4
Ld	63.2	53.2	54.8	52.2
Ln	62.8	48.2	45.6	46.4
Ldn	69.3	56.0	55.1	54.3
L(00-05)	62.1	44.8	43.6	43.3

Although the worst case house is a little louder, the ambient sound throughout the neighborhood is virtually the same and is right on the accepted 55 dBA Ldn guideline. The range of variation is only 1½ to 2 dBA among the three residential stations, no matter how the averages are developed. Even during the relatively quiet 0000 to 0500 time window, the sound at the worst case house is only 1½ dBA more than at the house down the street 2 blocks away. Meanwhile, the yard noise just on the other side of the barrier is 17 dBA louder. If the barrier isn't doing its job of protecting the neighborhood, this is the time when that would be perfectly clear.

A sound barrier is an important abatement weapon in the fight to keep busyard noise out of nearby neighborhoods. Also important is realizing where busyard sounds originate, placing these sources as far away as possible and either reducing them or directing their radiations away from sensitive sectors.

3. AUTOMOTIVE FACTORS

3.1 PERFORMANCE BENCHMARKS

When deciding on noise control treatment, it is necessary to consider what side effects there might be. A set of performance benchmarks developed by Tri-Met for this purpose are:

POWER OUTPUT
- Terminal Speed on 4% Grade.
- Acceleration on Flat...200-foot Time.
- Engine Dynamometer at Full Throttle, 2,000 rpm.

FUEL MILEAGE
- Average Mission mpg

COOLING SYSTEM
- ATB on 4% Grade
- Fan Delivery at 1,500 rpm

ASPIRATION (Full Power, 2,000 rpm)
- Exhaust Back Pressure
- Exhaust Temperature
- Intake Restriction

3.1.1 Terminal Speed at 4% Grade

S.W. Canyon Road typifies a real-world mission of sustained hill climbing coupled with high traffic flow where maximum power performance is demanded of a transit bus and maximum thermal stress falls upon the cooling system. Other bus operators wishing to make their own tests can find their own counterpart local grades.

An alternative is to do similar testing on a chassis dynamometer, but it has disadvantages. Results are overly sensitive to operator skill. Repeat test results scatter. Up a hill, all a driver has to do is floorboard the throttle and steer. The only thing that can go wrong is traffic congestion. Air flow through the cooling system and engine aspiration behavior is not the same in a chassis dynamometer room as it is out in the open in the real world, easily leading to false conclusions. The dynamometer room is not weather-dependent and the potential for comparing results with investigators elsewhere is good, but for Tri-Met, with its mild climate and need mainly to measure subtle differences, the dynamometer is less desirable than the real road up a hill.

S.W. Canyon Road is a major 6-lane arterial climbing westward through the hills bordering the Willamette River in Portland. The top of the grade (Sylvan exit) is about 3 miles from the start of the rise at S.W. Jefferson and 18th Avenue. There are no bus stops along this section. Long after stabilization, onboard measurements are made over the last 2,000 feet from the S.W. Highland Road exit to the entrance of the Sylvan exit where the elevation is 730 feet above sea level. Over the last 2,000 feet, the grade undulates slightly, averaging 4.05%. Posted speed limit is 50 mph. Convenient turnarounds exist at the top and bottom of the grade.

To measure terminal speed, the gear step is noted and both the original equipment speedometer and a temporary tachometer are read. These instruments are calibrated by a local automotive speedometer service shop on wheel rollers. Figure 29 is an example of a speedometer calibration. Usually, indicated speed is about 2 mph high. Speeds below 20 mph are not easily read.

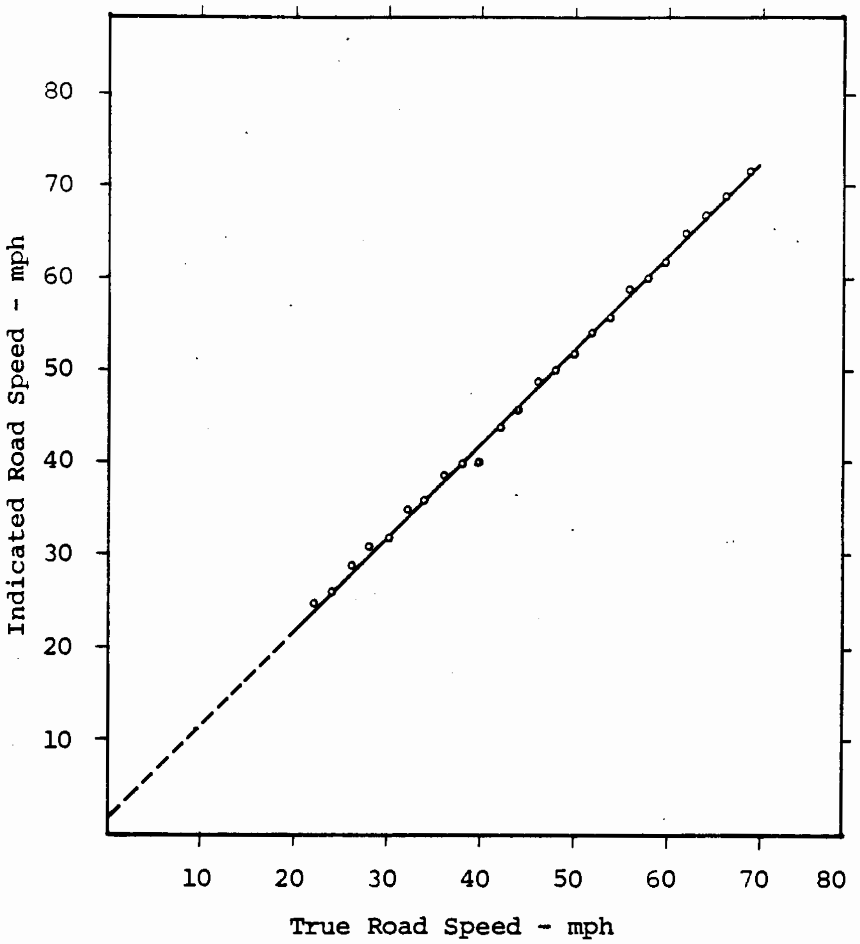


FIGURE 29.

SPEEDOMETER CALIBRATION
Bus 807
1973 GMC Model T8-H5307A
Instrument Sales & Service, May 1980

Two temporary engine tachometers are commonly available: a SAE cable type and an alternator type.

Example 1: Engler ET 338.815 head and 340.808/1/1 sender. The standard tachometer mechanical drive outlet at the engine's fuel pump is used to turn a cable leading into the sender. The sender is a permanent magnet generator that produces a DC voltage and the tachometer head is a damped voltmeter.

Example 2: Motorola Model 12AT03. Easier to install quickly, this instrument is connected to the field terminals of the alternator where the AC voltage is directly proportional to engine speed. The readout is a damped rectifying voltmeter.

Tests are normally conducted with no onboard load other than a driver, the instrument person, and a full tank of fuel. The following formulae are for validating test results and for making estimates. It is customary to allow 150 lbs per passenger.

$$RHP = GRP + ARP + RRP$$

$$RHP = eff \times BHP$$

$$GHP = \frac{\% \text{ grade} \times GVW \times MPH}{375}$$

$$ARP = \frac{.0901 \times p \times Cd \times W \times H \times MPH^3}{375\ (460 + t)}$$

$$RRP = \frac{r \times GVW \times MPH}{375}$$

$$p = 14.7\ (1 - .00000697)^{5.167}$$

Where:	Is	Units
RHP	Road horsepower.	hp
GHP	Grade resistance power.	hp
ARP	Air resistance power.	hp
RRP	Rolling resistance power.	hp
BHP	Output of bare engine.	hp
% grade	Slope of grade. Rise over run.	%
GVW	Gross vehicle weight.	lbs
MPH	Road speed.	mph
p	Barometric pressure.	psi
t	Temperature.	°F
Z	Altitude above sea level.	ft
Cd	Air drag coefficient.	dimensionless
r	Rolling resistance factor.	dimensionless
eff	Overall mechanical efficiency of bus.	%
W	Width of bus.	ft
H	Height of bus.	ft

Typical values for transit motorbuses:

Cd = .6
r = .009
eff = 73%
H = 10.3'
W = 8.5'

3.1.2 Acceleration

Since a transit bus is largely involved in a stop-and-go mission, acceleration performance is important. To measure this, a bus, with its headlights on low beam, is brought up to a stop on a flat straightaway 6 feet short of a starting mark. The fan is artificially made to turn at full speed, achieving a worst-case condition. The brakes are released and the bus is allowed to creep forward under zero throttle depression. When the starting mark is reached, the throttle pedal is fully depressed. Simultaneously, the headlights are switched to high beam as a signal to a timer downrun. Three runs are made to establish times to reach distances in steps up to 500 feet. Figure 30 is a plot of a typical data set.

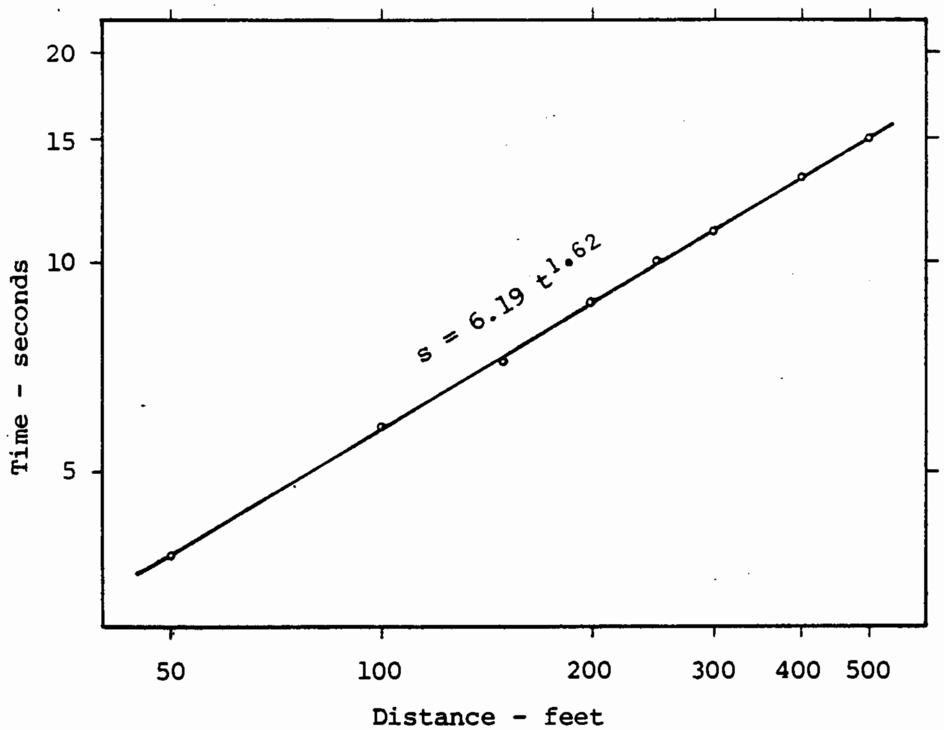


FIGURE 30.

TYPICAL ACCELERATION PERFORMANCE
Bus 357
1972 Flxible® Model 111D-D061
8V-71N Engine with 71C5 Injectors, #1 Diesel
Fan Drive On

Downtown Portland blocks measure 200' x 200'. A routine maneuver is to accelerate away from a stop at close to full throttle up to merger with city traffic moving at about 25 mph. A typical bus can reach 25 mph in 200 feet on a level. All things considered, the time to reach 200 feet is a good gauge of acceleration performance. In Tri-Met's experience:

Acceleration	200-foot Time Seconds
Typical	8½ to 9
Marginal	9 to 10
Unacceptable	over 10

Curiously, bus acceleration data always appears as a linear curve on a log-log plot, despite the complex behavior of the power train during this maneuver. The engine runs up to a stall speed and slowly increases speed until dropped back by the shift of the automatic transmission at about the 180-foot mark. The throttle delay mechanism is damping the onset of full throttle for the first 8 seconds or so. The first part of the run is spent with the torque converter in action, but with always varying slip speed. The whole maneuver defies prediction, yet there seems to be a uniform behavior.

The equation for the best-fit curve in Figure 26 is:

$$s = 6.19\ t^{1.62}$$

If the bus were behaving as if being propelled by a constant *force*, then:

$$s = \left(\frac{F\ g}{2\ Wt}\right)\ t^2$$

If the bus were behaving as if being propelled by a constant *power* (force decreases as speed increases), then:

$$s = \left(\frac{2}{3}\sqrt{\frac{2\ P\ g}{*Wt}}\right)\ t^{1.5}$$

*Includes the equivalent weight of rotary wheel inertia:

$$Wt = GVW + We$$

$$We = g\left(\frac{2\pi\ Nt}{5280}\right)^2\ (n\ It)$$

Where:	Is	Units
s	Distance	ft
t	Time	sec
F	Force	lbs
g	Gravity	32.2 ft/sec^2
Wt	Apparent vehicle weight	lbs
We	Equivalent weight of wheel inertia	lbs
GVW	Gross vehicle weight	lbs
P	Power to overcome inertia	hp
Nt	Tire size	rev/mile
It	Polar moment of inertia of a tire and rim	ft-lb-sec^2
n	Number of tires	dimensionless

Typical values for transit motorbuses:

$$n = 6$$
$$Nt = 476\ \text{rev/mile}$$
$$It = 9.3\ \text{ft-lb-sec}^2$$
$$We = 577\ \text{lbs}$$
$$GVW = 22{,}700\ \text{lbs}$$
$$Wt = 23{,}277\ \text{lbs}$$

Since the empirical exponent of time, 1.62, is closer to 1.5 than it is to 2, it appears that the bus's behavior resembles constant power acceleration more than constant force acceleration. We can go on to estimate the quantity of that power through the use of the term $\frac{2}{3}\sqrt{\frac{2\ P\ g}{Wt}}$. A value for Wt of 23,277 lbs gives 52 hp.

It is even more curious, now, to discover that a bus with an engine rated at 218 hp is only able to exert an average of 25% of that potential against the ground to overcome inertia during an acceleration maneuver. No more than another 50 hp is necessary to handle average air and rolling resistances. The rest is attributable to losses in the power train, particularly in the torque converter, and to the derating effect of the throttle delay.

3.1.3 Engine Rating

The most common way of gauging power output is by the engine manufacturer's rating obtained from test cell dynamometer studies. Table 30 lists this value, along with several other useful parameters, for the several engines used in Tri-Met's program.

TABLE 30.

[1]DETROIT DIESEL ENGINE COMPARISON
[2]2,000 rpm
Full throttle
#1 Diesel
Factory Standard Settings

	Engine	8V-71N	8V-71T	8V-71TAC	6V-92TAC
	Injectors	71C5	71C5	7A50	9B70
[6]Output	bhp	218	220	222	244
Air Flow	cfm	830	1030	975	1070
Water Flow	gpm	123	123	123	138
Heat to Coolant	BTU/min	6322	7040	8190	7320
[4]BSFC	lb/bhp-hr	.394	.402	.389	.366
[5]Back Pressure	"Hg	6	3	3	3
[5]Intake Vacuum	"H_2O	25	20	20	20
Peak Torque	ft-lbs	660	740	745	722
at Speed	rpm	1200	1200	1200	1200
Exhaust Emissions					
NO_2	ppm	1150	980	466	600
HC	ppm	153	111	232	110
CO	ppm	140	154	126	150

1 Courtesy of Detroit Diesel Allison Division.
2 Except for peak torque item.
3 All are nominally the 50 mm^3/stroke size.
4 Brake Specific Fuel Consumption.
5 Maximum allowed.
6 Standard SAE conditions: 85°F, 29.00 "Hg.

3.1.4 Fuel Economy

Fuel mileage is determined by averaging miles travelled and gallons consumed over long periods. Any impact of a noise control treatment on fuel economy would be of major importance. Turbocharging the 8V-71N engine has only a slightly beneficial effect. Table 31 compares test buses against their fleet.

TABLE 31.

FUEL MILEAGE COMPARISON

M = million

		[1]Miles	[2]Hours	[3]Gallons	mpg	[4]mph	[4]gph
[5]Fleet	FY 77-78	20.68M	1.43M	4.76M	4.34	17.0	3.92
	FY 78-79	20.09M	1.44M	4.92M	4.34	16.4	4.02
	FY 79-80	21.43M	1.53M	5.29M	4.06	16.5	4.07
	FY 80-81	21.80M	1.56M	5.20M	4.19	16.4	3.92
	4 Years	84.00M	5.96	20.17M	4.16	16.6	3.98
[6]Test Buses	[7]Bus 341	15,011	1,047	3,408	4.40	16.9	3.83
	[8]Bus 809	50,496	3,630	11,850	4.26	16.4	3.84

1 Scheduled platform miles.
2 Scheduled platform hours.
3 Fillings.
4 Running time used. 85% of platform hours.
5 70% 8V-71N, 30% 6V-71N.
6 Turbocharged engines.
7 8V-71T. Tested Mar 79 to Aug 79 (166 days).
8 8V-71TAC. Tested Aug 80 to Jul 81 (314 days).

The fuel mileage statistics for one of the test buses were grouped into 1,000 mile intervals and plotted on Figure 31. There is no persistent trend, but there is much variation.

More variation is evident from the histogram for the same bus presented by Figure 32. Individual fuel mileages range from less than 2 mpg to more than 9 mpg.

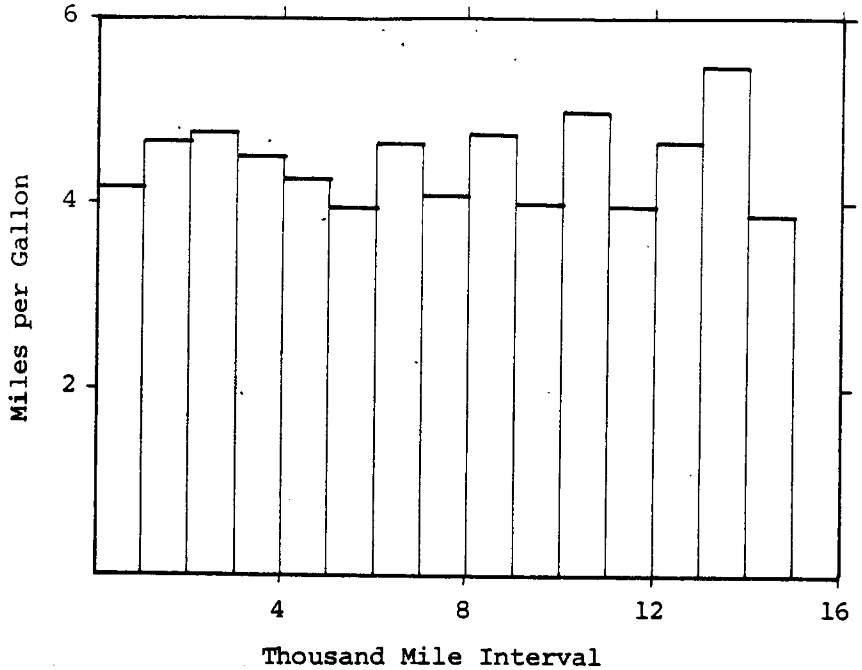


FIGURE 31.

TREND OF FUEL MILEAGE
Bus 341
1972 Flxible® Model 111D-D061
8V-71N with 71C5 Injectors, #1 Diesel
15,000 Mile Test
March to August 1979

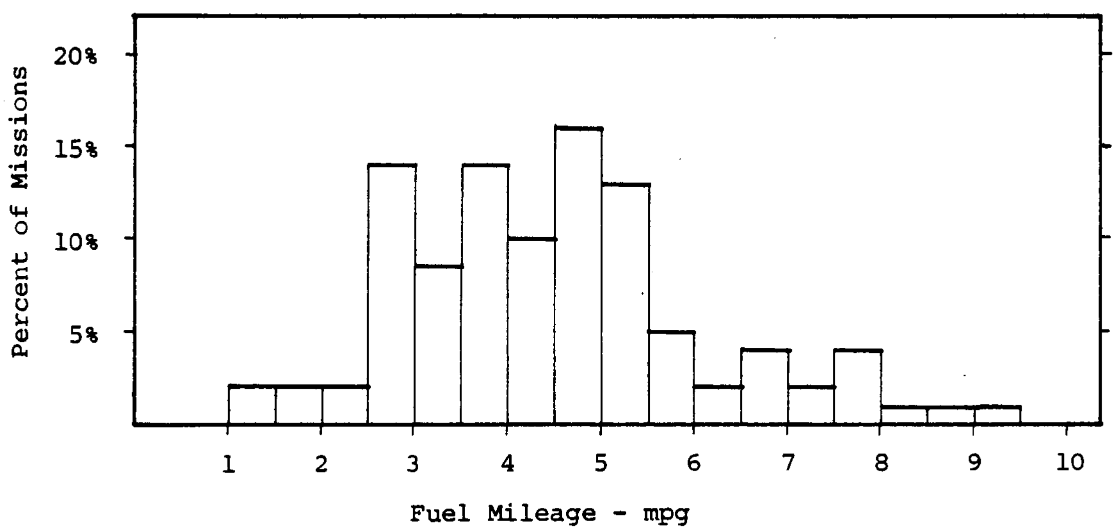


FIGURE 32.

HISTOGRAM OF FUEL MILEAGE
Bus 341
1972 Flxible® Model 111D-D061
8V-71N with 71C5 Injectors, #1 Diesel
15,000 Mile Test
100 Missions
March to August 1979

The test bus's records were searched for repeat runs. These are listed on Table 32. Repeatability of fuel consumption argues against record keeping error and driver behavior as sources of variation. Some possibilities left are inherent mission characteristics such a terrain, traffic conditions, and service intensity. More investigation into these areas is indicated if fuel economy is to be pursued.

TABLE 32.

REPEATED MISSIONS

Bus 341
1972 Flxible® Model 111D-D061
8V-71N with 71C5 Injectors, #1 Diesel

Date	Hours	Miles	Gallons	mpg	mph
3-29	8.52	154	31	4.97	18.08
5-8	8.52	154	35	4.40	18.08
6-5	1.57	25	10	2.50	15.92
6-12	1.57	25	10	2.50	19.92
6-20	14.85	217	40	5.43	14.62
6-22	14.85	217	40	5.43	14.62
3-27	14.00	204	50	4.08	14.57
4-13	14.00	204	48	4.25	14.57
4-18	14.00	204	47	4.34	14.57
[1]4-16	13.50	190	25	7.64	14.90
4-30	13.50	190	45	4.24	14.90
5-2	14.62	217	44	4.93	14.84
5-22	14.62	217	46	4.72	14.84

[1]Possibly due to having been fueled twice in one day, the second filling not being recorded.

Variation of fuel from batch to batch or supplier to supplier or even from #1 to #2 diesel is not going to affect fuel efficiency and obscure the fuel economy comparison of a test bus to its fleet. Figure 33 graphs fuel heat value as a function of density.

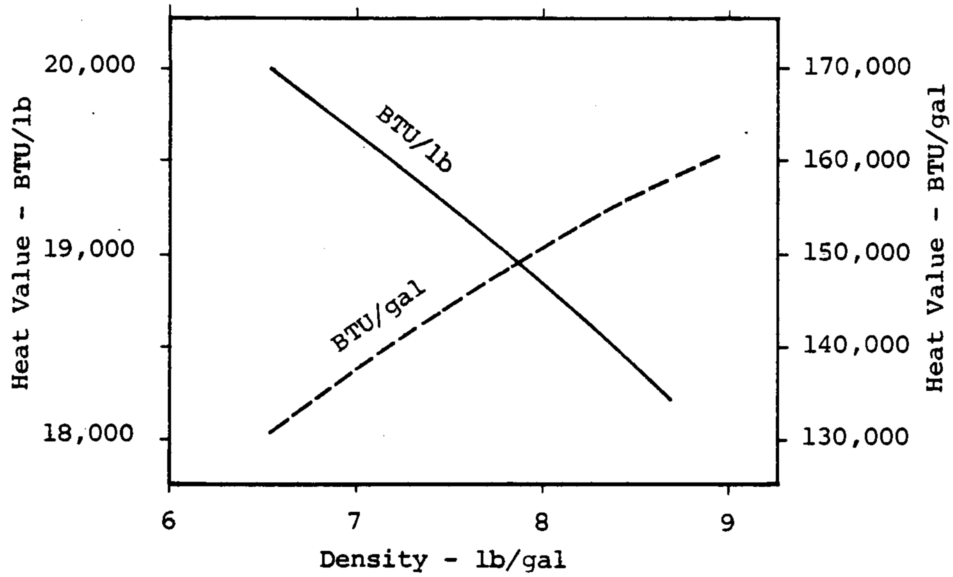


FIGURE 33.

[1]HEAT VALUE OF PETROLEUM FUELS

It is conventional to express fuel density by the API (American Petroleum Institute) scale. The relationship is:

$$\text{Degree API} = \frac{141.5}{{}^*\text{sp. gr.}} - 131.5$$

*sp. gr. is the specific gravity of the fuel at 60°F.

Specific gravity is the ratio of the mass of the liquid to the mass of an equal volume of water at a standard temperature. Thus, the density of the liquid is its specific gravity times the density of water at the standard temperature. The density of water at 60°F is 8.337 lb/gal. Therefore:

$$\text{Fuel Density} = \frac{1179.7}{\text{Degree API} + 131.5} \quad \text{lb/gal at } 60^\circ\text{F}$$

[1]Courtesy of Socony Mobil Oil Company.

Suppliers gave data on their products and a sample of Tri-Met fuel was tested by a local laboratory. Table 33 summarizes. A slight edge of heat value per gallon goes to #1 diesel over #2, but this can easily be more than lost by the price difference.

TABLE 33.

COMPARATIVE FUEL HEAT VALUES

		#1 D	#2 D
Data Source	Charlton Labs	[1]42.2	-
	Union Oil	41	33
	Mobil Oil	42	34.7
	Texaco	42.5	34.2
	Detroit Diesel	42.1	34.4
Averages	°API	42.0	34.1
	lb/gal	6.80	7.12
	BTU/lb	19,780	19,580
	BTU/gal	134,500	139,000

[1]Also reported Cetane Index of 48.

Turbocharging an engine does not change its fuel efficiency much. Table 30 lists the brake specific fuel consumption values at full output for 4 engines of interest. The naturally aspirated engine is slightly more fuel efficient than one turbocharged version and slightly less fuel efficient than two other turbocharged versions.

Comparing bsfc's at full output may be misleading since a transit bus engine spends the bulk of its time at part throttle, part speed. Considering actual fleet miles, hours, and gallons together with characteristic engine fuel efficiencies leads to this approximation: the average transit bus consumes fuel as if it is at very low output levels half the time and the other half of the time, as if it is producing 140 bhp at 1,500 rpm. Table 34 lists comparative fuel efficiencies for this part throttle condition. "Old" 8V-71N means before tradeoffs for the sake of exhaust emission reduction were taken. From this point of view, the naturally aspirated and turbocharged engines are grouped even closer.

TABLE 34.

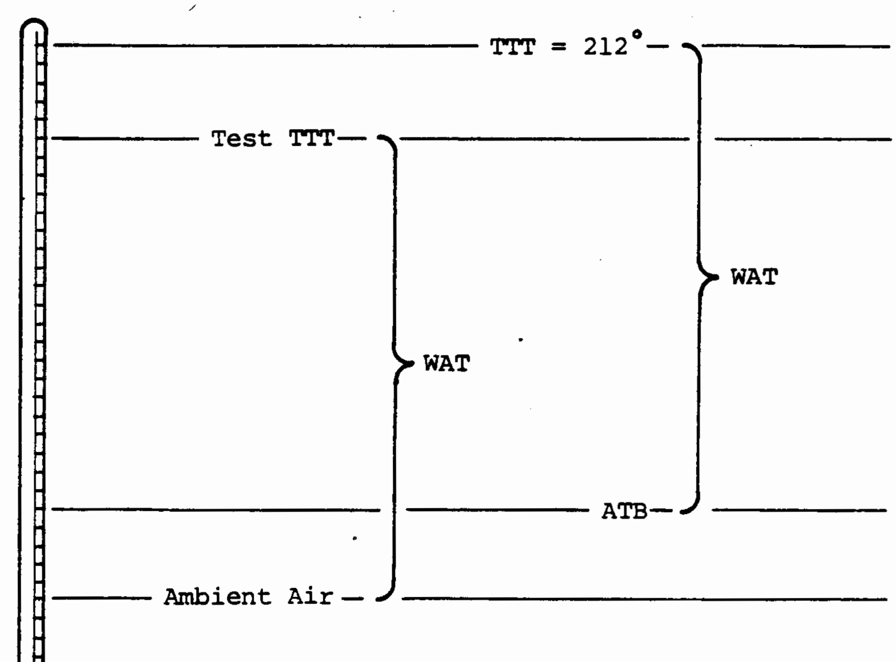
COMPARATIVE PART THROTTLE FUEL EFFICIENCIES
Brake Specific Fuel Consumption
140 bhp at 1,500 rpm
#1 Diesel
Detroit Diesel Engines

	lbs/bhp-hr	Relative Scale
Old 8V-71N	.379	1
New 8V-71N	.389	.97
8V-71TAE	.398	.95
6V-92TA	.376	1.01

3.1.5 Air-to-Boil Rating

The cooling system's job is to bring the engine's temperature quickly up to a steady preset operating level, then hold it there steadily while loads and ambients vary. It must be sized to handle the worst of conditions with capacity to spare. Deterioration due to fouling, damage, and age must be allowed for. A long standing benchmark for gauging a cooling system's capacity is the "ATB" rating. ATB stands for air-to-boil. Figure 34 illustrates the concept.

The hottest place in the cooling system is the radiator top tank. It is receiving the coolant being discharged by the engine. If a bus is tested under full power with the cooling system operating at full strength, the top tank temperature will stabilize at a certain level depending on the ambient temperature. One wonders how high the ambient can go without the top tank exceeding some limit. The ATB rating uses the 212 °F boiling point of water as the limit and it approximates that if the ambient air temperature should rise, the top tank temperature would rise by the same amount. For example, if the test TTT were 150 °F in a 50 °F ambient, the ATB rating would be 112 °F. The bus could boil over on an 112 °F day.



ATB = 212 °F - WAT

ATB = 212 °F + Ambient Air Temp - Test TTT

Where:

TTT = Top tank temperature
WAT = Water-to-air temperature
ATB = Air-to-boil temperature

FIGURE 34.

AIR-TO-BOIL CONCEPT

Tri-Met uses S.W. Canyon Road to exercise the cooling system in a realistic manner. The bus is prepared by fully engaging the fan drive, cleaning the radiator core, and blocking the by-pass thermostats open. The 4% grade is climbed at full throttle. After stabilization, engine speed, ambient air temperature, and top tank temperature are noted. The ATB rating is derived as above.

The engine will produce more waste heat at governed speed than it does at the test speed, but it is not necessary to insist on a cooling system test at governed speed. Speeding up the engine also speeds up the fan, and this increase in the cooling system's capacity nearly compensates for the added heat load.

3.1.6 Fan Air Delivery

Maintenance people probably worry more about the deleterious effect of noise control treatment on the cooling system's air flow than about any other single side effect. Tri-Met has developed a means of checking for troubles here.

An 8-inch deep 32" x 32" wooden frame is attached over the radiator grille. The margin is taped so that all air passing through the radiator core must pass through the frame. Wires 4" apart are strung, forming a lattice of 64 four-inch squares. An anemometer, such as the Taylor Model 3132 4" diameter propeller-type, is held at each square for 10 seconds. There will be variation across the grid with higher-than-average air speeds in the upper half. The bus is parked with the fan drive fully engaged. The engine is idled at such a steady speed as to cause the fan to turn at close to 1,500 rpm. Fan speed is measured by some means such as the Pioneer Photo-Tach Model 1030 photo-electric tachometer. Air speed readings are averaged and then multiplied by the area of the frame to give volumetric air flow. This is corrected to the standard speed of 1,500 rpm by multiplying with the ratio of 1,500 to the test rpm. This correction conforms to the "fan laws" and is verified by the experiment exhibited by Figure 35 in which fan air flow was measured at three different fan speeds.

Tri-Met's tests roughly indicate that a noise control treatment (such as a bellypan) will not significantly affect air flow unless it changes the "bottleneck". Air flows through the cooling system as if it were essentially an incompressible fluid passing through a series of doors, expanding about 15% as it is warmed by the radiator core. The doorways are the radiator decorative grille, the radiator core, the hole in the fan shroud, and the exits from the engine compartment. The least doorway, the bottleneck, should be the fan shroud hole. If the area of any other doorway becomes less than the shroud hole, the fan's air delivery will be less than it could be. If any other doorway, already more open than the shroud hole, is made larger still, it will do no good.

Table 35 lists the results of an experiment where the radiator grille was varied. In the baseline condition, the grille was the bottleneck. Its area was 25% less than the shroud hole. The grille was then remodeled, giving it 45% more area than before. The shroud became the bottleneck. Bottleneck area increased 25% and air flow increased 15%. Removing the grille altogether still left the shroud as the bottleneck. The air flow increased a modest 5%.

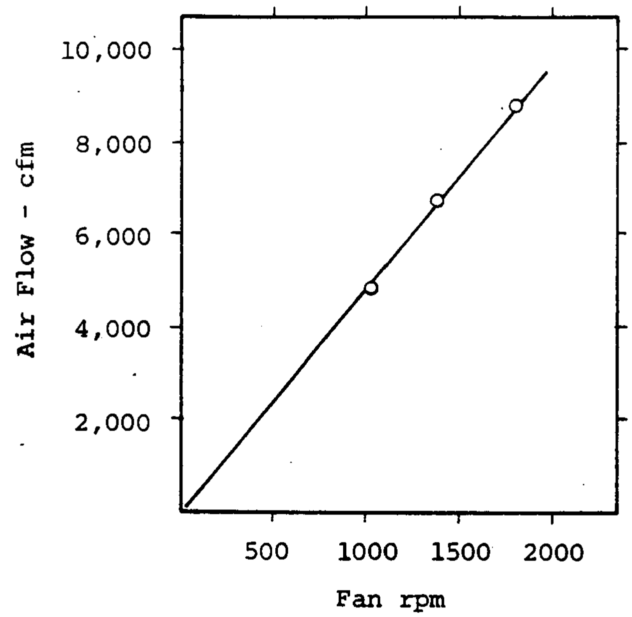


FIGURE 35.

FAN AIR FLOW EXPERIMENT
Bus 357
1972 Flxible® Model 111D-D061
28" Dia, 8-bladed Fan
Shrouded with 1" Tip Clearance
Core Cleanliness As-Is
Center Street Yard, November 1978

TABLE 35.

RADIATOR GRILLE EXPERIMENT
Bus 341
1972 Flxible® Model 111D-D061
Fan 1,500 rpm
Bus parked, Engine Idling
October 1979

		Baseline	New Grille	No Grille
Net Area in^2	Grille	[1]490	710	∞
	Core	722	722	722
	[2]Shroud	615	[1]615	[1]615
	ft^3/min	10,471	12,054	12,680

[1]Bottleneck
[2]Actual net open area is 707 in^2. This has been reduced 15% for fair comparison to other doorways upstream of core.

3.1.7 Exhaust Back Pressure

Exhaust back pressure is measured by tapping into the exhaust pipe a short distance downstream from the manifold outlet on the neutral axis of any curvature, away from any region of turbulence. Readings are taken by a suitable pressure gage, such as a Dwyer Magnahelic Model 2080 and 2150, as the engine is exercised at full throttle. Tri-Met uses the 4% grade on S.W. Canyon Road. The benchmark is in terms of inches of mercury at 2,000 rpm. It is seldom possible to establish stabilization at 2,000 rpm. Accordingly, readings are taken at various lesser engine speeds and the value at 2,000 rpm is extrapolated from their trend.

The Detroit Diesel factory standard is a 6 "Hg limit for naturally aspirated engines and 3 "Hg limit for the more sensitive turbocharged engines.

Tri-Met has found that when back pressure is plotted against engine speed on a log-log graph, a linear curve forms. This indicates that back pressure is proportional to engine speed raised by a certain exponent. Figure 36 displays typical results. The exponent of engine speed seems to be very close to 1.5 for naturally aspirated engines. The exponent is close to 2.0 for their turbocharged counterparts.

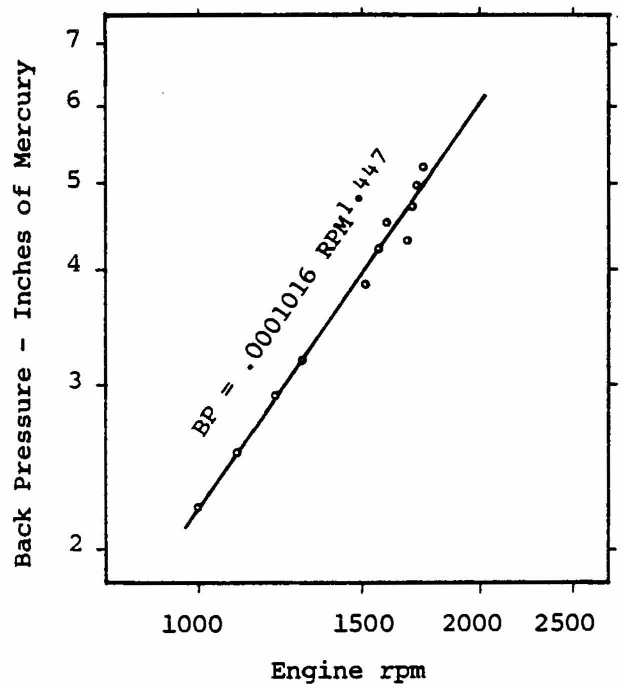


FIGURE 36.

TYPICAL EXHAUST BACK PRESSURE
Bus 807
1973 GMC Model T8-H307A
8V-71N with 71C5 Injectors, #1 Diesel
OEM Muffler with 4" Exhaust Pipes
Full Throttle up S.W. Canyon Road
May 1980

3.1.8 Intake Restriction

Intake restriction is measured by tapping into the air inlet piping anywhere between the intake manifold and the clean side of the air cleaner on the neutral axis of any curvature, away from any turbulence. Readings are taken by a suitable pressure gage, such as a Dwyer Magnahelic Model 2030. Naturally aspirated engines may be measured under no-load idle conditions. The benchmark, inches of water at 2,000 rpm, may be read directly. Figure 37 is a typical plot. But with turbocharged engines, intake air flow is load-dependent and intake vacuum must be determined, like back pressure, by extrapolation from trends at lower speeds while exercising the engine at full throttle.

The Detroit Diesel factory standard is a 25 "H_2O limit for naturally aspirated engines and 20 "H_2O for turbocharged engines. These limits tell when to change the air cleaner filter element. The difference between the intake restriction with a clean element and the limit for the engine is an indication of the service interval, but a relatively high clean-element level is not necessarily bad. It depends also on the rate of vacuum buildup with time. For example, a pre-cleaner device will add restriction initially, but by rejecting a large amount of dirt before it gets to the filter element, the pre-cleaner might well extend the filter's service interval.

As with back pressure patterns, Tri-Met has found that intake vacuum appears to be proportional to engine speed raised to a certain exponent. This exponent is close to 2.0 for naturally aspirated engines and close to 3.0 for their turbocharged cousins.

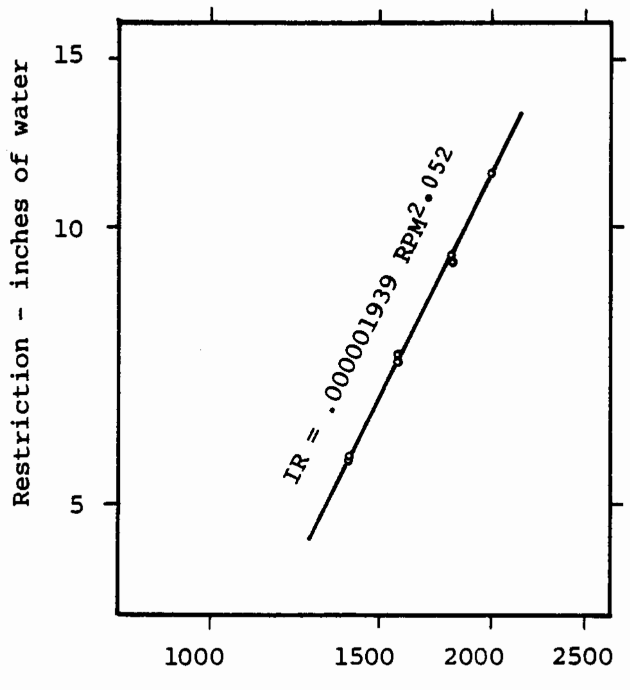


FIGURE 37.

TYPICAL INTAKE VACUUM
Bus 807
1973 GMC Model T8-H307A
8V-71N Engine
Donaldson EBA 13-0026 Air Cleaner
Donaldson PVH 000885 Pre-separator
5½" Dia Inlet Tubing
May 1980

3.1.9 Exhaust Temperature

Although it is customary to discipline exhaust restriction by controlling exhaust back pressure, it is appropriate to include exhaust temperature with a battery of performance tests. Readings are taken from a pyrometer (such as an ISSPRO R602-14) inserted into the exhaust stream a short distance downstream from the engine's outlet. This is downstream from the turbocharger, if there is one. The engine is exercised at various speeds under full throttle. The rating is stated in degrees Fahrenheit at 2,000 rpm, corrected to 85 °F ambient. The correction is made by finding the difference between ambient and 85 °F and adding this to the raw datum.

Figure 38 displays typical results from testing a naturally aspirated 2-stroke cycle Detroit Diesel engine and its turbocharged counterpart. Because the exhaust gas stream gives up energy to the turbocharger, it cools. One can see that the exhaust from the turbocharger is about 250 °F cooler, to the benefit of the exhaust piping and muffler.

Because of the engine's large thermal inertia, a long time is necessary to achieve stabilized exhaust temperature. This tends to scatter repeat run data. Generally, full throttle exhaust temperature is independent of engine speed.

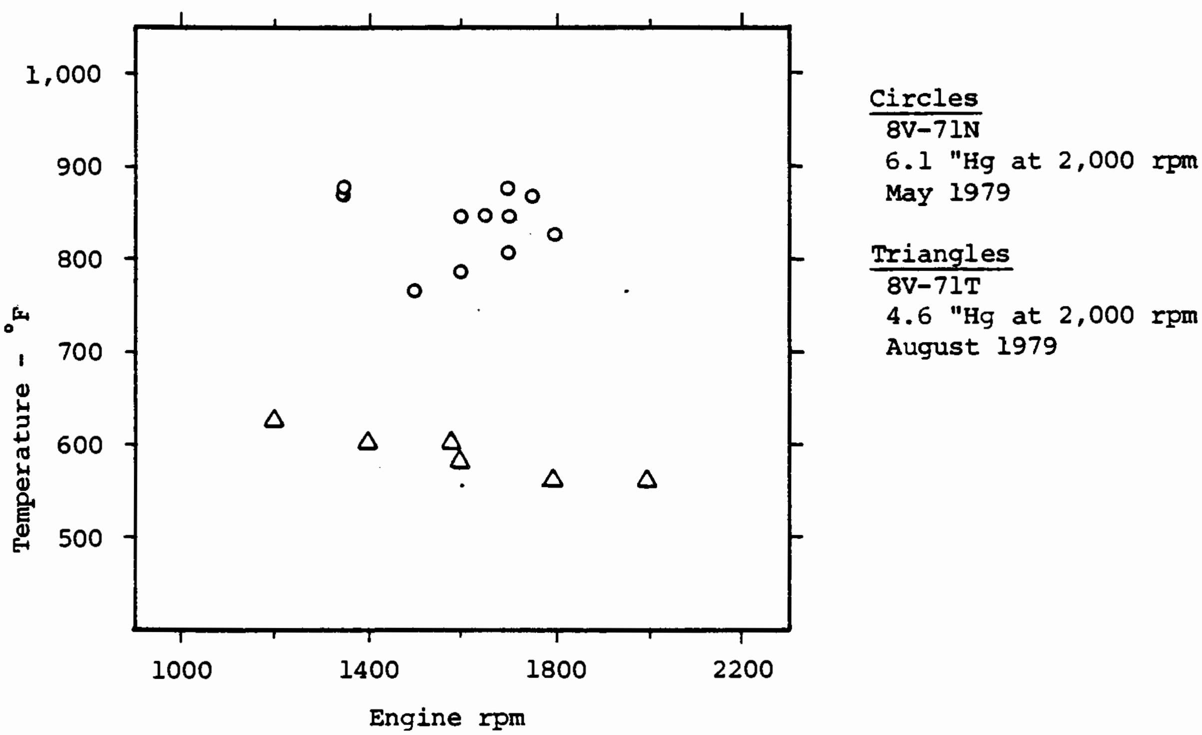


FIGURE 38.

TYPICAL EXHAUST PIPE TEMPERATURES
Full Throttle
71C5 Injectors, #1 Diesel
Corrected to 85 °F Ambient

3.2 THE COOLING SYSTEM

Often there is a reluctance to embark upon bus noise treatment programs because of a deepseated concern for overheating problems. For instance, it is sometimes feared that acoustically absorptive engine compartment lining will suffocate cooling system air flow...The same goes for bellypans. Accordingly, a discussion of various fundamental aspects of the typical bus cooling system is given here in hopes of providing an improved basis for making noise treatment judgments.

The engine cooling system's job is to bring the engine quickly up to an optimum temperature and then keep it there despite changes of load and environment. Every automotive cooling system employs a fan...a potent source of noise. The proper functioning of the cooling system is vital to the engine's safety. Motorbus noise control efforts seem to collide with the cooling system sooner or later, either because of trying to stop excessive fan noise or because of concern for overheating.

The typical cooling system uses a water solution to remove heat from the engine and take it to the "radiator", a water-to-air heat exchanger, where the heat is discharged to the ambient air. A fan insures air flow through the radiator. Controls regulate the water flow through the radiator and the speed of the fan.

3.2.1 Radiator Core

Coolant enters the radiator top tank where it is deaireated, then falls down a bank of tubes to the bottom tank where it is gathered and returned to the engine. A dense array of fins draws the coolant's heat from the tubes and delivers it to the air flowing through. Typical radiator core measurements are:

Height	30 1/2"	Frontal Area	7.28 ft^2
Width	34 3/8"	Number of Tubes	222
Thickness	3 3/8"	Fins per Inch	11
Weight	112 lbs.		

Fin and tube details are:

	Stock thickness	Material	Features
Fins	.010"	Copper	Dimpled
Tubes	.003"	Brass	1/8" x 3/4" Oval

Tube arrangement:

4 rows. Staggered.
Rows ≈ 13/16" apart.
Tubes ≈ 5/8" apart in each row.

Figure 39 is an empirical log-log plot of the heat transfer capabilities of our typical radiator core. The equations for the family of linear curves are:

4-row core: $K = .838\ m^{.626}$
5-row core: $K = 1.279\ m^{.601}$
6-row core: $K = 1.915\ m^{.566}$

where:

K = Heat transfer coefficient - BTU/min/ft^2/°F*
m = Specific mass flow - lbs/min/ft^2

* °F is the difference between the average water temperature and the average air temperature. These are assumed to be half way between the in and out temperatures in each case.

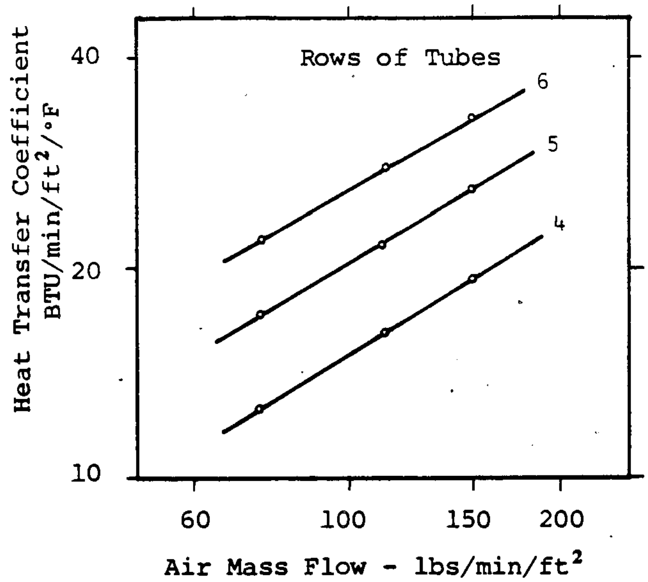


FIGURE 39.

HEAT TRANSFER COEFFICIENTS
OF TYPICAL RADIATOR CORES

Adding rows of tubes also adds resistance to air flow as the log-log plot in Figure 40 shows. The equations for the family of empirical linear curves are:

4-row core: $SP = .000377\ m^{1.562}$
5-row core: $SP = .000405\ m^{1.591}$
6-row core: $SP = .000435\ m^{1.619}$

where:

SP = Static pressure across core - "H_2O
m = Specific mass flow - lbs/min/ft^2

Note that adding another row of tubes to a 4-row core and keeping the same static pressure drop would have the same beneficial effect as increasing fan speed by 50%...and would avoid the extra 10 dBA of fan noise.

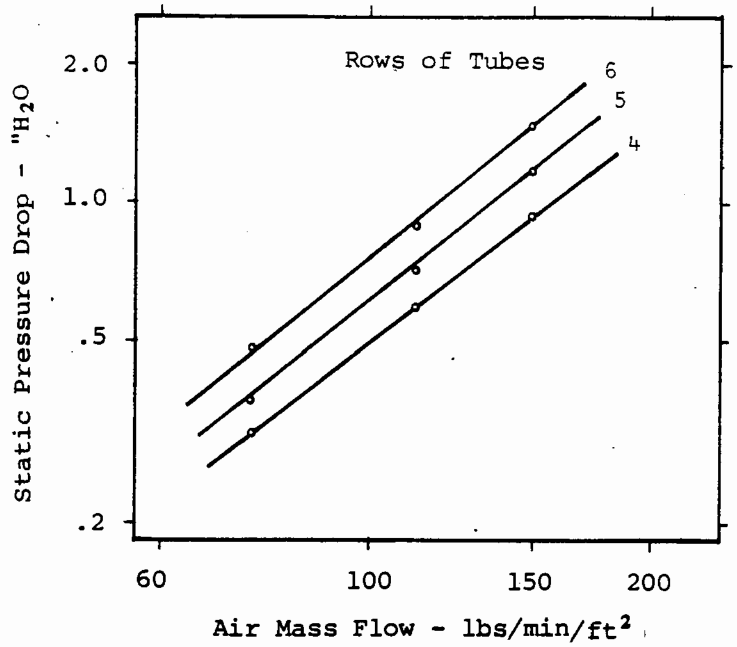


FIGURE 40.

AIR FLOW RESISTANCE
OF TYPICAL RADIATOR CORES
Standard Air, .075 lb/ft^3

3.2.2 The Coolant

It is common practise to mix anti-freeze with water in order to lower the coolant's freezing point to a safe level. Most anti-freeze is ethylene glycol blended with various additives to inhibit corrosion and foaming. A mixture to give a -20 °F freezing point is typical. Table 36 gives this coolant's properties, comparting them to those of pure water. Notice that with the normal 7 psi radiator pressure-relief cap, the coolant's boiling point is elevated well over the 212 °F for water at sea level.

TABLE 36.

PROPERTIES OF TYPICAL COOLANT

	[1]Anti-freeze Mixture	Pure Water
Freezing Point - °F	-20	32
[2]Boiling Point - °F at seal level at 12,000 ft.	 258 238	 232 217
Specific Heat at 200 °F	.90	1.02
Density at 200 °F lb/gal	8.80	8.10

[1]45% mixture by weight of ethylene glycol with water.
[2]with ordinary 7 psi radiator cap.

3.2.3 Ambient Air

Air pressure varies with altitude by the following approximation:

$$p = 14.7\ (1 - .0000069 \times Alt)^{5.167}$$

where:

p = Barometric pressure - psi.
Alt = Elevation above sea level - ft.

Air density depends on its temperature and pressure.

$$\rho = \frac{144\ p}{53.5\ (t + 460)}$$

where:

p = Barometric pressure - psi.
t = Ambient air temperature - °F.
ρ = Air density - lbs/ft^3.

Air's enthalpy depends on its temperature and the amount of water vapor it holds.

$$h = .24t + \frac{wV\ (.4065t + 1065)}{7000}$$

where:

h = Enthalpy of air - BTU/lb dry air.
t = Air temperature - °F
*wV = Specific humidity - grains of water per pound of dry air.

*Can be found from the Psycrometric Chart if relative humidity or wet & dry bulb temperatures are known.

3.2.4 Fan Laws

The efficiencies of geometrically similar axial flow fans are nearly the same over a wide range of sizes and speeds. Useful approximations are:

$$\frac{D^3N}{CFM} = Ca \qquad \frac{D^2N^2\rho}{TP} = Cb \qquad \frac{D^5N^3\rho}{HP} = Cc$$

$$TP = SP + VP \qquad VP = \rho\left(\frac{V}{1096}\right)^2$$

where:

D = Fan profile diameter - ft.
N = Fan rotation - rpm.
CFM = Fan delivery - ft^3/min.
V = Air speed - ft/min.
ρ = Air density - lb/ft^3

HP = Fan input power - hp
TP = Total fan pressure - "H_2O
SP = Fan static pressure - "H_2O
VP = Fan velocity pressure - "H_2O
Ca, Cb, Cc = Constants

3.2.5 Heat Balances

These heat flows are equal and interrelated:

The waste heat given up by the engine to the coolant.
The heat given up by the coolant to the core.
The heat transferred by the core from the coolant to the air.
The heat absorbed by the air.

$$Q_{engine} = Q_{coolant} = Q_{core} = Q_{air}$$

→ $$Q_{coolant} = GPM \times d \times cp\ (TTT - BTT)$$

$$GPM_2 = GPM_1 \left(\frac{RPM_2}{RPM_1}\right)$$

→ $$Q_{core} = K \times A \times \Delta Tc$$

$$K = C_1 \left(\frac{M}{A}\right)^a$$

$$\Delta Tc = \tfrac{1}{2}\ (TTT + BTT - t_2 - t)$$

$$m = \frac{M}{A} \quad cfm = \frac{CFM}{A}$$

→ $$Q_{air} = M\ (h_2 - h) \approx .245M\ (t_2 - t)$$

$$h = .24t + \frac{wV \times hg}{7000} \approx .24t + \frac{wV\ (.4065t + 1065)}{7000} \approx .245t$$

where:

Q = Heat flow - BTU/min.
GPM = Coolant flow - gal/min.
d = Coolant density - lb/gal.
cp = Coolant specific heat - BTU/lb.
TTT = Top tank temperature - °F.
BTT = Bottom tank temperature - °F.
RPM = Engine rotation - rpm.
K = Core's heat transfer coefficient - BTU/min/ft^2/°F.
A = Core's frontal area - ft^2.
m = Specific air mass flow - lb/min/ft^2.
cfm = Specific air flow through core and fan - ft^3/min.
M = Air mass flow - lb/min.
CFM = Air flow - ft^3/min.
C_1, a = Constants characterizing the core.
ΔTc = Core's temperature difference - °F.
t_2 = Air temperature as it leaves core - °F.
t = Air temperature entering the core, the ambient - °F.
h_2 = Enthalpy of air leaving core - BTU/lb.
h = Enthalpy of air entering core, the ambient - BTU/lb.
wV = Specific humidity - grains of water per pound of dry air.
hg = Enthalpy of water vapor - BTU/lb.

The flow of air through the core stabilizes where the encouragement of the fan and the bus's movement over the road equals the resistance to flow of the core and the rest of the air path. Where the core's resistance is paramount:

→ $SPc + VPc + SPb = TPf + TPb$

$$SPc = C_2 \ (m)^b = C_2 \left(\frac{M}{A}\right)^b$$

$$M = \rho \times CFM = \frac{\rho_2 D^3 N}{Ca}$$

$$TPf = SPf + VPf$$

$$VPf \simeq VPc = \rho \left(\frac{V}{1096}\right)^2 \qquad V = \frac{CFM}{A}$$

$$TPf = \frac{D^2 N^2 \rho_2}{Cb}$$

$$\rho_2 = \rho \left(\frac{t + 460}{t_2 + 460}\right)$$

$$TPb = C_3 \times \rho \ (MPH)^2$$

$$SPb = C_4 \times M^2$$

where:

- SPc = Static pressure drop across core - "H_2O.
- VPc = Velocity pressure through core - "H_2O.
- TPf = Total pressure head of fan - "H_2O.
- TPb = Total pressure head produced by bus's movement - "H_2O.
- SPf = Static pressure rise across fan - "H_2O.
- VPf = Velocity pressure produced by fan - "H_2O.
- C_2, b = Constants characterizing core.
- M = Air mass flow - lb/min.
- A = Core's frontal area - ft^2.
- ρ = Ambient air density - lb/ft^3.
- V = Velocity of air entering core - ft/min.
- CFM = Air flow entering core - ft^3/min.
- D = Profile diameter of fan - ft.
- N = Fan rotation - rpm.
- RPM = Engine rotation - rpm.
- C_3, C_4 = Constant characterizing bus and its core air path.
- Ca, Cb = Constants characterizing fan family.
- ρ_2 = Density of air leaving fan - lb/ft^3.
- t = Ambient air temperature - °F.
- t_2 = Temperature of air leaving core - °F.
- MPH = Bus's speed over the road - mph.
- m = Specific air mass flow - $lb/min/ft^2$.
- SPb = Static pressure drop across the grille, etc. - "H_2O

Typical values are:

$d = 8.80$ lb/gal at 200 °F.
$cp = .90$ BTU/lb at 200 °F.
$GPM = 123$ gal/min at 2,000 rpm for 8V-71 engines.
$A = 7.28\ ft^2$ (clean).
$a = .626$
$b = 1.562$
$C_1 = .838$
$C_2 = .000377$
$C_3 = .0008035$
$Ca = 1.803$ with 490 in^2 open area grille & 1" tip clearance.
$Cb = 964,800$ with 1" tip clearance.
$C_4 = .0000004211$ with 490 in^2 open area grille.

3.2.6 Thermal Controls

The typical bus has two thermal controls: the bypass thermostat and the fan drive thermostat. Together, they must keep the coolant at an optimum temperature without turning the fan unnecessarily fast, causing it to make more noise, rob more power, and consume more fuel than it needs to. At 50% speed, the fan makes 18 dBA less noise and requires only one-eighth the input horsepower than at 100% speed.

On a vee engine, there are two identical bypass thermostats, one for each side. They actuate a valve that shunts coolant past the radiator, allowing the engine to warm up quickly and to keep from running too cold. The valve has a modulating action. Its range from full open to full close is 18°. A small wax-filled chamber is immersed in the coolant stream leaving the engine for the top tank. When the coolant reaches a certain temperature, the wax begins to melt and expand, diverting the valve. The valve is adjusted to begin opening during warmup at the rated temperature. Commonly available are 170° and 180° thermostats. Figure 41 plots stroke vs. temperature for the 170° model ordinarily used by Tri-Met. Hysteresis is on the order of 10°.

A similar wax control is used to modulate fan speed for the GMC fan drive. This valve is commonly called a "Vernatherm". Its orginal manufacturer no longer exists, but the trade name lives on. Figure 42 plots stroke vs. temperature for the model used by Tri-Met. Two Vernatherm ratings are available: 160° and 172°. The Vernatherm is inserted into the coolant stream leaving the bottom tank for the engine. At full load, full performance, the bottom tank's temperature is about 12° less than the top tank's. When the bottom tank reaches the rated temperature, the Vernatherm begins to choke off oil flowing out of the drive's torus, causing it to fill more and turn the fan faster. Figure 43 shows how the fan speeds up as the tank warms.

Opinions differ on the ideal top tank temperature. No one is especially worried about boiling the coolant. Lubricating oil breakdown is the prime consideration. Tri-Met has had very little trouble with older buses whose top tanks range between 175° and 185°. Some say a common target is 190°, but that in no case should TTT exceed 210°. Tri-Met has encountered premature automatic transmission failures on buses whose top tanks tested in the 186° to 205° range. If the top tank runs too cool, complaints about inadequate heating for passengers arise. Also feared are intake port ash deposits and incomplete combustion.

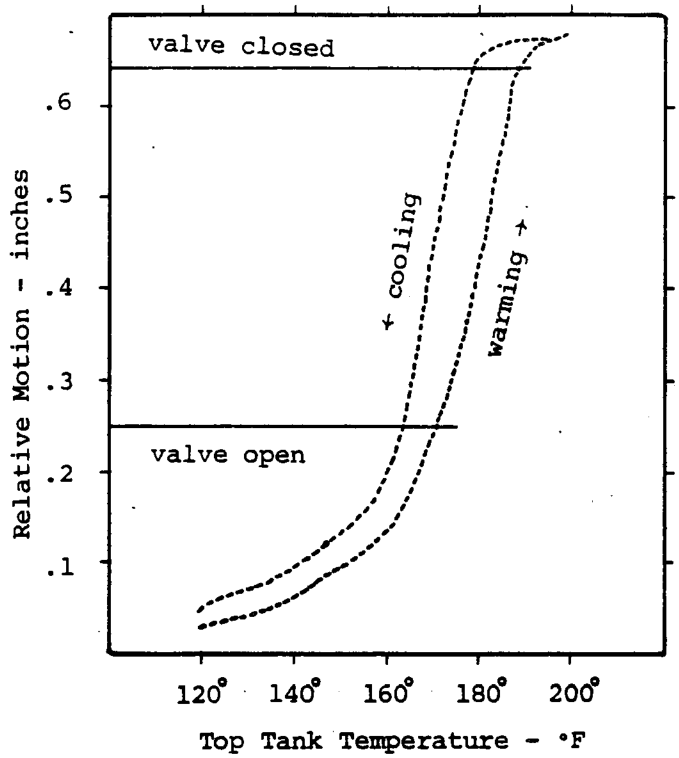


FIGURE 41.

BYPASS THERMOSTAT THERMAL RESPONSE
Elac 170BB

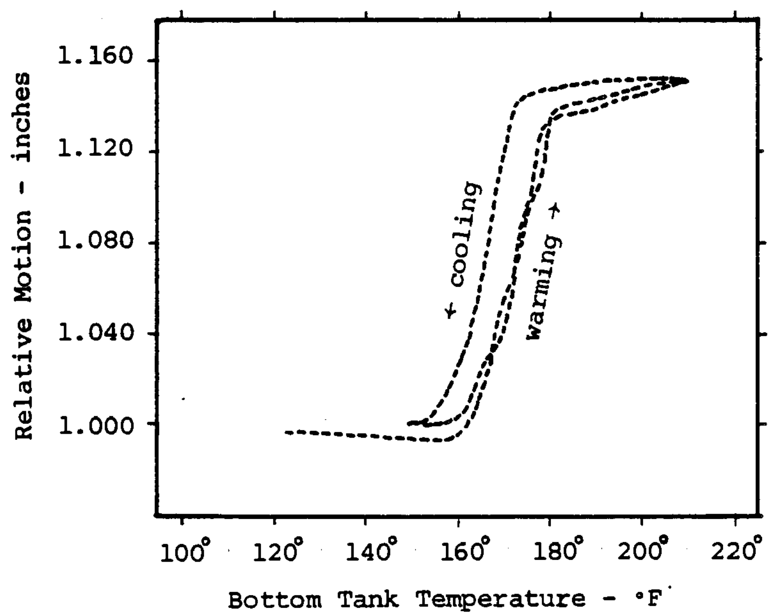


FIGURE 42.

GMC FAN CONTROL THERMAL RESPONSE
Vernatherm 160 D79

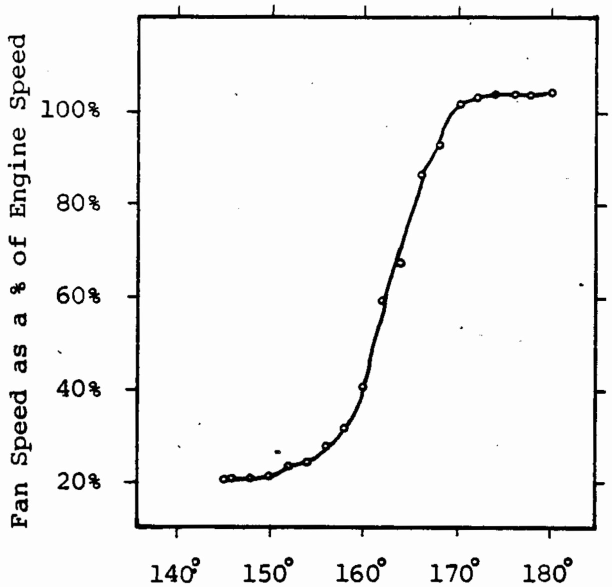


FIGURE 43.

MODULATED FAN SPEED OF GMC DRIVE
Vernatherm 160 B78
Bus 807
1973 GMC Model T8-H5307A
S.E. Powell Yard
September 1981

On most of Tri-Met's "New Look" buses, the combination is 170° bypass thermostats with a 160° Vernatherm. This is a safe, but not perfect, match as Table 37 shows. Fan speed is 60% when the bypass valve is only 10% open.

A 172° Vernatherm was successfully field tested in combination with 170° bypass thermostats. Usual top tank temperatures rose only 5°, but the fan speed did not build up until nearly all the coolant flow was through the radiator.

TABLE 37.

THERMAL CONTROL COORDINATION
170 °F Bypass Thermostats
160 °F Vernatherm

	TTT	BTT	Bypass	Fan
Warmup Sequence	<170	<TTT	Closed	20% on
	170	TTT	Cracked open	20% on
	173	158	10% open	60% on
	180	168	50% open	95% on
	184	172	75% open	100% on
	188	176	100% open	100% on

3.2.7 Fan On-Time

Before allocating resources to the noise reduction of fans with thermally-automatic drives, one should have an idea of how much time the fan is actually turning fast enough to cause a problem. Appropriate instrumentation is not feasible for recording modulating speed drives such as the GMC Torus or the Schwitzer Viscous types, but Tri-Met's group of 1976 AMG buses affords an alternative. On these buses, the fan is driven at a 1:1.02 speed ratio by the engine through an on-off magnetic clutch. All that is necessary is to connect an electric clock in the fan circuit and another in the engine systems circuit. Periodically, the two clock readings are recorded. By dividing engine time into fan time, percent fan-on time for the intervening period is determined.

Data collected over a 12-month service interval and compared against average daily temperatures in Portland provided by the National Weather Service is displayed in Figure 44. Over the year's experience, the fan was on 16% of the time. The best-fit empirical relationship is:

$$\% \text{ fan-on time} = (.8132 \times \text{ave}°F) - 28.7$$

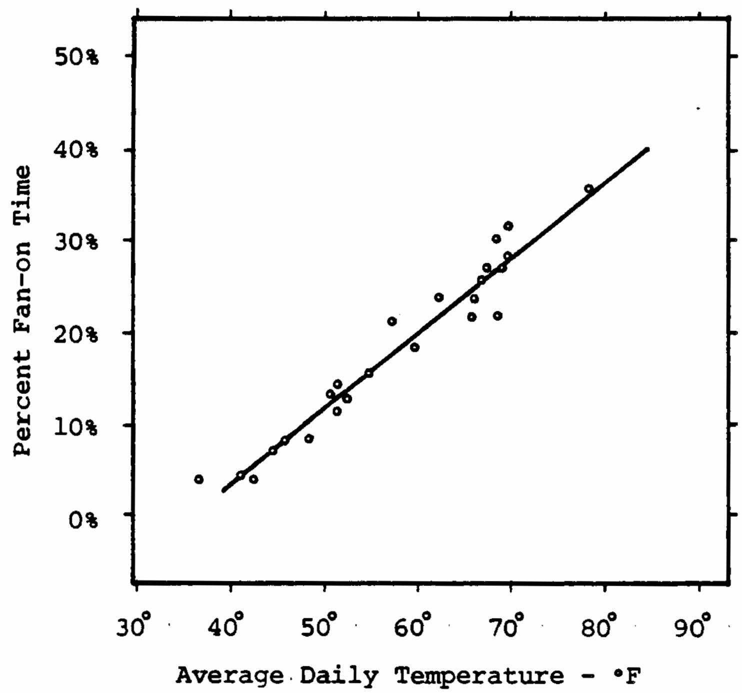


FIGURE 44.

FAN-ON TIME
Bus 1030
1976 AMG Model 10240B-8
8V-71N Engine
Facet Model 4907-15 Clutch
February 1979 to February 1980
2,771 Engine Hours
444 Fan Hours

A peculiarity of the 1976 AMG buses is that the air conditioning system condenser is mounted in front of the cooling system radiator so that the fan supplies air flow for both. Either a coolant thermostat switch or an air conditioning system control switch energizes the fan clutch. No abrupt change in fan-on time occurred when the air conditioning system was activated in May, nor when the radiator was thoroughly cleaned in April.

On hot days when maximum temperatures reached into the range of the 90's and over 100 °F, average temperatures were about 15° less. During cooler weather, the gap between maximum and average narrowed. When the maximums were in the 40° to 50° range, the averages were only 5° less.

APPENDIX A.

A manual prepared for the
Urban Mass Transportation Administration
by Michael C. Kaye, consultant to the
Tri-County Metropolitan Service District
of Portland, Oregon.

December, 1981

MAKING BUS NOISE TESTS

This article provides the person in the field with a practical recipe for measuring transit motorbus noise. Instrumentation, test method and data analysis are described in ways understandable to the technologically oriented layman. Results can be compared to studies done by others as well as be used to pursue in-house programs of noise control.

INTRODUCTION

The official EPA noise rating method* is a little hard to follow because it has to be what it is...an all-inclusive regulation for new vehicle manufacturers. This manual helps the uninitiated by explaining in detail how to do the test for an ordinary rear engine diesel motorbus with an automatically-shifted transmission and torque converter.

See Figure 45. The general idea is to measure the peak noise as the bus accelerates past a microphone 50 feet to one side of the lane center.

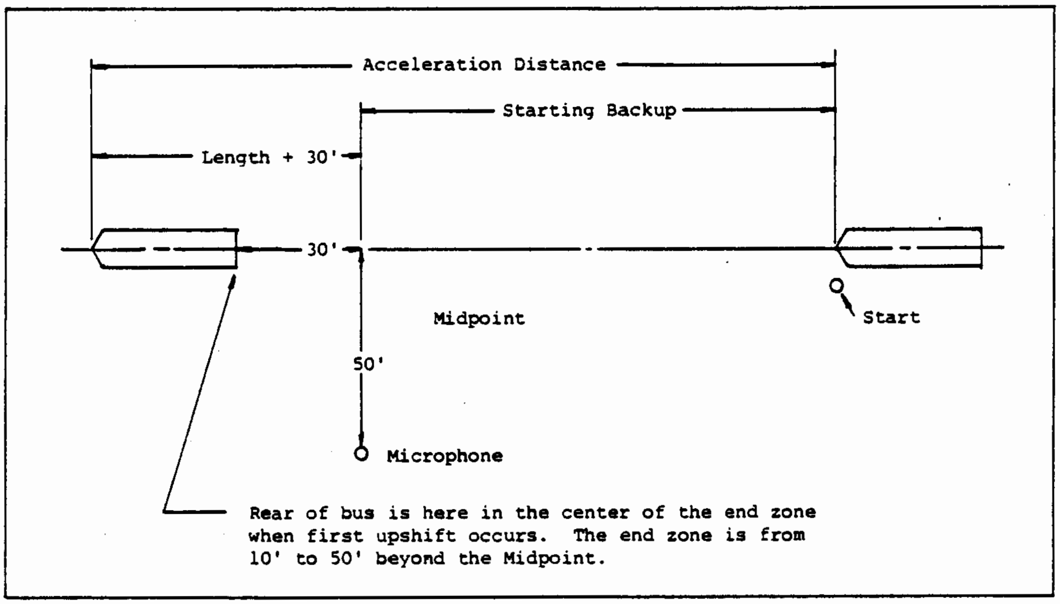


FIGURE 45.

GEOMETRY OF TEST TRACK

*Proposed Noise Emission Standards for Transportation Equipment - Buses, U.S. Environmental Protection Agency, 40 CFR Part 205, Docket ONAC 77-6, Federal Register, Vol. 42, No. 176 - Monday, September 12, 1977.

SITE

Look for these things when picking a site. Airport aprons and drag strips are usually good bets.

1. At least a 400-foot straightaway.
2. Turnarounds at each end.
3. No sound reflecting objects within 200 feet of where the microphone will be for the exterior test.
4. Dead flat.
5. Asphalt concrete pavement.*
6. Sparse traffic.
7. Low background noise.
8. Clean of loose gravel, etc.

*Ratings on porous asphalt can be as much as 2 dBA less than on smooth cement. Porous asphalt is favored because it is easier to find and more representative of the real world.

INSTRUMENTS

Listed below is one set of adequate instruments. There are many equivalents.

1.	Bruel & Kjaer 2218	ANSI Type 1 Sound Level Meter
2.	Bruel & Kjaer 4165	½" condenser microphone
3.	Bruel & Kjaer 2619	Microphone Preamplifier and 10 M cord
4.	Bruel & Kjaer UA 0196	Flexible extension rod
5.	Bruel & Kjaer 4230	1,000 Hz Acoustic Calibrator
6.	Bruel & Kjaer UA 0237	Windscreen
7.	Rolatape MM 45	2-wheel distance meter
8.	Two red plastic 12" pylons	Start marks
9.	Tripod	Microphone support
10.	Chalk	Pavement marker
11.	Benjamin 132	.22 cal air pistol
12.	Carpenter's powered chalk	Capsule filling
13.	5 mm pill capsules	Gelatin chalk pellets
14.	Cotton	Wadding
15.	Motorola 12AT03 and 50' cable	Alternator type engine tachometer
16.	Pioneer Photo-Tach 1030	Wireless tachometer
17.	Thermometer	Ambient air temperature
18.	Clipboard	Data recording

PREPARATIONS

1. Automatic Fan Control.

   If the bus has an automatic fan clutch, install overriding valves, switches or mechanisms. Rehearse how to turn the fan on or off or leave it in normal mode (in the case of a modulating-speed type of clutch) when out on the track.

2. Onboard Speedometer.

   Calibrate the indicated road speed with true road speed at a local automotive instrument shop having wheel rollers.

3. Onboard Tachometer.

   In case there is no engine tachometer installed, place a temporary one where the driver can see it. The Motorola alternator-connected instrument (from the instrument list on page 2) is calibrated by determining true engine speed with a Pioneer Photo-Tach.

4. Fuel.

   Since fuel can affect noise, contact the supplier and find out the °API and the cetane number. Normally, #1D is 42 °API and #2D is 34 °API.

5. Exterior Microphone.

   Establish a travelled-lane centerline on the track. Locate the microphone 50 feet to one side with only pavement between. No grass or dirt. The microphone is held by the extension rod which is clamped to the tripod. Adjust the microphone 4 feet above the ground in a vertical position. Put the windscreen on the microphone. It will always be there when noise is measured, outside or inside the bus, wind or no wind.

6. Sound Level Meter.

   Before coming to the track, make sure the batteries are all right. At the track, connect the sound level meter to the microphone with the extension cord and move the meter as far as the cord will allow away from the track centerline, placing it on something like a small table. Set it on the A-weighting network and on "fast" response.

SETTING START MARKS

Use the air pistol to set the starting marks. Moisten the lid of the gelatin pill capsule to keep it from falling apart too soon. The cotton wadding follows the pill into the chamber to keep it from falling out. The pellet will leave a chalk spot on the pavement that washes harmlessly away.

Set out pylons temporarily about 150 feet up and down the track from the Midpoint and about 10 feet from the travelled-lane centerline on the driver's side looking toward the Midpoint. Tell the driver to do the following:

1. Warm up the engine. Turn the air conditioner off. Turn all lights off. Place fan drive in normal operation.
2. Load the air pistol with a chalk pellet and wadding. Give it 6 pumps, and put it on safety.
3. Hold the pistol out the window in the left hand, arm extended, aiming straight down.
4. Position the bus so that it is headed toward the Midpoint and the pistol is even with the pylon. Hold the brake pedal down. Let the engine idle.
5. Safety off. Release the brake pedal. Fully depress the throttle pedal and keep it depressed until a short time after first upshift.
6. Steer straight down the track, straddling the centerline.
7. Shoot the pistol when first upshift occurs.
8. Come to a safe stop and return to pylon. (Usually just backing up.)
9. Reload pistol.
10. Repeat run twice more in same direction.

Locate the middle of the three pellet shot marks and measure back to the pylon. This is the acceleration distance for that direction. Subtract the length of the bus plus 30 feet. This is the "starting backup". Adjust the pylon so that it is this far from the Midpoint. When it comes time to make the noise rating runs, the bus will start here and the rear end will be 30 feet beyond the microphone's perpendicular when first upshift occurs.

It might be that the track isn't quite level, and if this is the case, have the driver go to the other pylon and repeat the procedure in the opposite direction. Ask the driver to note both the engine rpm and the road speed at upshift. Record these, the acceleration distances, and the ambient air temperature.

EXTERIOR TESTS

When all is ready, turn the fan off and have the driver stand by with the front bumper even with the first pylon, engine warm. Tell him to accelerate past the microphone until first upshift occurs, just like he did when the acceleration distances were measured.

At the last moment, check these things:

1. The track is clear of people.
2. No traffic coming.
3. Background noise level is less than 60 dBA.
4. No oncoming noise such as an airplane flyover.
5. Wind is less than 10 mph.
6. Track is dry.

When ready, hold up a number of fingers to tell the driver which run this will be, and then wave him on.

Watch the sound level meter readout and note the peak value as the bus goes by. Read to the nearest ½ dBA and record, noting which side of the bus was nearer the microphone. Throw out the reading and do the run over if some spurious noise event causes a false peak, such as the exhausting of the compressed air system dryer.

After the first five runs, have the driver do another five runs in the opposite direction. Then, retest the bus with the fan on. If the bus has a modulating-speed fan drive, test the bus a third time with the fan drive in "normal". Calibrate the sound level meter before each set of ten runs.

To extract a rating from a group of five runs, average the highest two readings. Obtain a rating for the fan on, off, and normal conditions, for right and left sides. Up to six ratings in all. They all mean something. To declare the EPA rating, take the highest one.

It would be simpler to just test for the EPA rating, but one wouldn't learn as much about the bus. The EPA rating is worst-case oriented. Supposedly, the logic is that if the loudest noise a bus makes is controlled, then the noise it normally makes would be reduced. Since this is not necessarily so, keep track of what happens to its normal noise as well as its loudest noise when trying to make it quieter. A useful exterior rating, called the "operational" rating, averages the right side noise and the left side noise with the fan in normal mode.

INTERIOR TESTS

To do the interior tests, move the sound level meter, microphone and tripod into the bus. The microphone has two positions: rear seat and driver's seat. For the rear seat position, the tripod is set up on the bus centerline as close to the rear seat as the legs will allow, but no closer than 20 inches from any wall. At both places, place the microphone 48 inches above the floor and tilt it 20° to 30° from the vertical. For the driver's seat position, set up the tripod opposite the driver's ear.

Have the driver turn on all inside fans and accelerate through first upshift just as he did for the exterior tests. He doesn't have to begin at the starting pylons. One place on the track is as good as another as long as the driver stays away from outside noise sources or large noise reflecting objects.

Note the peak reading for five runs at the front and at the rear of the bus. Average the highest two of each group of five readings to get each rating. The EPA interior rating is the higher of the two ratings. The "operational" interior rating is the average of the front seat and rear seat levels with the fan operating normally.

FINAL TIPS

When there is a 2 dBA or more spread in the five readings for a single test setup, suspect something has gone wrong. Keep testing until you know what is going on.

In Europe, the microphone is placed 7.5 M from the track centerline. This is about 25 feet, half the distance used in the U.S. The European rating will be about 6 dBA more than the U.S. rating.

A perspective prepared for the
Urban Mass Transportation Administration
by Michael C. Kaye, consultant to the
Tri-County Metropolitan Service District
of Portland, Oregon.

November, 1981

Making the Transit Motorbus Quieter

It *is possible* to make the ordinary diesel motorbus really quiet ... down almost to the level of a trolleybus. Possible, yes, but not easy. It requires complete enclosure of the engine ... a radical alteration; not yet seen in this country. Even so, a number of things *can* be done to bring down a bus's noise. They range from fixing exhaust pipe leaks to developing an operationally acceptable engine compartment bellypan and turbocharging the engine.

Despite knowing the possibilities, the person with the responsibility for reducing the noise from a fleet of buses cannot expect much help from the vehicle manufacturers. Marketplace demand for lower bus noise has not supported the necessarily higher new vehicle prices. Neither has general societal concern materialized into a broad public policy that would require exceptional noise control by the manufacturers. Primarily, this is simply because buses ordinarily move in a stream of traffic having many other members as loud or even louder. If all the buses were suddenly silenced, the overall noise level would not be reduced significantly.

Usually, we consider going out of our way to do something about the problem only where there is a buses-only situation such as a transit mall or a turnaround in a quiet neighborhood...or there is pressure being exerted by a special interest group striving for improved urban environment.

The state of the art is reflected in Figure 46.

STEP	RANGE - dBA	BUDGET PER BUS
Before fixes	84 - 87	$500
Ordinary	78 - 83	Baseline
Bellypan	75 - 77	$2,000
Bellypan & Turbocharging	71½ - 73½	$13,000
Engine Encapsulation	66 - 68	$30,000

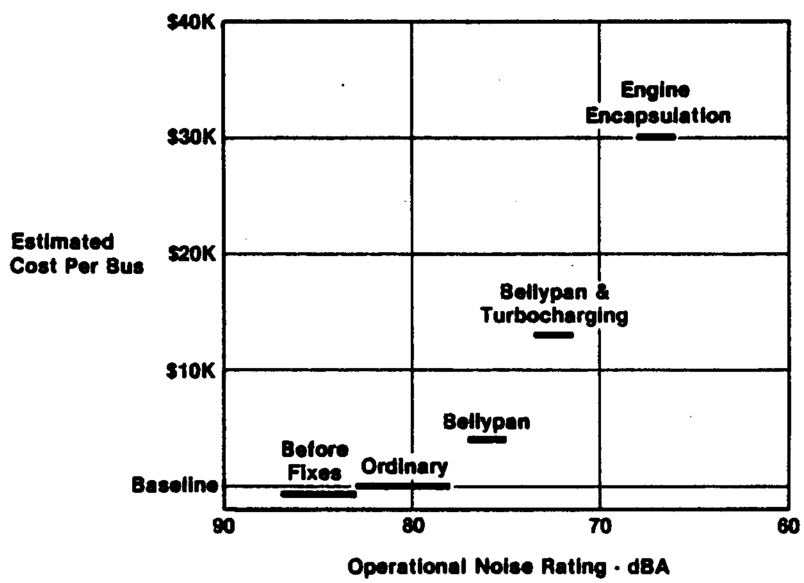


FIGURE 46.

STEPS OF TRANSIT MOTORBUS NOISE TREATMENT

For this article, "operational" noise ratings are used instead of the EPA or SAE "worst case" ratings. The operational rating is preferred for evaluating alternatives because it is more representative of very real and valuable noise reductions under ordinary circumstances favoring curbside pedestrians and most passengers. This rating is the average of the peak sounds from the left and right sides at first transmission upshift during a 50-foot acceleration passby with the radiator fan drive normal. The EPA rating is peak left side sound (always the louder) with the fan artificially at full speed. Similarly, the operational interior rating averages peak sound at the rear and at the front of the bus, fan normal. The EPA interior rating considers only sound at the rear of the bus, fan fully on.

Estimated costs, in mid-1981 dollars, are those faced by typical bus operators with rehearsed workmen, at volume levels below quantity discounts, and after amortization of engineering and special tooling.

Acoustical aspects of a large scale noise reduction campaign can be handled by one specialist with no more than $5,000 invested in instrumentation. That person will establish a standard test procedure, and with it, systematically sort buses and monitor improvements.

STEP 1: "FIXES"

Any fleet probably has a group of buses that are unnecessarily loud. This is due to exhaust pipe leaks (it is *essential* to retrofit flexible tube sections), disfunctional thermostatic fan drives, and excessive body rattles and poor door fits. Figure 47 illustrates a few of the Step 1 possibilities. A bus can be readily retrofitted with a modulated speed fan drive such as a self-contained viscous clutch or the common torus drive. Care should be taken to coordinate the thermal reaction of the fan drive and radiator coolant bypass controls in order to minimize fan speed. For the same reason, radiator core air sides should be kept clean.

The annoying hiss from the air brake system dryer can be eliminated by running an exhaust tube to the engine compartment. Engine and exhaust noise can be reduced by derating with 50 mm injectors. The transmission can be adjusted to perform the first upshift at the lowest road speed (keeps the engine speed as low as possible). Although not always blameless, exhaust mufflers are often accused when the real culprits are exhaust leaks, outright engine noise, or high-speed fan noise.

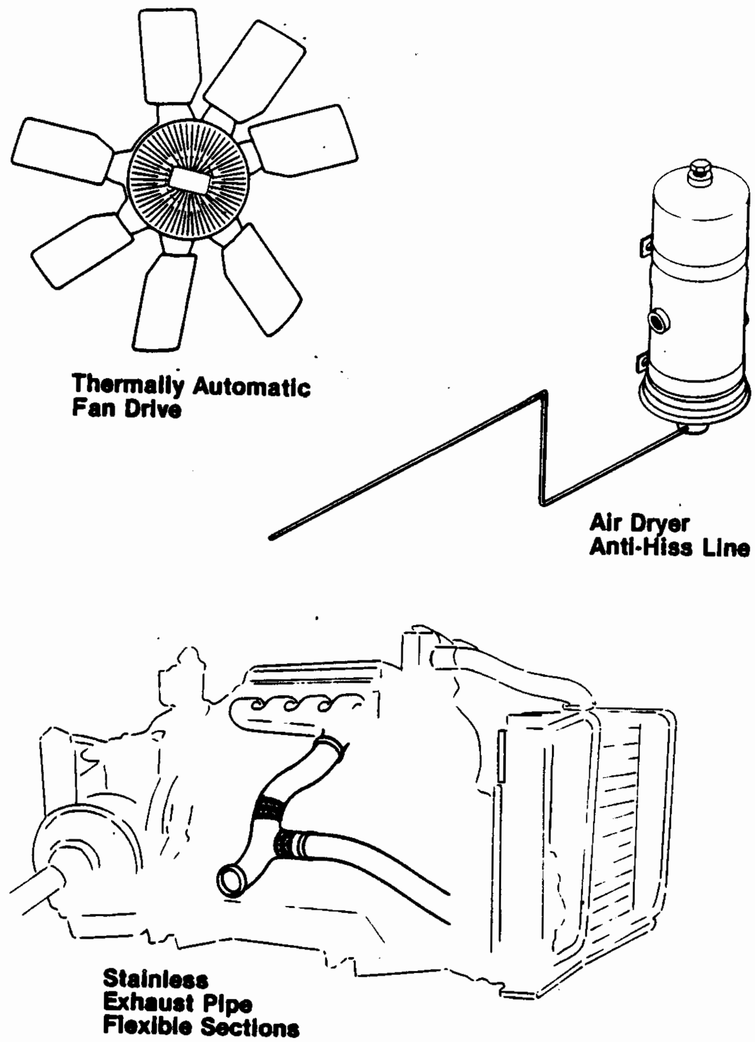
For the sake of interior noise, the three rubber cushions mounting the engine and the two engine cradle cushions should be replaced if deteriorated. The rear seat access doors to the engine compartment sould be refitted and sealed as necessary.

STEP 2: "BELLYPAN"

Once the readily available fixes are applied, the main noise sources will be the engine and the fan. Exhaust noise from the outlet and shell radiation from the pipes and muffler body are of much less importance. Noise from the air intake, from the transmission, and from the tires and chassis (at downtown speeds) are not important at all. A bellypan is the next step having the most benefit for the least cost. It would stop sound from reaching either side of the bus by way of reflection off the pavement beneath the engine. If properly designed and fitted, a bellypan would keep most of the road dirt out of the engine compartment.

FIGURE 47.

SOME FIXES



Unfortunately, generally acceptable bellypans have yet to be built for the bulk of the nation's buses. Multi-piece pans of louvered sheet metal have long been available, but are either not specified or are discarded after use. They obstruct maintenance accessibility, are easily and often deformed by road bumps and are then difficult to dismount and remount. They collect offensive deposits of oily dirt. Unfairly blamed for overheating problems, they do reduce surplus cooling system capacity.

Figure 48 conceptualizes an acceptable and accoustically effective bellypan. A 1 psf plastic molding, it resists permanent deformation and is inexpensive to replace. Because of inevitable oil drippings, there is no soft anti-noise lining. Baffles deflect debris thrown back by the rear wheels. Cooling system air flow escapes forward through existing large apertures in the firewall. More frequent thorough cleaning of the radiator is required.

Lining the rear door of the engine compartment with a combination sound absorptive and sound barrier padding is a proper accompaniment to the bellypan. Neither requires the engine to be removed. Infrequent repairs to door padding are necessary.

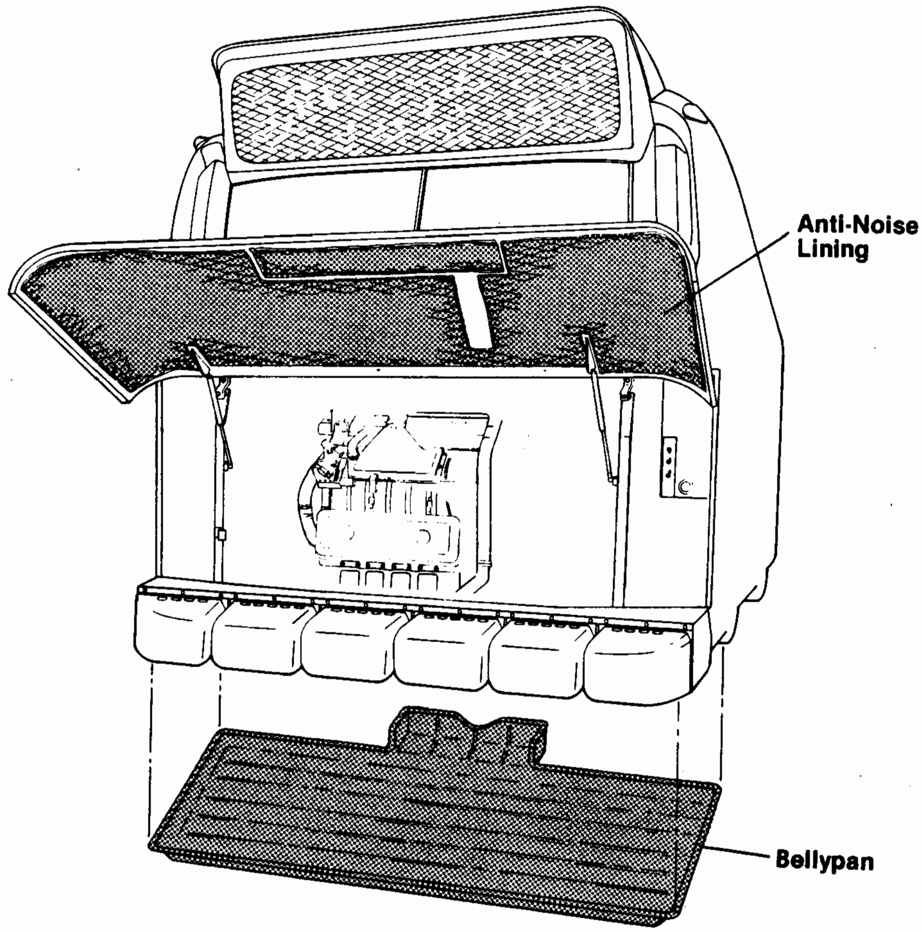


FIGURE 48.

THE BELLYPAN STEP

STEP 3: "TURBOCHARGING"

Retrofitting both the turbocharger and the turbocharged engine into the "New Look" style bus have been demonstrated. This treatment is intended to be done when the engine requires overhaul. As Figure 49 implies, whole new exhaust and intake systems are entailed. A new, more compact jacketed muffler with simpler piping takes the place of the original and is situated between the engine and fan. The air intake features a snorkle taking in cleaner air. This partly makes up for the shortened filter service interval resulting from the pre-cleaner having been displaced. While the engine is removed, its compartment is lined with anti-noise padding. Minor relocation of engine compartment components is widespread.

Turbocharging reduces engine noise by softening the cylinder pressure rise at the moment of combustion. This benefits noise received inside the bus and on both sides outside the bus. The newer Detroit Diesel 6V-92TA engine is 2 dBA quieter than the 8V-71TA. Unmuffled exhaust noise is reduced by the turbocharger, but there is more sensitivity to air flow restriction. Even though the exhaust source remains the easiest to deal with, the need for sophisticated muffler design is unalleviated. The padded muffler blocks some of the engine noise escaping through the radiator opening and prevents a good deal of fan noise echoing.

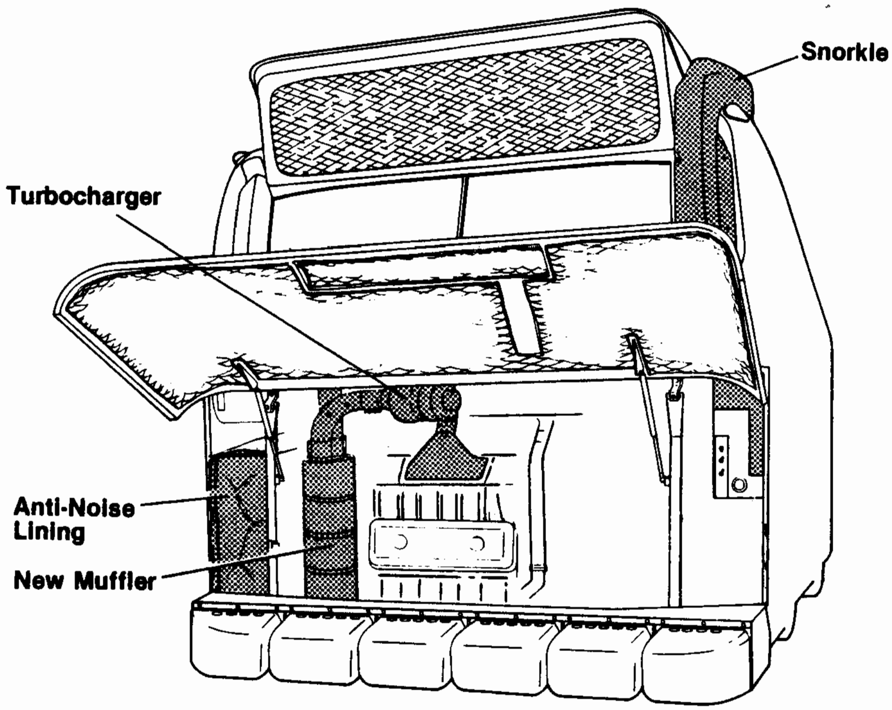


FIGURE 49.

THE TURBOCHARGING STEP

Turbocharging improves high altitude performance, gives slightly better fuel economy, and results in much cleaner exhaust. It may be necessary to substitute one size larger fuel injectors for restoration of satisfactory acceleration performance.

The turbocharging step without a bellypan, but with engine compartment lining, brings the operational noise rating down to the 74½ - 76½ dBA range. Adding the bellypan improves the noise rating by 3 dBA.

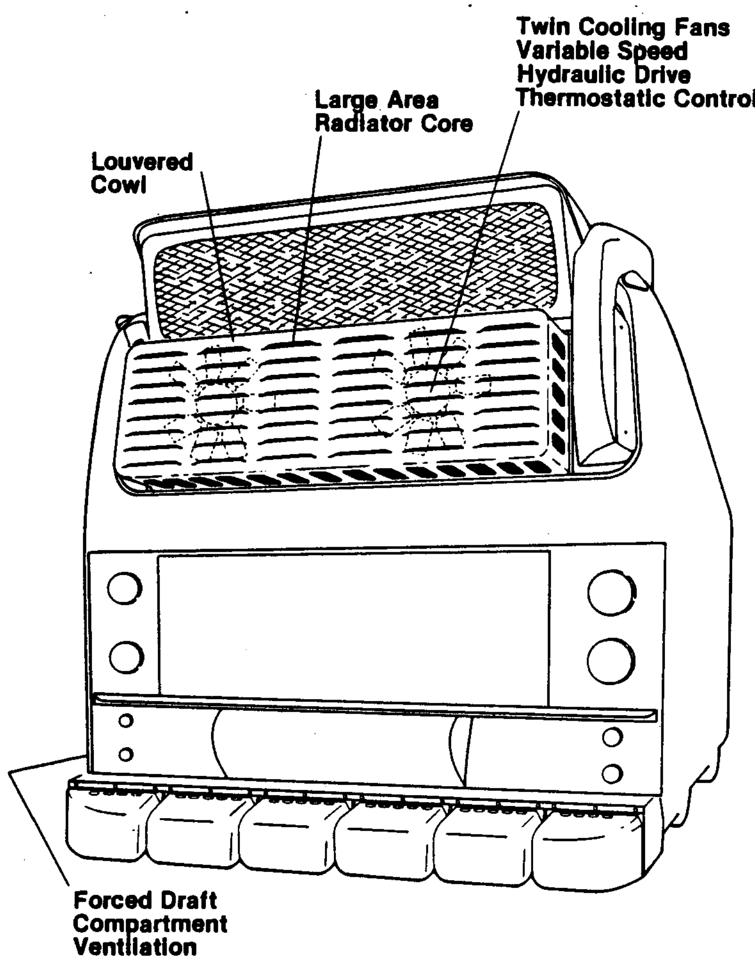
STEP 4: "ENGINE ENCAPSULATION"

For the ultimate in bus noise treatment, the engine must be completely boxed in. This means finding a new home for the cooling system. Such radical surgery has been done in Europe where mass transit has evolved further than in the U.S. General Motors has experimented with the configuration, but the idea has not found its way into production. Pity. Rear engine buses are such a natural. Reported noise levels are in the region of 70 dBA and below.

Such a "blue sky" treatment is depicted in Figure 50. The rear window is given over to the new cooling system, covered by a louvered cowl. Efficient twin hydraulic-driven fans pull air in through a large area radiator core located up and away from the usual road dirt and threat of collision damage. Thermally-automatic fan speed control insures constant temperature and low fan speed (and low noise). All components with moving parts are off-the-shelf and well proven. The rest are readily fabricated by local shops. This treatment does not have to be done by the factory.

FIGURE 50.

THE ENCAPSULATION STEP



With the fan and radiator out of the way, the engine compartment can be completed... closed against dirt getting in and noise getting out. Rubber boots seal the drive shaft passageways. There is room for a larger, more effective muffler to keep the exhaust source from overly contributing even at this low overall level. Either new turbocharged engines or older naturally aspirated engines can be accommodated without noise tradeoffs. Compartment ventilation, by means of a small automatic blower, lined ductwork, and clean inlet, carries away fumes and air warmed by the exhaust manifold and piping beyond cooling system control.

Down to this point, interior noise levels have enjoyed parallel reductions without special effort. Front seat noise is dominated by tire, chassis and wind noises. Noise at the rear seats is increasingly due to structureborne engine noise as more treatments are applied. A more effective vibration isolation engine mounting system is called for. This will feature tension springs for the rear cradle hangers and softer front cradle cushions, turning the cradle into an isolated mass. The result will be a rear seat noise level lower than at the front before treatments began.

The large budget required relegates encapsulation to few, if any, justifiable situations at the present time.

APPENDIX C.

HIGHLIGHTS 1975-1982

A number of transit bus noise control projects and programs were carried out under the supervision or with the cooperation of Tri-Met during the 7-year period from 1975 to 1982. The highlights of this work are listed here chronologically, organized by:

- Eningeering Activities
- Formal Reports and Other Documents
- Formal Presentations

Practically all this work was directly or indirectly funded by the Urban Mass Transportation Administration. Asterisks denote the items especially funded under the Grant Contract OR-06-0005.

ENGINEERING ACTIVITIES

Jun - Sep 1975	BUS NOISE ABATEMENT STUDY Development of noise rating procedures for transit motorbuses. Ratings and source contributions of 4 representative buses. Two-speed engine governor. Canyon effect. Acceleration performance. Driver discretion.
Oct - Dec 1975	NOISE CONTROL OF S.E. POWELL SUBSTATION Environmental impact of planned bus yard. Ambient sound measurements. Barrier design. Yard layout.
Mar 1976 - Aug 1978	TRANSIT MALL NOISE COMMITTEE Conferences attended by [1]Tri-Met, Oregon [2]DEQ, [3]HUD, Mayor's Staff, Portland Noise Control Officer. DEQ noise samplings. Mall noise standard. DEQ White Paper. Trolleybus impact. HUD projects. Plans for Tri-Met antinoise kit demonstration. Briefing of [4]EPA officials.
Jul 1976	BUS ROUTE NOISE Ambient levels along 4 bus routes with and without bus sound.
Jul 1977	AMG BUS NOISE RATING Rating by 3 methods of 1976 AM General with Detroit Diesel 8V-71N engine.
Aug 1977	BARRIER EVALUATION Ambient sound levels bordering S.E. Powell busyard sound barrier.
Oct - Nov 1977	RESPONSE TO NEW-BUS NOISE STANDARD PROPOSED BY EPA Tri-Met's response made in collaboration with Oregon DEQ and City of Portland.
Nov 1977	21 ADDITIONAL BUS NOISE RATINGS Noise ratings by 2 methods of more Tri-Met buses.
*Jul - Nov 1978	BUS NOISE TEST METHODOLOGY Refine external and internal bus noise rating procedures. Locate test sites. Acquire instrumentation and special apparatus. Noise source identification and contribution.
*Jul 1978 - May 1980	FLXIBLE® BUS ANTINOISE TREATMENT Design, development, installation, test, evaluation, and field test. Baseline tests.
*Aug 1978 - Jul 1979	BUS PERFORMANCE TEST METHODOLOGY Develop test procedures to rate hill climbing, acceleration, fan air flow, cooling system capability, fuel mileage, exhaust back pressure, intake restriction, turbocharger output pressure, exhaust temperature, engine speed, fan speed. Locate test sites. Acquire instrumentation and special apparatus.

*Sep 1978	METRO TURBOCHARGED BUS NOISE TEST Noise rating of 1976 AM General with Cummins VT-903 turbocharged engine.
*Nov 1978 - Feb 1980	COOLING SYSTEM BEHAVIOR Facet drive on-off time. GMC torus speed modulation. Vernatherm temperature response. Basic thermodynamics. Grille drag.
*Aug - Oct 1979	NEW GENERATION TURBOCHARGED ENGINE INSTALLATIONS Retrofit, noise rate, and field test Detroit Diesel 6V-92TA engine in AM General Bus 1024 and 6V-92TAC in Bus 1007.
*Oct 1979 - Jul 1981	GMC BUS ANTINOISE TREATMENT Design, development, installation, test, evaluation, and field test. Baseline tests.
*Aug 1981	DATALINER TESTS Tested treated GMC bus with unique new mobile noise spectrum analizer developed by Freightliner Corporation for highway truck noise studies.
Sep 1981	ENGINE DYNO ROOM NOISE Typical sound levels of Tri-Met engine dynamometer room. Proposed treatment.
Apr 1982	ENGINE DYNO ROOM TREATMENT Antinoise treatment. Engine noise studies.
Apr 1982	CROWN-IKARUS BUS NOISE RATING Rating of 1981 Ikarus articulated bus with Cummins NHHTC-290 engine.
Sep 1982	BUSYARD BARRIER TEST FOR METRO Ambient levels inside and outside Tri-Met's S.E. Powell substation sound barrier. Description of barrier and evaluation of its performance.

FORMAL REPORTS AND OTHER DOCUMENTS

Aug 1975 — TRI-MET BUS NOISE
A noise survey of the current Tri-Met fleet of diesel transit coaches and an evaluation of the dual range engine governor as a means of noise control.

Nov 1975 — NOISE CONTROL OF THE NEW TRI-MET BUS SUBSTATION
Ambient sound levels. Barrier design. Environmental impact of new busyard at S.E. 97th Avenue and Powell Boulevard.

Jul 1976 — THE EXTRA NOISE MOTORBUSES MAKE ON STREETS
Ambient sound samplings along 4 Portland busroutes. Sound levels with and without buses.

Jun 1977 — NOISE RATING AND SOURCE LEVELS OF NEW AMG BUS
Exterior and interior ratings by 2 methods of Bus 1066. Source contributions.

*Feb 1979 — KOIN-TV CHANNEL 6 DOCUMENTARY
Status report of Tri-Met bus noise treatment program by local TV studio. Appeared on "Northwest Illustrated" show.

*Sep 1980 — NOISE REDUCTION RETROFIT FOR A "NEW-LOOK" FLXIBLE® BUS
A detailed recipe for an early-level antinoise treatment complete with installation instructions, bills of material, engineering drawings, and technical discussion.

*Nov 1981 — NOISE REDUCTION RETROFIT FOR A "NEW-LOOK" GMC BUS
A detailed recipe for an early-level antinoise treatment complete with installation instructions, bills of material, engineering drawings, and technical discussion.

*Nov 1981 — MAKING THE TRANSIT MOTORBUS QUIETER
Illustrated pamphlet for the layman giving a brief and broad perspective of bus noise control.

*Dec 1981 — MAKING BUS NOISE TESTS
Article for the layman on how to obtain transit bus noise ratings by the EPA method.

Apr 1982 — ACOUSTIC EVALUATION OF DYNO ROOM
Sound levels before and after engine dynamometer room treatment. Engine load and speed noise sensitivities.

Apr 1982 — NOISE RATING OF A CROWN-IKARUS BUS
Exterior and interior EPA ratings of Bus 714.

Sep 1982 — TRI-MET'S BUSYARD NOISE BARRIER
Description and evaluation of S.E. Powell barrier [5]METRO. Sound levels in yard and around neighborhood.

*Feb 1983 — NOISE CONTROL OF THE CONTEMPORARY TRANSIT MOTORBUS
An extensive amalgamation of transit bus noise control engineering done at Tri-Met from 1975 to 1982. Covers virtually every facet of the subject.

FORMAL PRESENTATIONS

11-1-77 San Francisco, CA	EPA HEARING ON PROPOSED NEW-BUS NOISE STANDARD Tri-Met's position.
11-17-77 Portland, OR	HUD ENVIRONMENTAL CLEARANCE OFFICER'S QUARTERLY MEETING Transit Mall noise review.
*10-5-78 Portland, OR	TRI-MET/UMTA BUS NOISE WORKSHOP Conference held at onset of OR-06-0005 contract work to pool information from representatives of local and federal agencies, bus and component manufacturers, and engineering consultants.
*5-8-79 Arlington, VA	THIRD CONTRACTOR'S BRIEFING ON EPA NOISE CONTROL TECHNOLOGY PROJECTS Status report of OR-06-0005 project work.
*5-14-79 Indianapolis, IN	[6]APTA BUS TECHNOLOGY COMMITTEE MEETING Status report of OR-06-0005 project work.
*5-16-80 Monterey, CA	APTA WESTERN CONFERENCE Tri-Met's noise abatement program.
*2-16-81 Honolulu, HI	[7]TheBUS CONSULTATION Transit bus noise consultation for TheBUS staff on behalf of UMTA.
*3-23-82 Jacksonville, FL	TRANSIT BUS NOISE CONTROL LECTURE - JACKSONVILLE Delivery of Tri-Met/UMTA lecture to [8]JTA staff.
*4-21-82 Seattle, WA	TRANSIT BUS NOISE CONTROL LECTURE - SEATTLE Delivery of Tri-Met/UMTA lecture to METRO staff.

[1] Tri-County Metropolitan Service District of Oregon.

[2] Oregon Department of Environmental Quality.

[3] US Department of Housing and Urban Development.

[4] US Environmental Protection Agency.

[5] Municipality of Metropolitan Seattle.

[6] American Public Transportation Association.

[7] Honolulu City and County Department of Transportation, Bus Systems Division.

[8] Jacksonville Transportation Authority.

*Funded by OR-06-0005

ACOUSTIC TERMINOLOGY

Absorption Coefficient. The sound absorption coefficient of a surface which is exposed to a sound field is the ratio of the sound energy absorbed by the surface to the sound energy incident upon the surface. The absorption coefficient is a function of both angle of incidence and frequency.

Acoustics. The science of sound, including (a) its production, transmission, and effects and (b) the qualities that determine the value of a room or other enclosed space with respect to distinct hearing.

Ambient Sound. The all-encompassing sound associated with a given environment, being usually a composite of many sources near and far.

Decibel. A unit of level which denotes the ratio between two quantities that are proportional to power. The number of decibels corresponding to the ratio of two amounts of power is 10 times the logarithm to the base 10 of this ratio.

Echo. A wave which has been reflected or otherwise returned with sufficient magnitude and delay to be perceived as a wave distinct from that directly transmitted.

Equivalent Sound Level. The level of a constant sound which, in a given situation and time period, has the same sound energy as does a time-varying sound. Technically, equivalent sound level is the level of the time-weighted, mean square, A-weighted sound pressure. The time interval over which the measurement is taken should always be specified.

Free Field. A field in a homogeneous, isotropic medium free from boundaries. In practise, it is a field in which the effects of the boundaries are negligible over the region of interest.

Frequency. The frequency of a function periodic in time is the reciprocal of the period. The unit is the cycle per unit of time, such as cycles per second.

Hertz. A unit of frequency; same as cycle per second.

Intensity. The sound intensity measured in a specified direction at a point is the average rate at which sound energy is transmitted through a unit area perpendicular to the specified direction at the point considered. For plane or spherical free progressive sound waves, intensity is related to the average pressure by the equation $I = p^2/\rho c$. ρc represents the characteristic impedance of air.

Intermittent Sound. Fluctuating sound whose level falls once or more times to low or unmeasurable values during an exposure. Usually, this means sound that is below 65 dBA at least 10% of any 1-hour period.

Loudness. Loudness is the intensive attribute of an auditory sensation, in terms of which sounds may be ordered on a scale extending from soft to loud. Loudness depends primarily upon the sound pressure of the stimulus, but it also depends upon the frequency and wave form of the stimulus.

Masking. The amount by which the threshold of audibility of a sound is raised by the presence of another (masking) sound.

Maximum Sound Pressure. The maximum absolute value of the instantaneous sound pressure occurring for a given cycle of a periodic wave.

Microbar. A unit of pressure commonly used in acoustics. Equals one dyne per centimeter

Noise. Unwanted sound.

Noise-Induced Permanent Threshold Shift (NIPTS). Permanent threshold shift caused by noise exposure, corrected for the effect of aging (Presbyacusis).

Peak Sound Pressure. The absolute maximum value (magnitude) of the instantaneous sound pressure occurring in a specified period of time.

Reverberation. The sound that persists at a given point after direct reception from the source has stopped.

Sound. (a) An alternation in pressure, stress, particle displacement, or shear in an elastic medium, or (b) an auditory sensation evoked by the above alternations. Not all sound waves evoke an auditory sensation. The medium in which the sound exists is often indicated by "airborne or "structureborne".

Sound Energy. The sound energy of a given part of a medium is the total energy in this part of the medium minus the energy which would exist in the same part of the medium with no sound waves present.

Sound Level. The quantity in decibels measured by a sound level meter satisfying the requirements of American National Standards Specification for Sound Level Meters S1.4-1971. Sound Level is the frequency-weighted sound pressure level obtained with the standardized dynamic characteristic "fast" or "slow" and weighting A, B, or C. Unless indicated otherwise, the A-weighting is understood. The symbol for sound level is dBA.

Sound Level Meter. A device used to measure sound pressure level or weighted sound pressure level, constructed in accordance with the standard specification for sound level meters set up by the American Standards Association. The sound level meter consists of a microphone, an amplifier, a calibrated attenuator, a readout, and weighting networks.

Sound Pressure. The root-mean-square value of the instantaneous sound pressures over a time interval.

Sound Pressure Level. In decibels, 20 times the logarithm to the base 10 of the ratio of a sound pressure to the reference sound pressure of 20 micropascals (20 micronewtons per square meter, .000 microbars).

LIST OF ABBREVIATIONS

ANSI	American National Standard Institute
AMG	American General
API	API
ASTM	American Society of Testing Materials
ATB	Air-to-boil temperature
bhp	Brake horsepower
B&K	Bruel and Kjaer
BTU	British Thermal Unit
°C	Degree Centigrade
cfm	Cubic feet per minute
C/I	Crown-Ikarus
CO	Carbon monoxide
#1D	Number one diesel fuel
#2D	Number two diesel fuel
dB	Decibel
dBA	A-weighted decibel
DDAD	Detroit Diesel Allison Division of General Motors Corporation
DEQ	Oregon Department of Environmental Quality
dia	Diameter
DOT	U.S. Department of Transportation
dyno	Dynamometer
EPA	U.S. Environmental Protection Agency
F	Front
°F	Degree fahrenheit
Flx	Flxible®
fpm	Feet per minute
fps	Feet per second
ft	Feet
FY	Fiscal year
gal	Gallon
GMC	General Motors Corporation
gph	Gallons per hour
gpm	Gallons per minute

GVW	Gross vehicle weight
HC	Hydrocarbon
"Hg	Inches of mercury
"H_2O	Inches of water
hp	Horsepower
hr	Hour
HUD	U.S. Department of Housing and Urban Development
Hz	Hertz
in	Inch
ISO.	International Organization for Standardization
J	Joule
kg	Kilogram
L	Level, left
L_{90}	Ninety percentile level
lb	Pound
Ld	Day level
Ldn	Day-night level
Leq	Equivalent sound level
Ln	Night level
log	Logarithm
m	Meter
METRO	Municipality of Metropolitan Seattle
M	Million
min	Minute
mm	Milimeter
mpg	Miles per gallon
mph	Miles per hour
NBS	National Bureau of Standards
NO_2	Nitrous Oxide
NPT	National pipe thread
NRC	Noise Reduction Coefficient
OAL	Overall length
OD	Outside diameter
OEM	Original equipment manufacture, or manufacturer
Opr	Operational rating
oz	Ounce

Pa	Pascal
PIR	Portland International Raceway
ppm	Parts per million
R	Right, rear, rate
rev	Revolution
psi	Pounds per square inch
rpm	Revolutions per minute
SAE	Society of Automotive Engineers
sec	Second
SP	Static pressure drop
sp gr	Specific gravity
STC	Sound Transmission Class
Tri-Met	Tri-County Metropolitan Transportation District of Oregon
TSC	Transportation Systems Center
TTT	Top tank temperature
UMTA	Urban Mass Transportation Administration
U.S.	United States of America
W	Watt
WAT	Water-to-air temperature

REFERENCES

1. "Noise Reduction Retrofit for a "New Look" Flxible® Transit Bus - Service Bulletin", Michael C. Kaye, Tri-County Metropolitan Transportation District of Oregon, UMTA-OR-06-0005-80-1, September 1980.

2. "Noise Reduction Retrofit for a "New Look" GMC Transit Bus - Service Bulletin Michael C. Kaye, Tri-County Metropolitan Transportation District of Oregon, UMTA-OR-06-0005-81-1, November 1981.

3. "Proposed Bus Noise Emission Regulation", U.S. Environmental Protection Agenc Office of Noise Abatement and Control, August 1977.

4. "Information on Levels of Environmental Noise Requisite to Protect Public Hea and Welfare with an Adequate Margin of Safety, U.S. Environmental Protection Agency, 550/9-74-004, March 1974.

5. "Guidelines for Considering Noise in Land Use Planning and Control", Federal Interagency Committee on Urban Noise, June 1980.

100 copies