Experimental investigation of in-duct insertion loss of catalysts in internal combustion engines

A. Selamet\textsuperscript{a, *}, V. Kothamasu\textsuperscript{a,1}, J.M. Novak\textsuperscript{b}, R.A. Kach\textsuperscript{c}

\textsuperscript{a}The Ohio State University, Department of Mechanical Engineering and Center for Automotive Research, 206 West 18th Avenue, OH 43210, USA
\textsuperscript{b}Powertrain Operations, Ford Motor Company, Dearborn, MI 48121, USA
\textsuperscript{c}Motorsport Technology Department, Ford Motor Company, Dearborn, MI 48121, USA

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Abstract

Acoustic performance characteristics of catalysts in the exhaust system are important in the development of predictive tools for the breathing system of internal combustion engines. To understand the wave attenuation behavior of these elements with firing engines, dynamometer experiments are conducted on a 3.0L V-6 engine with two different exhaust systems: one with the catalysts on the cross-over pipe, and the other that replaces the catalysts with equal length straight pipes. The instantaneous crank-angle resolved pressure data are acquired at wide open throttle and 500 rpm intervals over the operating range of the engine (from 1000 to 5000 rpm) at various locations in both exhaust systems. The effect of the catalyst is then isolated and discussed in terms of insertion loss at critical locations in the exhaust system. The analysis is presented both in terms of time-domain and order-domain. The predictive capability of a finite-difference based time-domain nonlinear approach is also demonstrated as applied to large amplitude waves in the exhaust system of firing engines. © 2000 Elsevier Science Ltd. All rights reserved.

1. Introduction

Extensive literature is available on the acoustic performance of automotive silencers subject to linear disturbances and no-mean or steady-mean flow conditions [1–5]. The studies of these silencers and the other vehicle exhaust components with firing...
engines are, however, rather limited. Among the early studies are Czarnecki and Davis [6,7] who conducted dynamometer experiments with a four stroke, opposed, six cylinder engine to investigate a number of different mufflers, including perforated tubes, straight-through, expansion chamber, and a combination type used in light-airplane noise reduction. Stokes and Davis [8] measured the attenuation characteristics of four fabricated mufflers on a field-test setup with a helicopter powered by a seven-cylinder engine. During the last couple of decades, a number of researchers have concentrated on determining the source characteristics in the exhaust systems of engines: In addition to providing an excellent survey of the earlier works in the area, Prasad and Crocker [9,10] determined the source impedance experimentally for a multi-cylinder engine exhaust system and used this information in the acoustic modeling of insertion loss and radiated sound pressure. Desmons et al. [11] used a four-stroke four-cylinder engine to characterize the source impedance on the exhaust side. While these studies have been instrumental towards understanding the acoustic behavior of engine exhaust systems and components, the sound attenuation performance of the full vehicle exhaust system of multi-cylinder internal combustion engines with a number of different elements continues to be a challenge for both designers and researchers. The task is further complicated because of the competing needs for reduction in noise versus flow losses.

One such important element on the exhaust side is the catalytic converter. Thus far, the pollutant emission control has been the primary objective of these elements, which also dictated the direction of research. The same structure is also effective in reflecting and dissipating the pressure waves, therefore acting as a reactive/dissipative silencer. Recent studies (see, for example, [12]) have examined the acoustic characteristics of these configurations under no flow, and small amplitude (linear acoustics) conditions.

The objective of the present experimental study is to investigate the effect of catalytic converters on the large amplitude wave dynamics and noise attenuation in the exhaust system of firing engines. This is achieved by comparing two exhaust systems (1) with the catalysts (base + catalyst exhaust) and (2) without the catalysts (base exhaust). In order to isolate the effect of catalytic converters alone, the muffler and resonator of the production exhaust system are replaced with a straight pipe. Such comparison clearly reveals the acoustic performance characteristics, including noise reduction and insertion loss of individual catalysts under firing engine conditions. Recent experimental studies have investigated a Ford 1.9L I4 Escort engine with full vehicle exhaust system [13] and a Ford 3.0L V-6 engine with two exhaust manifolds [14]. Kothamasu [15] continued the work with 3.0L V-6 engine and concentrated on the wave attenuation characteristics of a number of elements both in the induction and exhaust systems. The present study focuses on two exhaust configurations coupled to the Ford 3.0L V-6 engine to determine the insertion loss due to the catalysts. The study provides: (1) the experimental findings for crank-angle resolved gas pressure at a number of key locations in the “base” and “base + catalyst” exhaust systems; (2) the overall sound pressure levels at the same locations for the two exhaust systems; (3) half and integer order-tracking (up to 10th order) for insertion loss due to catalysts; and (4) finite difference based time-domain nonlinear model predictions for the base exhaust system.
Following this introduction, Section 2 describes the dynamometer experiments, including the setup, the steady state and cycle-resolved data acquisition, and the data processing. The insertion loss due to catalysts at a number of key locations is discussed in Section 3 in terms of individual order tracking as a function of speed, while the overall sound pressure levels at the same locations are included in Appendix A. The computational model predictions for large amplitude wave propagation are presented in Section 4, followed by concluding remarks in Section 5.

2. Dynamometer experiments

The experiments are conducted on a Ford 1992 3.0L V-6 (Taurus) engine. The basic features of this engine are summarized in Table 1. The wide open throttle (WOT) data are acquired at 500 rpm increments from 1000 to 5000 rpm for two different exhaust systems: (1) A “base + catalyst” exhaust system which includes the catalytic converters of the production exhaust system, while the muffler is replaced with an “equal length” straight pipe; and (2) A “base” exhaust system, where the catalysts on the cross-over pipe in addition to the muffler are also replaced with straight pipes. The primary objective of these experiments is to determine the insertion loss due to catalysts. Fig. 1 shows the exhaust system with numbered circles identifying the locations of pressure transducers and thermocouples. Locations 1–8 are in the exhaust manifold, 9–12 are in the Y-pipe and 13–18 are in the exhaust pipe. Fig. 2 shows primarily the right bank during experiments with the pressure transducers installed at locations 1, 9, 11, 12, and 13. The dimensions of catalyst housing are provided in Fig. 3. The overall length of the exhaust system is retained the same in order to facilitate a direct comparison at different locations. The exhaust pipe is connected to a surge tank (large volume) to dampen the pressure waves in place of expansion to ambient. The system is instrumented with absolute pressure transducers and gas thermocouples to quantify the noise reduction and flow losses due to

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Engine specifications</th>
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<tbody>
<tr>
<td>Number of cylinders</td>
<td>6</td>
</tr>
<tr>
<td>Bore</td>
<td>8.9 cm</td>
</tr>
<tr>
<td>Stroke</td>
<td>8.0 cm</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>14.05 cm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>9.25</td>
</tr>
<tr>
<td>Firing order</td>
<td>1 4 2 5 3 6</td>
</tr>
<tr>
<td>Intake valve open (cylinder#1)</td>
<td>334.5 ATDC</td>
</tr>
<tr>
<td>Intake valve close (cylinder#1)</td>
<td>622.5 ATDC</td>
</tr>
<tr>
<td>Exhaust valve open (cylinder #1)</td>
<td>105.5 ATDC</td>
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<tr>
<td>Exhaust valve close (cylinder #1)</td>
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<tr>
<td>Intake valve diameter</td>
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</tr>
<tr>
<td>Exhaust valve diameter</td>
<td>3.428 cm</td>
</tr>
</tbody>
</table>
each element. The measurement locations as well as other critical dimensions are specified in Table 2.

2.1. Steady-state data acquisition

The engine is connected to a General Electric Type TLC 2554S DC electric dynamometer capable of absorbing 300 HP, delivering 225 HP and a maximum speed of 6000 rpm. The engine operating conditions are controlled by Dyne Systems DYN-LOC IV digital dynamometer controller and the throttle position by the Dyne Systems DTC-1 digital throttle controller. Horiba EDTCS-1000 system, a combination of hardware and software communicates with the dynamometer and throttle controller to control and monitor the engine-dyno operating conditions as well as acquire low speed (sampling rate ≤ 10 Hz) data. The WOT data are recorded after the engine reaches an equilibrium state, determined by exhaust and coolant temperatures, torque and fuel consumption at each particular speed. Engine coolant and oil temperatures are controlled by a closed loop cooling stand within ±2°F. Engine torque is obtained by a load cell. The brake power is calculated using the brake torque and engine speed measured by a Lucas Ledex Series DG25 optical encoder. The brake torque for the base and base + catalyst is almost the same at all engine speeds. The consumption rate of fuel mass is determined with a Pierburg Instruments FT22-34 fuel measurement system with FT10E flow meter. Exhaust gas analysis is done by MEXA 7100 exhaust gas analyzer. The analyzer is capable of measuring the oxygen (O₂), carbon-monoxide (CO), carbon-dioxide (CO₂), total hydrocarbon (THC), and nitrogen oxide (NOₓ) concentrations in the exhaust gas and therefore yields air/fuel ratio through chemical analysis. The Horiba EDTCS-1000 system is connected to the analyzer for monitoring and data acquisition of species concentrations. Engine oil and outlet cooling water temperatures are measured using RTD probes with 1/4" tip diameters. The average inlet and exhaust gas temperatures are measured near the

![Base + Catalyst Exhaust System](image_path)

Fig. 1. 3.0L V-6 Base + Catalyst exhaust system: pressure transducer and thermocouple locations.
pressure tap points by Ωmega type K (chromel–alumel junction) ungrounded probe thermocouples with 1/8” tip diameters. All thermocouples and sensors are connected to Horiba EDTCS-1000 system for real time display and data acquisition.

2.2. Time-dependent pressure data acquisition

The crank-angle resolved pressure data at a number of locations in the exhaust system (recall Fig. 1) is acquired using Concurrent Masscomp 7250 RTU, high speed data acquisition system. The system is capable of acquiring data simultaneously for 32 channels at a sampling rate of 2 MHz and has a 12 bit A/D converter. Kistler Type 4045A2 and Type 4045A5 piezoresistive pressure transducers (2 and 5 bar, respectively) are used to measure the instantaneous exhaust pressures. The 5 bar

Fig. 2. The right bank and the Y-junction of exhaust system with water cooled pressure transducers during high speed engine experiments.
transducers are used for primary exhaust runner locations and the 2 bar transducers for the rest of the locations. Water cooled adapters are used to keep the exhaust transducers below their maximum operating temperature of 120°C. Kistler Type 4603A00726 digital piezoresistive amplifiers amplify the pressure signals from the piezoresistive transducers to a 0–10 Volts scale and send the output to the sample and hold card of the data acquisition system.

Lucas Ledex Series DG25 optical encoder connected to the crankshaft in front of the engine sends a θ pulse every crank angle as control signal to the A/D board of the data acquisition system. A proximity sensor mounted in the distributor cap,

Table 2
Exhaust system dimensions\(^a\)

<table>
<thead>
<tr>
<th>Duct</th>
<th>Length (cm)</th>
<th>Diameter (cm)</th>
<th>Duct</th>
<th>Length (cm)</th>
<th>Diameter (cm)</th>
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<td>3.3</td>
<td>k2</td>
<td>11.02</td>
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</tr>
<tr>
<td>a2</td>
<td>7.0</td>
<td>3.3</td>
<td>l</td>
<td>31.5</td>
<td>Base exhaust: 4.3; base + catalyst: variable (see Figs. 1 and 2)</td>
</tr>
<tr>
<td>b</td>
<td>14.9</td>
<td>3.3</td>
<td>m1</td>
<td>8.9</td>
<td>4.3</td>
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<td>m2</td>
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<td>n1</td>
<td>27.94</td>
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</tr>
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<td>8.3</td>
<td>3.3</td>
<td>n2</td>
<td>9.18</td>
<td>4.3</td>
</tr>
<tr>
<td>f</td>
<td>23.14</td>
<td>3.3</td>
<td>o</td>
<td>31.5</td>
<td>Base exhaust: 4.3; base + catalyst: variable (see Figs. 1 and 2)</td>
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<tr>
<td>g</td>
<td>15.2</td>
<td>3.3</td>
<td>p1</td>
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<td>p2</td>
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<td>q1</td>
<td>395.25</td>
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<td>q2</td>
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<td>r</td>
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<td>7.29</td>
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</table>

\(^a\) The notation is given in Fig. 1.
provides a pulse once every cycle at the TDC combustion of cylinder #1. The θ and the TDC signals are passed through an optical isolator to filter out the noise that may be present. The noise-free θ and the TDC signals are then used to set the clocks of the data acquisition system to acquire instantaneous pressure data at every crank angle for a sampling period of 64 cycles simultaneously at all locations.

2.3. Data processing

The Horiba EDTCS-1000 system acquires steady-state engine parameters such as speed, spark advance, brake torque, engine oil and coolant temperatures, fuel consumption rate, temperatures at different locations, exhaust gas species concentration and air/fuel ratio data. These parameters are acquired for a given engine speed at a sampling rate of 10 Hz for total of 15 s and their averaged values represent the steady state conditions at that operating point. Horiba system software calculates other engine parameters from the measured quantities using standard relationships. HyperScript Tools is used by the software to process the engine parameters data and prepare the experimental reports in the spreadsheet format. The raw exhaust pressure data that are collected for 64 cycles are averaged per crank angle degree to determine a repeatable, average time-dependent pressure profile at every location for

Fig. 4. Pressure versus CAD in the base exhaust at location 1: ——, 1000 rpm; ——, 3000 rpm; ——, 5000 rpm.
a given engine speed. The averaging is preceded by an inspection of several engine cycles to insure the repeatability of the individual cycles. The data is also averaged over the entire sampling period to determine a mean static pressure of exhaust gases at the measurement locations.

Fig 5. Pressure versus CAD in the base and base+catalyst exhaust at location 1 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: — , base; — — , base+catalyst.

3. Insertion loss (IL)

The time domain pressure data may be converted to frequency (or order) domain information through a discrete Fourier transformation [16]. The sound pressure level for each frequency component may then be expressed in terms of the root mean

Fig. 6. Pressure versus CAD in the base and base+catalyst exhaust at location 9 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: — , base; — , base + catalyst.
square (rms) value of pressure as \( L_{p,i} = 20 \log_{10} \left( \frac{P_{\text{rms},i}}{P_{\text{ref}}} \right) \) where \( P_{\text{ref}} = 2 \times 10^{-5} \text{ Pa} \) and the units for \( L_p \) being dB (decibel). For convenience of presentation, orders are used here instead of frequencies, where the \( m \)th order is defined as \( m \times \left( \frac{n \text{ [rpm]}}{60} \right) \). The insertion loss is the difference in the sound pressure levels measured at a specific location without and with the acoustic filter: Insertion loss \( (IL_i) = L_{p, \text{ without filter}} - L_{p, \text{ with filter}} \).

Fig. 7. Pressure versus CAD in the base and base+catalyst exhaust at location 11 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: —–, base; — —, base+catalyst.
Fig. 4 shows the time variation of pressure in the exhaust primary runner at location 1 of the base exhaust system to illustrate the increase in pressure amplitude with speed. The peak-to-valley pressure amplitudes in the exhaust primary runners vary from 0.15 to 1.5 bar and the mean pressure from 1 to 1.5 bar with engine speed.

The discussion on the time-domain pressure traces and the corresponding $L_p$ is limited to locations 1, 9, 11 (right bank), 13, and 18 only and three engine speeds:

![Pressure versus CAD in the base and base+catalyst exhaust at location 13 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: ——, base; ——, base+catalyst.](image_url)
1000, 3000, and 5000 rpm due to space considerations (see Ref. [15] for the left bank locations 6, 10, and 12). Fig. 5 shows the superimposed pressure variation in the exhaust primary runner at location 1 for the base (solid line) and base + catalyst (dashed line) exhaust systems for three engine speeds: 1000, 3000, and 5000 rpm. Also included in the same figure are the exhaust valve opening (EVO) and closing

Fig. 9. Pressure versus CAD in the base and base + catalyst exhaust at location 18 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: —— , base; — — , base + catalyst.
(EVC) events. The catalysts in the exhaust system modify the pressure wave form and reduce the overall amplitudes in the primary runners relative to the base exhaust. In the primary runners, the mean pressure and the amplitude increase with speed. Particularly at higher speeds, the insertion of catalyst raises the mean pressures [Figs. 5(b) and (c)]. For the three engine speeds, Figs. 6 and 7 compare the pressure variation across the right catalyst (locations 9 and 11) in the “Y-pipe”, and Figs. 8

Fig. 10. Pressure versus CAD in the base and base+catalyst exhaust at location 1 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: — , base; — — , base + catalyst.
and 9 at the exhaust pipe locations (13 and 18). In general, in both exhaust systems, the mean and the peak pressure decreases with distance from the exhaust valve. At the high engine speed (5000 rpm), the peak pressure in the base exhaust decreases from about 2.2 bar in the exhaust primary runners to 1.5 bar at locations immediately downstream of catalysts and further decreases to 1.25 bar in the tail pipe. The presence of catalysts in the exhaust system increases the mean back pressure at the

![Fig. 11. Sound pressure level versus order in the base and base+catalyst exhaust at location 9 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: —– , base; — — , base + catalyst.](image)
upstream locations due to flow separation and viscous friction losses and, in general, decreases the pressure amplitudes at all locations. These trends are enhanced with engine speed due to increasing mass flow rate.

The sound pressure level versus order corresponding to the pressure-time data of the preceding Figs. 5–9 are given in Figs. 10–14 which show the relative $L_p$ up to the 10th engine order. In both exhaust systems the $L_p$ of the 3rd order is one of the

Fig. 12. Sound pressure level versus order in the base and base+catalyst exhaust at location 11 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: --- , base; --- , base+catalyst.
important orders contributing to the overall sound pressure level, $L_{po}$, at all engine speeds and locations, as expected. Figs. 10–12 show the $L_p$ variation with order in the exhaust manifold (location 1), and across the right catalyst (locations 9 and 11). At locations upstream of catalysts (1 and 9), the $L_p$ of the $4\frac{1}{2}$ order at the low engine speed and the $1\frac{1}{2}$ at higher engine speeds are significant contributors to the $L_{po}$. At higher engine speeds, the $L_p$, in general, decreases with order in the primary runners.

Fig. 13. Sound pressure level versus order in the base and base+catalyst exhaust at location 13 for (a) 1000 rpm, (b) 3000 rpm, (c) 5000 rpm: ——, base; ——, base + catalyst.
At these locations, the sound pressure levels in the base and base+catalyst exhaust are different for almost all orders. At higher engine speeds, the $1\frac{1}{2}$ and $4\frac{1}{2}$ orders are important in the base exhaust in comparison to the 3rd order. The IL due to catalyst at location 11 tends to increase with increasing engine speed.

Figs. 13 and 14 show the variation of $L_p$ with order downstream of catalysts in the exhaust pipe locations 13 and 18. Similar to location 11, the $4\frac{1}{2}$ order at location 13
(Fig. 13) of the base exhaust is important besides the fundamental 3rd order at the low engine speed, and the $1\frac{1}{2}$ and $4\frac{1}{2}$ orders at the higher engine speeds. In the base + catalyst exhaust system, the contribution from higher orders ($\geq 4\frac{1}{2}$) decreases with increasing speed, and only the $1\frac{1}{2}$ order is important at the high engine speed.

Fig. 15. Individual order sound pressure level versus engine speed in the base and base + catalyst exhaust systems at location 1 (a) order = 1.5, (b) order = 3, and (c) order = 4.5: — , base; — — , base + catalyst.
(5000 rpm) apart from the 3rd order. The IL due to catalysts may be as high as 15 dB, and, in general, are higher at the high engine speeds. At location 18 (Fig. 14), the 3rd order dominates at low speed, while several others contribute significantly at higher speeds in the base and base + catalyst exhaust systems, and the IL due to catalysts remain within 10 dB.

![Graph](image)

**Fig. 15.** Individual order sound pressure level versus engine speed in the base and base + catalyst exhaust systems at location 1: (d) order = 6, (e) order = 7.5, and (f) order = 9: ——, base; — —, base + catalyst.

(ii) Second Part
The foregoing details on the order analysis are included also to shed light on the impact of Y-pipes with unequal branch lengths inherent to V-engine design. Numerous half orders discussed above may be attributed to the inequality of path lengths on both sides of the cross-over pipe.

Fig. 16. Individual order sound pressure level versus engine speed in the base and base + catalyst exhaust systems at location 9 (a) order = 1.5, (b) order = 3, and (f) order = 4.5: — , base; — — , base + catalyst.
3.1. Order tracking with speed

The acoustical performance of the catalysts can be understood better from the variation of $L_p$ of the dominant orders with engine speed. Figs. 15–19 show the individual sound pressure levels versus engine speed for the first few half and integer orders.
orders (1 $\frac{1}{2}$, 3rd, 4 $\frac{1}{2}$, 6th, 7 $\frac{1}{2}$, and 9th) for the two exhaust systems as a function of location. Some information in Figs. 10–14 is repeated in Figs. 15–19. The relative $L_p$ of the half and integer orders up to the 10th for three representative engine speeds is illustrated in Figs. 10–14, whereas Figs. 15–19 track the variation of $L_p$ of the dominant

(i) First Part

Fig. 17. Individual order sound pressure level versus engine speed in the base and base+catalyst exhaust systems at location 11 (a) order = 1.5, (b) order = 3, and (c) order = 4.5: —- , base; —— , base + catalyst.
orders with speed and therefore provide directly the IL of catalysts at any speed and location.

The IL due to catalysts at a location is the difference between the $L_p$ in the base and base+catalyst exhaust systems. Fig. 15 shows the $L_p$ versus speed at location 1.

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Fig. 17. Individual order sound pressure level versus engine speed in the base and base+catalyst exhaust systems at location 11 (d) order = 6, (e) order = 7.5, and (f) order = 9: ——, base; ———, base+catalyst.
of the exhaust manifold and Figs. 16 and 17 at locations 9 and 11 of the Y-pipe. The $L_p$ of the $1\frac{1}{2}$ order increases, in general, with speed. The IL for the $1\frac{1}{2}$ order due to catalysts at locations 1 and 9 is negative up to 3000 rpm. As the speed increases further, the IL for the $1\frac{1}{2}$ order becomes positive and tends to increase. The IL due

![Graphs showing individual order sound pressure level versus engine speed in the base and base+catalyst exhaust systems at location 13 (a) order = 1.5, (b) order = 3, and (c) order = 4.5: ——, base; ———, base + catalyst.](image-url)
to catalysts in the Y-pipe for the 3rd order are always positive at any engine speed. There appears to be no general trend for the IL due to catalysts at these locations at other orders. The IL due to catalysts vary between −10 and 20 dB in the exhaust manifold and the Y-pipe.

Fig. 18. Individual order sound pressure level versus engine speed in the base and base + catalyst exhaust systems at location 13 (d) order = 6, (e) order = 7.5, and (f) order = 9: ——, base; ———, base + catalyst.
The $L_p$ variation with speed in the exhaust pipe at locations 13 and 18 is shown in Figs. 18 and 19. The comparison of base and base + catalyst exhaust systems in these figures, clearly demonstrate the effect and IL due to catalysts. Consider, for example, location 13. The addition of catalysts to the exhaust system leads to about 5 to

![Diagram](image_url)

Fig. 19. Individual order sound pressure level versus engine speed in the base and base + catalyst exhaust systems at location 18 (a) order = 1.5, (b) order = 3, and (c) order = 4.5: — , base; — — , base + catalyst.
12 dB insertion loss at this location for the dominant fundamental firing frequency [3rd order (Fig. 18b)]. With the exception of couple of engine speeds for some orders, the insertion loss due to catalysts tends to become stronger with increasing order, therefore frequency.

Fig. 19. Individual order sound pressure level versus engine speed in the base and base + catalyst exhaust systems at location 18 (d) order = 6, (e) order = 7.5, and (f) order = 9: — , base; — — , base + catalyst.

(ii)Second Part
4. Computational approach

The numerical technique is based on the finite difference approximation of the balance equations in the time domain for mass, momentum, and internal energy. For one-dimensional flow in ducts of variable cross-section with neglected axial conduction, the governing equations may be expressed, as

![Graphs showing pressure versus crank angle for different speeds](image)

Fig. 20. Comparison of the predictions and experiment for pressure versus crank angle in the base exhaust primary runner 1 at (a) 1000 rpm, (b) 3000 rpm, and (c) 5000 rpm: —–, experiment; ——, computational model.
\[
\frac{\partial}{\partial t} (\rho A) + \frac{\partial}{\partial x} (\rho A U) = 0,
\]
\[
\frac{\partial}{\partial t} (\rho A U) + \frac{\partial}{\partial x} (\rho A U^2) + \frac{\partial}{\partial x} (PA) - \tau_w \varphi = 0,
\]

Fig. 21. Comparison of the predictions and experiment for \(L_p\) versus order in the base exhaust primary runner 1 at (a) 1000 rpm, (b) 3000 rpm, and (c) 5000 rpm: —— experiment; —— computational model.
\[
\frac{\partial}{\partial t}(\rho Ae) + \frac{\partial}{\partial x}(\rho A U e) + P \frac{\partial}{\partial x}(U A) - \tau_w \phi U + q \phi = 0,
\]

(3)

where \( \rho \) is the density, \( A \) is the cross-sectional area, \( U \) is the velocity, \( P \) is the pressure, \( \tau_w \) is the wall shear stress, \( \phi \) is the perimeter, \( e \) is the specific internal energy, and \( q \) is the wall heat transfer rate. The ideal gas equation of state,

\[
P = (\gamma - 1) \rho e,
\]

(4)

where \( \gamma \) is the ratio of specific heats, is used to relate the thermodynamic variables and close the system of equations. Eqs. (1–3) are discretized by employing the explicit finite difference method of Chapman et al. [17]. The staggered mesh used in the discretization divides a duct into cells with vector quantities located at node points and scalar quantities at cell midpoints. The predictions presented in this study use a grid size of \( \Delta x = 0.5 \) cm.

This computational technique has already been used successfully for simple [18] as well as relatively complex [19] mufflers in the absence of mean flow and temperature gradients. Here, in the presence of oscillating mean flow, large amplitude waves, and temperature gradients, the model predictions for the time-domain pressure and the corresponding \( L_p \) are presented for the base exhaust system only. For illustration, Figs. 20 and 21 compare the predictions and experiments in the exhaust primary runner location 1. The predictions compare reasonably well with experiments. At the low

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**Fig. 22.** Comparison of \( L_{po} \) in the base exhaust system between the experiment and the computational model at 1000 rpm: ●, experiment; O, computational model.
Fig. 23. Comparison of $L_{po}$ in the base exhaust system between the experiment and the computational model at 3000 rpm: ●, experiment; O, computational model.

Fig. 24. Comparison of $L_{po}$ in the base exhaust system between the experiment and the computational model at 5000 rpm: ●, experiment; O, computational model.
engine speed, except for the shift in CAD, the predicted pressure wave form is nearly the same as the measured one. With increasing engine speed, the phasing as well as the amplitudes have been predicted with reasonable accuracy. Across the speed range, the $L_p$ predictions for the first few half and integer orders up to 10th order show good agreement with experiments at the lower orders and have some settled differences at the higher orders.

Fig. 25. Overall sound pressure level versus engine speed in the base and base+catalyst exhaust systems on the right bank at locations (a) 1, (b) 9, and (c) 11: O, base; □, base+catalyst.
The experimental and computational results for the $L_{po}$ in the base exhaust system are compared in Figs. 22–24 at 1000, 3000, and 5000 rpm. The computational model predicts the $L_{po}$ in the base exhaust system rather accurately. Figs. 22–24 also reveal the amplitude pattern as a function of location (the shape of quasi-standing wave). These comparisons suggest that the time-domain computational tool, which is briefly introduced here, is a viable and powerful technique to examine and understand the wave propagation in internal combustion engines.

Fig. 26. Overall sound pressure level versus engine speed in the base and base + catalyst exhaust systems on the left bank at locations (a) 6, (b) 10, and (c) 12: O, base; □, base + catalyst.
5. Concluding remarks

The present experimental work provided for the first time the insertion loss characteristics of catalytic converters at WOT firing engine conditions. The instantaneous crank-angle resolved pressure at key locations in the exhaust system with and without the catalysts exhibit strong spatial variation in the mean and peak pressure. The discrete Fourier transformed time-domain pressure data show high sound
pressure levels reaching 185 dB, substantially above the linearized acoustic theory. The comparison of the $L_p$ of the first few half and integer orders at the key locations show that the fundamental firing frequency is one of the important orders contributing to the $L_{po}$ in the exhaust system, while the significance of other orders varies with speed and depends on the exhaust system.

Crank-angle resolved pressure data and the corresponding $L_p$ for the half and integer orders (up to 10th order) in the base and base+catalyst exhaust systems demonstrate that the insertion of catalysts in the exhaust system affects the pressure at their upstream, as well as downstream locations. At locations upstream of catalysts, the mean pressure increases with the insertion of catalysts in the exhaust system due to a combination of flow separation and viscous friction losses. The insertion of catalysts is shown (through reflection and absorption of the pressure waves) to reduce, in general, the $L_p$ at the fundamental firing frequency, as well as higher frequencies at locations downstream of catalysts. The overall insertion loss tends to be higher at higher engine speeds.

The present study employed two essentially identical catalysts. It is therefore expected that the results will somewhat depend on the design of these elements. While this is true for the quantitative observations, it is also expected that the qualitative behavior and the trends presented in this study will remain applicable for similar elements in the exhaust system of internal combustion engines. It should also be noted that the use of “insertion loss” is generalized in the present work to upstream locations of the elements as well. For these upstream locations, an alternative wording for the same definition such as “insertion impact” may possibly be more appropriate.

The nonlinearities observed in the foregoing exhaust systems, such as those due to high pressure amplitudes measured in this study, may be difficult to treat with the linear acoustic theory in the frequency-domain. The ultimate objective of the work currently in progress is then to validate a time-domain finite difference approach for the prediction of both acoustic and flow performances based on the work of Chapman et al. [17], which solves the one-dimensional, variable cross-sectional area, nonlinear balance equations of mass, momentum and internal energy coupled with the equation of state for compressible flows (see also [20]). Illustrative base exhaust results from this approach are included in the present, predominantly experimental work; while further details will be reported later, including the modeling of catalysts.

Appendix A. Overall sound pressure level tracking with speed

The $L_{po}$ in the exhaust at various locations as a function of speed is shown in Figs. 25–27. The $L_{po}$ in the base exhaust system is, in general, higher than the base+catalyst exhaust systems. The difference in $L_{po}$ between the base and base+catalyst exhaust at a specific location is the overall IL due to catalysts.

The $L_{po}$ in the primary exhaust runners of base and base+catalyst exhaust systems (Figs. 25a and 26a) varies from 170 to 185 dB, and, in general, increases with speed. The $L_{po}$ at locations 1 and 6 of the exhaust manifolds (Figs. 25a and 26a) and locations 9 and 10 of the Y-pipe (Figs. 25b and 26b) is lower in the base+catalyst
exhaust than the base exhaust at low and high engine speeds, while being nearly the same in the mid speed range. The $L_{po}$ at locations 11, 12, and 13 (Figs. 25c, 26c and 27a) is lower in the base+catalyst than the base exhaust. At location 18 of the tailpipe (Fig. 27b), $L_{po}$ varies from 165 to 175 dB in the base exhaust and 160 to 170 dB in the base+catalyst exhaust and the IL due to catalysts varies from 2 to 8 dB with speed. The attenuation increase at higher engine speeds, therefore frequencies, may be attributed to the basic behavior of catalysts investigated in Ref. [12], which in general, is higher at high engine speeds.

References

