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Lean NOx Technology - Phase 1

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by
Engine Technologies
ORTECH Corporation
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16. Abstract

Exhaust emissions standards are becoming more and more stringent. One promising approach to the reduction of oxides of nitrogen (NOx) emissions is exhaust aftertreatment in the form of lean NOx catalysis. The catalytic reduction of NOx in oxygen-rich diesel exhaust has been demonstrated to be feasible if a suitable reducing agent, such as diesel fuel, is added to the exhaust stream. However, with this technology, problems with low catalyst efficiencies and increased hydrocarbon emissions from unreacted diesel reductant must still be overcome.

The catalytic NOx reduction efficiency can be increased by the optimization of parameters such as the phasing and quantity of reductant to be introduced. To better understand exhaust flow dynamics and guide future development of laboratory synchronous NOx reductant metering systems, a dynamic exhaust flow model was developed and then refined using engine test data. The model predicts instantaneous exhaust and emissions mass flow rates at the ports, turbocharger inlets and catalyst inlet of a heavy duty 4-cylinder diesel engine.

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16. Résumé

Les normes concernant les émissions polluantes deviennent de plus en plus strictes. Une technologie prometteuse de réduction des émissions d'oxydes d'azote (NOx) est le traitement postcombustion de ces émissions par réduction catalytique selon le procédé «lean NOx». Il a été démontré que la réduction catalytique des NOx dans les gaz d'échappement riches en oxygène des moteurs diesel est possible lorsqu'on ajoute un agent réducteur approprié, notamment du carburant diesel, aux gaz d'échappement. Il est à noter toutefois que les problèmes découlant de cette technologie - faible efficacité des catalyseurs et augmentation des émissions d'hydrocarbures provenant du réducteur diesel subsistant après catalyse - doivent encore être surmontés.

On peut accroître l'efficacité de la réduction catalytique des NOx en optimisant des paramètres tels que la distribution du réducteur ainsi que la quantité injectée. Pour mieux comprendre la dynamique des gaz d'échappement et aider au développement futur de systèmes de dosage synchronisé de réducteur des NOx, on a développé un modèle dynamique des débits d'échappement que l'on a affiné à l'aide des données obtenues aux essais sur moteur. Le modèle prévoit des débits massiques d'échappement et d'émissions instantanés aux lumières d'échappement, à l'admission de la turbosoufflante et à l'entrée du catalyseur d'un moteur diesel poids lourds à 4 cylindres.

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EXECUTIVE SUMMARY

Today's heavy duty diesel engine manufacturers are continuously striving to meet ever more stringent exhaust emissions standards. While significant progress to this end has been made through the use of electronic control of diesel injection timing and duration, and the optimization of the diesel combustion process and fuel injector design, these approaches alone are not sufficient to achieve the emissions targets of the 21st century. One promising approach to further reducing exhaust emissions, particularly those of oxides of nitrogen (NOx), is exhaust aftertreatment. The catalytic reduction of NOx in oxygen-rich diesel exhaust has been demonstrated to be feasible if a suitable reducing agent, such as diesel fuel, is added to the exhaust stream. However, low catalyst efficiencies and increased hydrocarbon emissions from unreacted diesel reductant still plague this technology.

Areas where improvements to the lean NOx reductant technology could be realized were identified as follows:

- reductant metering system design;
- reductant injection control (amount, phasing);
- reductant introduction location, mixing, temperature and phase (gas or liquid);
- catalyst formulation and size.

This phase of a proposed multi phase lean NOx technology program focused on the generation of a computer model to predict exhaust gas flow rates in a diesel engine. The ability to predict exhaust flows, pressures and temperatures will enhance the understanding of the degree of reductant/exhaust homogeneity or stratification which could be achieved by the proper control (phasing) and introduction of reductant. The results of this work will help guide future phases in the development of laboratory hardware and on-engine testing to seek:

- the optimal phasing of reductant and exhaust;
- the optimal reductant mixing state (homogenous or stratified);
- the optimal reductant introduction state (liquid/gas, temperature, particle size distribution, momentum);
- the optimal reductant introduction site (port, turbocharger inlet, catalyst inlet);
- the optimal quantity of reductant.

Thus, the specific objectives of this program were to develop a dynamic exhaust flow model and then refine and calibrate the model using engine test data.

A literature review established that several complex engine flow models existed. However, it was deemed that a less sophisticated model would provide the required information to

determine the reductant phasing needed to deliver homogeneous or stratified reductant/ exhaust charge to the catalyst. The flow of exhaust gas from an engine can be obtained from the flow across its exhaust valve(s). Equations giving exhaust mass flow rate as a function of valve geometry (lift, area) and operating conditions (pressures, temperatures, gas composition) were developed, as well as equations to give the phasing of the exhaust pulse (one per cylinder per engine cycle). The dynamic exhaust flow model was developed as an EXCEL spreadsheet.

To verify and calibrate the flow model, engine tests on a DDC Series 50 diesel engine were undertaken. The engine was instrumented to measure exhaust pressures and temperatures, emissions, intake air flow and fuel flow. The engine was then operated at each of 16 steady state speed load points over its operating range. A high speed data acquisition system was used to record the data. As a result of the simplifying assumptions, the dynamic flow model's predictions differed from the measurements, in terms of scaling only. The flow model was adjusted based on actual engine air/fuel throughput rates and measured exhaust conditions.

The flow model's predictions indicated certain implications for the future development of synchronous reductant metering systems. The model predicts that fully homogeneous or stratified exhaust/reductant mixtures are attainable when reductant is introduced synchronously at the ports and turbocharger inlets. However, fully homogeneous or stratified mixtures are not attainable if reductant is introduced at the catalyst inlet. The model's predictions concerning instantaneous mass flow rates of NOx, HC and CO will help to determine the quantities of reductant that need to be introduced at a given operating condition.

The current exhaust flow model is based on a 4-cylinder engine only. To apply to the prediction of flows in engines having a greater number of cylinders, it would have to be modified. Future work should include a "Lean NOx Parametric Study", involving the participation of engine and catalyst manufacturers, to investigate the identified system issues and, thus, to help establish lean NOx technology.

SOMMAIRE

À l'heure actuelle, les fabricants de moteurs diesel poids lourds s'efforcent continuellement de respecter des normes antipollution de plus en plus strictes. Même si on a déjà réalisé de grands progrès à cet égard en recourant à la commande électronique de la durée et de l'avance à l'injection ainsi qu'en optimisant le processus de combustion diesel et la conception de l'injecteur, ces approches ne suffisent pas pour atteindre les niveaux visés pour le XXI^{e} siècle. Une technologie qui promet de réduire davantage les émissions d'échappement, surtout celles d'oxydes d'azote (NOx), est le traitement postcombustion de ces gaz. On a démontré que la réduction catalytique des NOx dans les gaz d'échappement riches en oxygène des moteurs diesel est possible si on y ajoute un agent réducteur, du carburant diesel notamment. Il reste toutefois certains problèmes à surmonter : faible efficacité des catalyseurs et augmentation des émissions d'hydrocarbures provenant du réducteur diesel subsistant.

Les aspects de la technologie de réduction catalytique selon le procédé «lean NOx» qui seraient susceptibles d'être améliorés sont les suivants :

- conception du système de dosage du réducteur;
- réglage de l'injection de réducteur (quantité, distribution);
- endroit d'injection, mélange, température et phase du réducteur (gazeuse ou liquide);
- formulation et dimension du catalyseur.

Cette phase du programme échelonné sur la technologie «lean NOx» visait la production d'un modèle informatique permettant de prévoir les débits de gaz d'échappement dans un moteur diesel. Ces prévisions concernant le débit, la température et la pression des gaz d'échappement faciliteront la compréhension du degré d'homogénéité ou de stratification du mélange réducteur/gaz d'échappement qu'il serait possible d'obtenir par une distribution adéquate du réducteur et son injection en quantité appropriée. Les résultats de ces travaux contribueront à guider les phases futures du programme dans le développement de matériel de laboratoire et d'essais sur moteur afin d'obtenir :

- la distribution optimale de réducteur et de gaz d'échappement;
- l'état optimal de mélange (homogène ou stratifié);
- l'état optimal du réducteur au moment de son injection (liquide/gaz, température, granulométrie, quantité de mouvement);
- l'endroit optimal d'injection du réducteur (lumière d'échappement, admission de la turbosoufflante, entrée du catalyseur);
- la quantité optimale de réducteur.

Ce programme avait donc pour objectifs précis de développer un modèle dynamique des débits d'échappement, puis de raffiner et d'étalonner le modèle à l'aide des données d'essai sur moteur.

Une recherche documentaire a permis d'établir qu'il existait déjà plusieurs modèles complexes de simulation des débits de gaz d'échappement de moteur. On a toutefois estimé qu'un modèle plus simple permettrait d'obtenir les renseignements nécessaires pour déterminer la distribution de réducteur à prévoir pour alimenter le catalyseur avec une charge homogène ou stratifiée de réducteur et de gaz d'échappement. Le débit de gaz d'échappement d'un moteur peut être établi d'après le débit à la ou aux soupapes d'échappement. On a développé des équations donnant le débit massique des gaz d'échappement en fonction de la géométrie de la soupape (hauteur d'ouverture, section) et les conditions de fonctionnement (pressions, températures, composition des gaz) ainsi que des équations permettant de déterminer la distribution des flux pulsés d'échappement (un par cylindre par cycle). Le modèle dynamique des débits d'échappement a été élaboré sous forme de tableur EXCEL.

Pour vérifier et étalonner le modèle des débits, on a procédé à des essais sur un moteur diesel DDC de la série 50 équipé d'instruments servant à mesurer les pressions et les températures d'échappement, les émissions ainsi que les débits d'admission d'air et de carburant. Puis, on a fait fonctionner le moteur en régime stabilisé à chacune des 16 valeurs de vitesse et de charge de référence dans la plage de fonctionnement. Les données ont été recueillies par un système d'acquisition haute vitesse. Les prévisions obtenues avec le modèle dynamique différaient des mesures prises mais uniquement en termes d'échelle, en raison des hypothèses simplificatrices adoptées. On a alors réglé le modèle en fonction des débits réels du mélange d'air et de carburant et des conditions d'échappement qui avaient été mesurées.

Les prévisions faites à l'aide du modèle dynamique indiquaient certaines orientations à prendre en compte pour le développement futur des systèmes de dosage synchronisé du réducteur. D'après le modèle, il est possible d'obtenir des mélanges gaz d'échappement/ réducteur entièrement homogènes ou stratifiés lorsqu'on introduit le réducteur de façon synchronisée aux lumières d'échappement et à l'admission de la turbosoufflante, mais ce n'est pas le cas lorsqu'on l'injecte à l'entrée du catalyseur. Les prévisions concernant les débits massiques instantanés de NOx, d'HC et de CO contribueront à déterminer les quantités de réducteur à injecter à un point de fonctionnement donné.

Le modèle actuel est fondé sur un moteur à 4 cylindres uniquement. Il faudra le modifier si l'on veut s'en servir pour des moteurs ayant un plus grand nombre de cylindres. Les travaux subséquents devraient prévoir une étude paramétrique du procédé «lean NOx», avec la participation de fabricants de moteurs et de catalyseurs, en vue d'analyser les questions soulevées et de contribuer ainsi à l'établissement de cette technologie.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

GLOSSARY

A_i	Cross sectional area of exhaust in section i [mm^2]
A_v	Area of exhaust valve opening [mm^2]
AFR	Measured air to fuel ratio
	Exhaust valve seat angle [degrees]
C_{CO}	Concentration (wet) of CO in exhaust [ppm]
C_d	Discharge coefficient of exhaust valve
C_{NOx}	Concentration (wet) of NOx in exhaust [ppm]
C_{THC}	Concentration (wet) of THC in exhaust [ppm]
CA	Crank Angle
CO	Carbon MoNOxide
D_m	Mean exhaust valve seat diameter [mm]
D_p	Exhaust port diameter [mm]
D_s	Exhaust valve stem diameter [mm]
D_v	Exhaust valve head diameter [mm]
DDC	Detroit Diesel Corporation
DDEC II	Detroit Diesel Electronic Control system (second generation)
$_j$	Phase delay of exhaust pulse from the valve to point in the exhaust [CA degrees]
	Specific heat ratio of exhaust gas
l_i	Length of exhaust section i [m]
L_v	Exhaust valve lift [mm]
	Excess air ratio
$\dot{m}_e$	Mass flow rate of exhaust gas predicted by model [g/s]
$\dot{m}_{er}$	Mass flow rate of exhaust gas predicted by refined model [g/s]
$\overline{\dot{m}}_e$	Average predicted exhaust mass flow rate per cylinder [g/s]
$\overline{\dot{m}}_{em}$	Average measured exhaust mass flow rate per cylinder [g/s]
$\dot{m}_{\mathrm{CO}}$	Mass flow rate of CO in exhaust [g/s]
$\dot{m}_{\mathrm{NOx}}$	Mass flow rate of NOx in exhaust [g/s]
$\dot{m}_{\mathrm{THC}}$	Mass flow rate of THC in exhaust [g/s]
$\hat{M}_e$	Molecular weight of exhaust gas [kg/kgmol]
N	Engine speed [rev./minute]
NOx	Oxides of Nitrogen
P_b	Barometric pressure [kPa]
P_i	Average pressure in section i of exhaust [kPa]

P_o	Stagnation pressure upstream of exhaust valve [kPa]
P_t	Static pressure at the restriction of the exhaust valve [kPa]
$_e$	Density of exhaust @ 1 atmosphere and 20°C [g/ft^3]
R_e	Specific gas constant of exhaust [J/kg-K]
t_i	Time required for exhaust pulse to traverse section i [s]
T_i	Average temperature in section i of exhaust [K]
T_o	Stagnation temperature upstream of exhaust valve [K]
TDC	Top Dead Centre
THC	Total Hydrocarbons
v_i	Velocity of exhaust in section i of the exhaust system [m/s]
w	Exhaust valve seat width [mm]
x	Carbon content of diesel fuel [$C_x H_y$]
y	Hydrogen content of diesel fuel [$C_x H_y$]

1.0 INTRODUCTION

1.1 Background

Today's heavy duty diesel engine manufacturers are continuously striving to meet ever more stringent exhaust emissions standards. While significant progress to this end has been made through the use of electronic control of diesel injection timing and duration, and the optimization of the diesel combustion process and fuel injector design, these approaches alone are not sufficient to achieve the emissions targets of the 21st century. One promising approach to further reduce exhaust emissions, particularly those of oxides of nitrogen (NOx), is exhaust aftertreatment.

Exhaust aftertreatment in the form of the three-way catalyst has been successfully used for years in light duty stoichiometrically fuelled spark ignition engines. The key to the success of this emissions control technology can be attributed to a combination of optimal catalyst formulation and precise fuel metering. Spark ignition engine exhaust aftertreatment technology has evolved over many years, and now a similar challenge faces the heavy duty diesel engine manufacturers.

The catalytic reduction of NOx in oxygen rich diesel exhaust has been demonstrated to be feasible if a suitable reducing agent is added to the exhaust stream. Because it is available, does not need a secondary reductant, and is not expensive, diesel fuel is the reductant of choice for mobile applications. ORTECH Corporation conducted a project with funding from TDC to investigate advanced NOx reduction technologies. As part of the study, ORTECH developed a reductant metering system to inject liquid diesel fuel into the exhaust upstream of lean NOx catalyst in an asynchronous, pulse width modulated fashion. After steady state engine mapping of the system, transient emissions tests were conducted to assess the effectiveness of the lean NOx catalyst and reductant system. The resulting relatively low NOx catalytic conversion efficiencies, excessive hydrocarbon (HC) slip (the portion of the introduced reductant that passes through the catalyst without recombining to benign species) and increased fuel consumption indicated that further development of lean NOx aftertreatment technology for mobile applications was required.

Areas where improvements to the lean NOx reductant technology might be realized were identified as follows:

- reductant metering system design;
- reductant injection control (amount, phasing);
- reductant introduction location, mixing, temperature and phase (gas or liquid);

- catalyst formulation and size.

Phase 1 of the proposed multi phase lean NOx technology program focused on the generation of a computer model to predict exhaust gas flow rates in a diesel engine. The ability to predict exhaust flows, pressures and temperatures will enhance the understanding of the degree of reductant/exhaust homogeneity or stratification which could be achieved by the proper control (phasing) and introduction of reductant. The results of Phase 1 are intended to help guide future phases in the development of laboratory hardware and on-engine testing to seek:

- the optimal phasing of reductant and exhaust;
- the optimal reductant mixing state (homogenous or stratified);
- the optimal reductant introduction state (liquid/gas, temperature, particle size distribution, momentum);
- the optimal reductant introduction site (port, turbocharger inlet, catalyst inlet);
- the optimal quantity of reductant.

1.2 Objectives

The specific objectives of this project were to:

- develop a computer model to predict the dynamic exhaust flow in a heavy duty diesel engine;
- generate data (flows, pressures, temperatures, emissions) using a typical 4 cylinder heavy duty diesel engine at several steady state speed/load points;
- verify and refine the exhaust flow model using the measured engine data.

2.0 LITERATURE REVIEW

A literature review was conducted to establish the state of technology in the area of diesel engine exhaust flow modeling. Although many sophisticated models exist, they are not readily applicable to different engines/configurations without a significant investment in time. It was deemed that a less sophisticated model would still provide the required information for determining the phasing of reductant needed to deliver homogeneous or stratified reductant/exhaust charge to the catalyst. Typical exhaust mass flow profiles and exhaust pressure profiles were obtained from the literature. In addition, a literature search was conducted to find methods to measure the exhaust system dynamic pressure.

3.0 DYNAMIC EXHAUST FLOW MODEL DEVELOPMENT

3.1 Approach

The flow of exhaust gas from an engine can be obtained from the flow across its exhaust valve(s). In order to compute the flow across the valve(s), information about the upstream and downstream gas conditions, and the exhaust valve flow area must be known. The exhaust gas conditions of temperature, pressure and composition will depend on the engine, operating conditions and exhaust system. The dynamic exhaust valve flow area will depend on valve geometry (diameter, seat width, seat angle) and on the valve lift as a function of crank angle position.

Valve flow area development undergoes three distinct stages as valve lift increases. The initial stage occurs as the valve begins to lift from its seat where:

$$0 \quad L_v \quad \frac{w}{\sin \quad \cos}$$

Then the first stage valve area as a function of valve lift is obtained from:

$$A_v \quad \cos \quad D_v \quad 2w \quad \frac{L_v}{2} \sin 2 \qquad (3\text{-}1)$$

Similarly, the second occurs when:

$$\frac{D_p^2 \quad D_s^2}{4D_m} \quad w^2 \quad {}^{1/2} \quad w\tan \quad L_v \quad \frac{w}{\sin \quad \cos}$$

The second stage valve area is given by:

$$A_v \quad D_m \quad L_v \quad w\tan \quad {}^{2} \quad w^2 \quad {}^{1/2} \tag{3-2}$$

The final stage occurs when:

$$\frac{D_p^2 \quad D_s^2}{4D_m} \quad w^2 \quad {}^{1/2} \quad w\tan \quad L_v$$

The final stage maximum valve flow area is given by:

$$A_v \quad \frac{}{4} \quad D_p^2 \quad D_s^2 \tag{3-3}$$

The equations governing exhaust gas flow past the valve are given by:

$$\dot{m}_e \quad \frac{C_d A_v P_o}{R_e T_o^{\;1/2}} \quad {}^{1/2} \quad \frac{2}{\quad 1}^{\;1/2 \quad 1} \tag{3-4}$$

and

$$\dot{m}_e \quad \frac{C_d A_v P_o}{R_e T_o^{\;1/2}} \quad \frac{P_t}{P_o}^{\;1/} \quad \frac{2}{\quad 1} \quad 1 \quad \frac{P_t}{P_o}^{\;1/} \quad {}^{1/2} \tag{3-5}$$

Equation (3-4) is used when:

$$\frac{P_t}{P_o} \quad \frac{2}{\quad 1}^{\;/ \quad 1}$$

The exhaust valve discharge coefficient as derived from Heywood (1) is given by:

$$Cd \quad 0.8033 \quad 3.5789\ \frac{L_v}{D_v} \quad 51.265\ \frac{L_v}{D_v}^{2} \quad 251.58\ \frac{L_v}{D_v}^{3} \quad 365.33\ \frac{L_v}{D_v}^{4} \tag{3-6}$$

The specific gas constant is computed for the exhaust gas as a function of its composition. The composition of the exhaust gas depends on the engines' operating conditions (excess air ratio - λ) and the diesel fuel's chemical composition (C_xH_y). Thus the specific gas constant for each operating condition is:

$$R_e \quad \frac{8314.6}{\hat{M}_e} \tag{3-7}$$

where the molecular weight of the exhaust gas is:

$$\hat{M}_e \quad \frac{9.08y \quad 44.01x \quad 16 \qquad 1 \;\; y/2 \quad 2x \qquad 52.68 \quad y/2 \quad 2x}{y/2 \quad x \qquad 1 \;\; y/2 \quad 2x \quad 1.88 \quad y/2 \quad 2x} \tag{3-8}$$

and the excess air ratio is:

$$\frac{\text{AFR } 12.011x \quad 1.008y}{137.9 \;\; x \quad y/4} \tag{3-9}$$

The dynamic flow rate of the exhaust emissions can then be computed using:

$$\dot{m}_{\text{NOx}} \quad \frac{\dot{m}_e C_{\text{NOx}}}{18464 \quad _e} \tag{3-10}$$

$$\dot{m}_{\text{THC}} \quad \frac{\dot{m}_e C_{\text{THC}}}{61237 \quad _e} \tag{3-11}$$

$$\dot{m}_{\text{CO}} \quad \frac{\dot{m}_e C_{\text{CO}}}{30331 \quad _e} \tag{3-12}$$

Where the density of the exhaust gas at standard conditions of temperature and pressure is given by:

$$ {}_e \quad \frac{9787.5}{R_e} \tag{3-13}$$

The valve lift profile as a function of crank angle (CA) position is specific to each engine application, and can either be obtained from the engine manufacturer or measurements. Cylinder pressure (P_o) as a function of CA and engine operating condition can be inferred by scaling a generic diesel cylinder pressure curve using the appropriate IMEP for the relative operating conditions. Similarly, the exhaust back pressure (P_t) and exhaust temperature (T_o) can be inferred for each operating by the appropriate scaling of the engine manufacturers' specified back pressures and temperatures at rated conditions. The exhaust valve geometry can also be obtained from the engine manufacturers' specifications. Steady state exhaust emissions (THC, NOx and CO) are typically available from the engine manufacturer.

The phasing of each exhaust mass flow pulse with respect to cylinder #1 TDC can be inferred by computing its flow velocity as it passes to the respective point along the exhaust system travel. The flow velocity in a given portion of the exhaust system is:

$$ v_i \quad \frac{\dot{m}_e R_e T_i}{A_i \; P_b \quad P_i} \tag{3-14}$$

The time taken to pass through each portion of the exhaust system is then given by:

$$ t_i \quad \frac{l_i}{v_i} \tag{3-15}$$

Finally, the phase delay of the exhaust pulse to reach a given point in the exhaust system in crank angle degrees from the point it leaves the exhaust valve is:

$$ {}_j \quad 6N \, {}_{i\;1}^{\;n} \, t_i \tag{3-16}$$

3.2 Assumptions

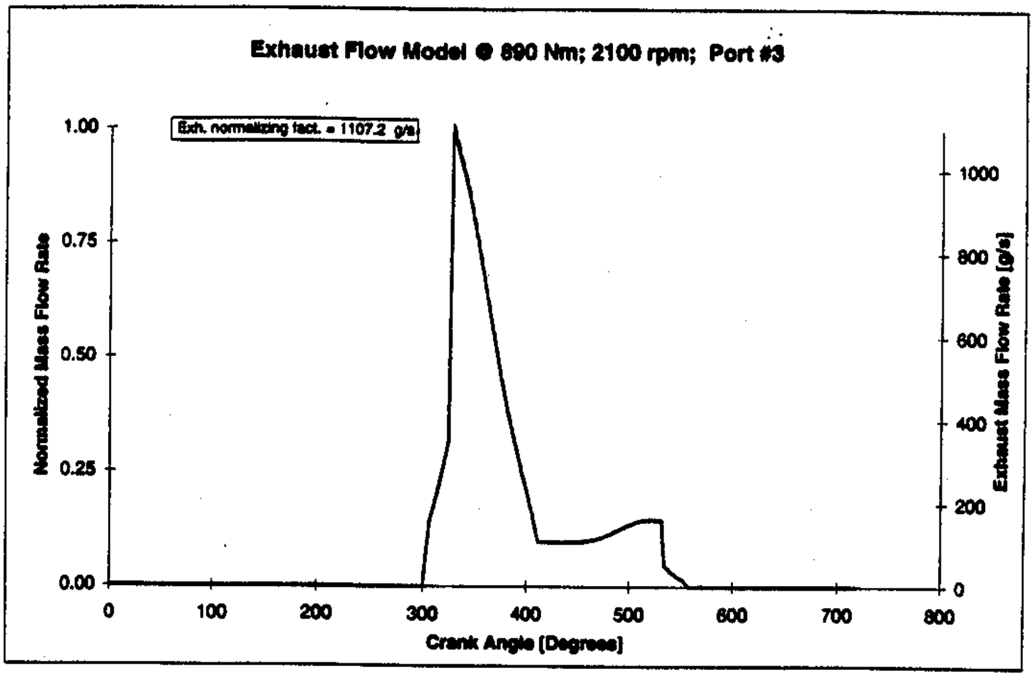
In order to simplify the dynamic exhaust flow model, several assumptions pertaining to the engine operating conditions were made. These assumptions were:

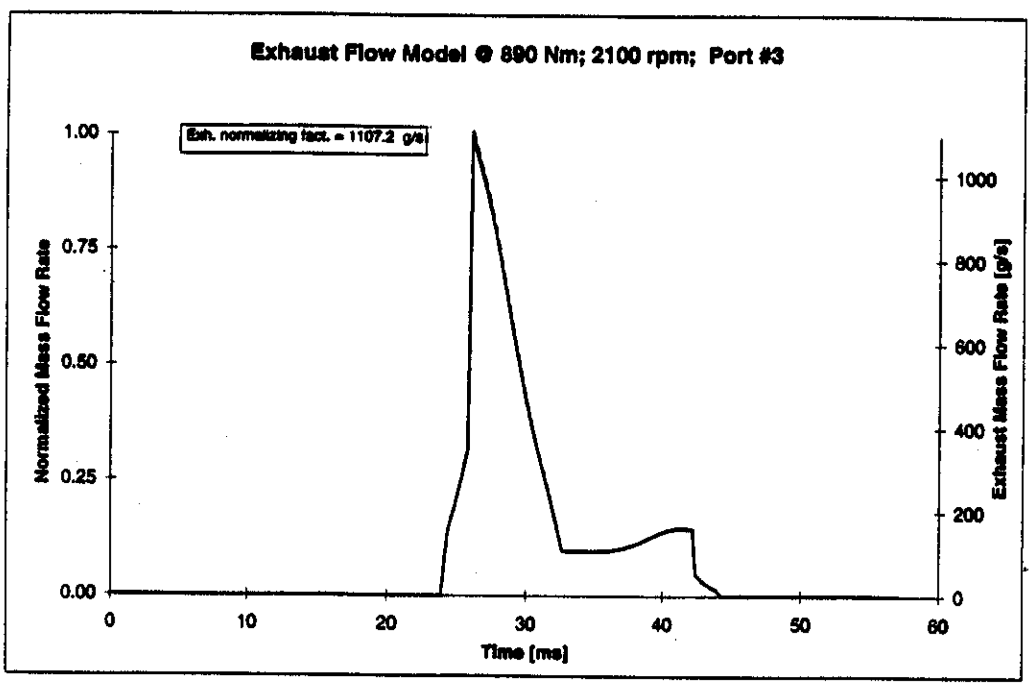
- the exhaust gas stagnation temperature (T_o) at the inlet of the exhaust valve was equal to the average downstream temperature and remained constant during the exhaust valve open period;
- the exhaust back pressure (P_t) at the exit of the exhaust valve was equal to the average exhaust system back pressure and remained constant during the exhaust valve open period;
- as the model applies to turbocharged diesel engines, exhaust tuning effects were neglected;
- the engine's exhaust emissions rate remains constant throughout the exhaust process;
- gas flow is one-dimensional, incompressible, adiabatic and frictionless in the exhaust system downstream of the exhaust valve;
- the output of each cylinder is equivalent to one quarter of the engines' total mass throughput.

3.3 Normalized Instantaneous Exhaust Mass Flow

The instantaneous exhaust mass flow rates obtained from the model were divided by the maximum predicted flow rate to give a normalized value. This permitted the graphical depiction of several sets of data having markedly different scaling. An example of the models' output at one engine operating condition is thus shown in Figure 3-1.

Figure 3-1 Normalized Exhaust Mass Flow Rate





4.0 ENGINE EXHAUST PRESSURE MEASUREMENTS

In order to provide input to the dynamic exhaust flow model, engine measurements were undertaken on a typical turbocharged heavy duty diesel engine. Measurements were made to provide the information necessary to refine the flow model, and hence, to more accurately predict exhaust flow dynamics.

4.1 Test Methodology and Apparatus

A DDC Series 50 engine was used in this test program. The engine was equipped with a DDEC II control system and conformed to 1993 US Federal emissions regulations (5.0 g/bhp-hr NOx and 0.1 g/bhp-hr particulates). Table 4-1 lists some the engines' specifications.

The engine was installed in the dynamometer test cell and instrumented to measure the following critical parameters:

- engine speed;
- torque;
- fuel flow rate;
- air flow rate;
- intake pressure;
- average exhaust pressure at the exhaust ports, turbocharger inlets, catalyst inlet and catalyst outlet;
- average exhaust gas temperature at the exhaust ports, turbocharger inlet, catalyst inlet and catalyst outlet;
- instantaneous (dynamic) static and stagnation exhaust pressure at the exhaust ports, turbocharger inlets, catalyst inlet and catalyst outlet (see Figures 4-1 and 4-2);

Table 4-1 DDC Series 50 Engine Specifications

Model	6047-GK28
Number of Cylinders	4 Inline
Bore and Stroke	130 x 160 mm
Displacement	8.5 L
Compression Ratio	15:1
No. Intake Valves per Cylinder	2
No. Exhaust Valves per Cylinder	2
Fuelling System	Direct Injection
Aspiration	Turbocharged
Rated Power	205 kW @ 2100 rpm
Rated Torque	1207 Nm @ 1200 rpm
Exhaust Flow at Rated Speed	46.7 m^3/min
Exhaust Flow at Peak Torque Speed	27.5 m^3/min
Exhaust Temperature at Rated Speed	366° C
Exhaust Temperature at Peak Torque Speed	492° C
Maximum Intake Pressure	172 kPa

In order to simulate the exhaust back pressure that might be encountered in an engine equipped with a lean NOx catalyst, a diesel oxidizing catalyst was installed just downstream of the turbocharger (see Figures 4-3 and 4.4). The oxidizing catalyst monolith was acquired from Engelhard Corporation and then canned by ORTECH. Table 4-2 gives some of the physical specifications of the exhaust catalyst.

Table 4-2 Oxidizing Catalyst Physical Specifications

Diameter	241 mm
Length	152 mm
Cell Density	200 cpi

Dynamic exhaust pressures were measured using fast response Validyne DP15 pressure transducers. In order to further improve the pressure measuring system response, the transducers were close coupled to the measurement point (typically within 200 mm). Also, in order to maintain the pressure transducers within their operational temperature limits, custom fabricated water cooled heat sinks were attached to their bodies as shown in Figure 4-5. Pitot tubes were used to measure dynamic exhaust stagnation pressures and flush wall mounted probes were used to measure dynamic exhaust pressure (Figures 4-1 and 4-2). A high speed digital storage oscilloscope was used to capture the data every three degrees of crank revolution. Several iterations of the experimental pressure measurement system design were made in an attempt to minimize system resonance (ringing). Although the ringing could not be physically eliminated completely, it was thought that post processing (Fourier analysis or deconvolving) could be used to remove the undesirable harmonics.

Raw exhaust emissions of THC, NOx, CO and O_2 were sampled upstream of the catalyst. Emission measurement of THC was made on a wet basis, while the other emissions were measured dry and then converted to wet concentrations. All measurements were made at each of the 16 points in the test matrix shown in Table 4-3.

Table 4-3 Engine Test Matrix

TEST POINT	SPEED [rpm]	POWER [kW]	TORQUE [Nm]	LOAD [%]
1	2100	205	932	100
2	2100	154	699	75
3	2100	103	350	50
4	2100	41	70	20
5	1800	205	1004	100
6	1800	154	753	75
7	1800	103	377	50
8	1800	41	75	20
9	1200	152	1207	100
10	1200	114	905	75
11	1200	76	453	50
12	1200	30	91	20
13	700	50	678	100
14	700	37	509	75
15	700	19	254	50
16	700	13	175	26

Figure 4-1 Exhaust Pressure Measurement at the Exhaust Ports and Turbocharger Inlet

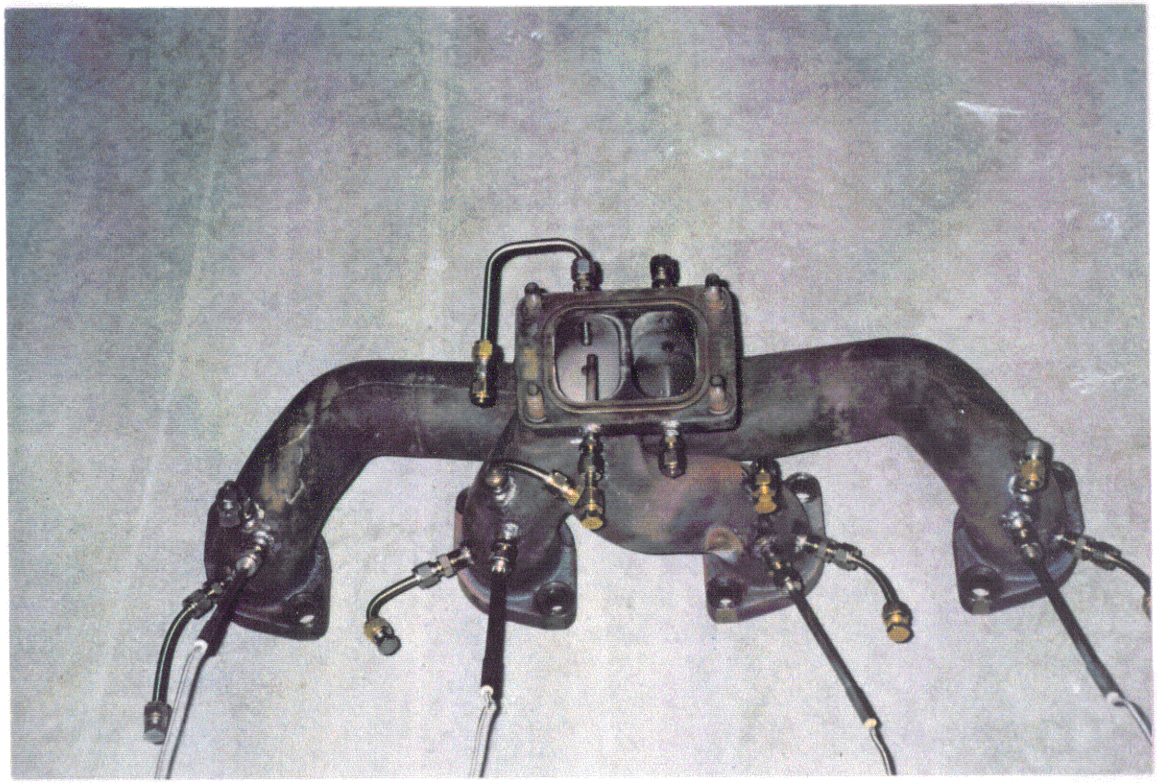


Figure 4-2 Exhaust Pressure Measurement at the Catalyst Inlet

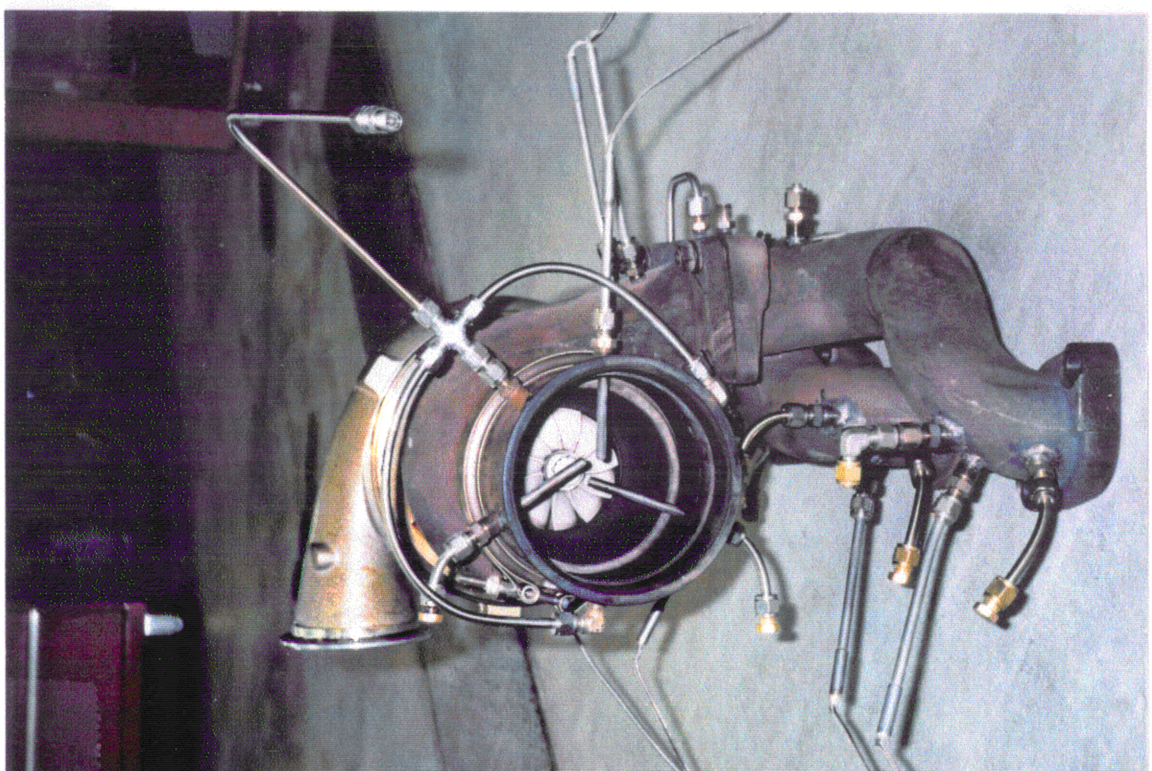
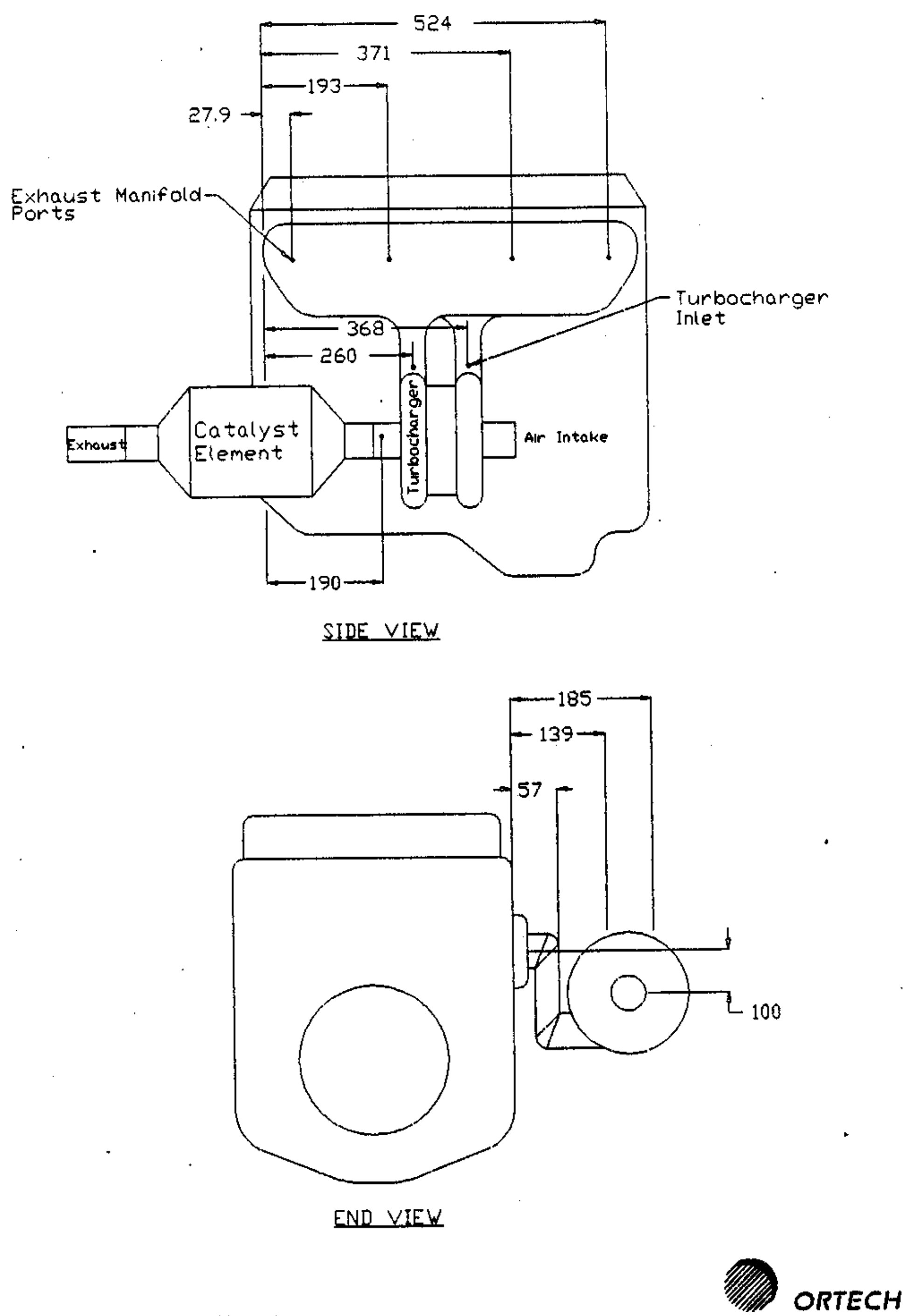


Figure 4-3 Exhaust Pressure Measurement Layout Schematic



Note: Dimensions are in Millimeters.

Figure 4-4 Exhaust Pressure Measurement System Setup

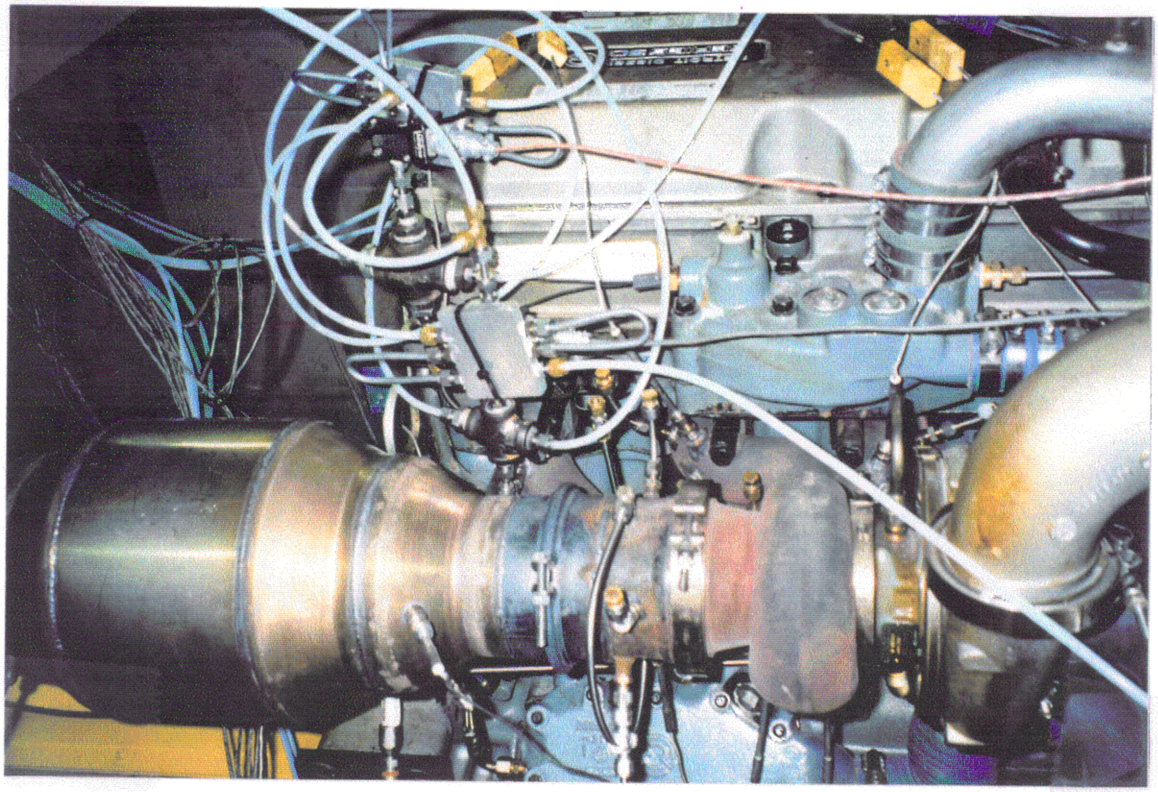
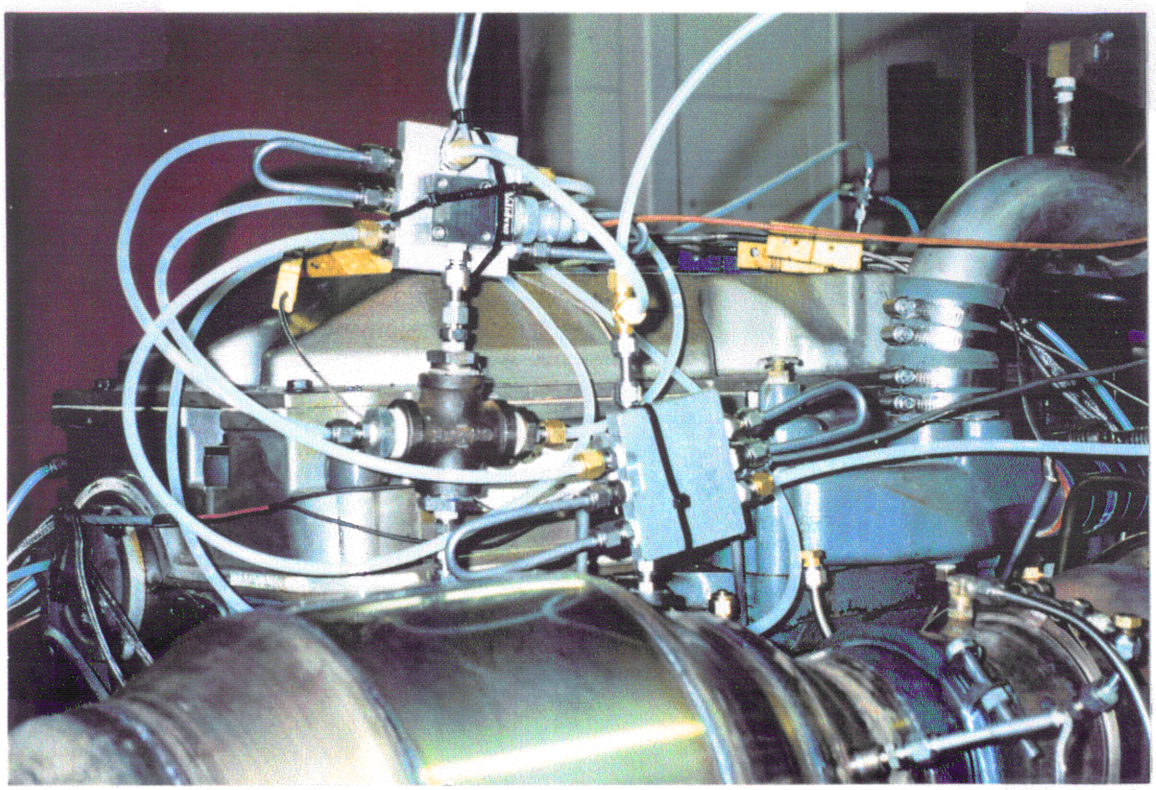


Figure 4-5 Pressure Transducer Cooling System



4.2 Results

As was discussed in Section 4.1, exhaust pressure measurements exhibited a resonant component in the pressure waveform. Post processing using Fourier analysis was unable to remove this undesirable harmonic component. This was due to the broad spectrum nature of the harmonic. Further attempts to deconvolve the pressure waveform were made using the measured response of the measurement system to a known step change in pressure. Unfortunately, this also proved unsuccessful as the system response was found to be non linear. The typical system response was 100 µs. At 2100 rpm, system response corresponded to about one degree of crank revolution. The complete set of steady state test data has been included in Appendix A.

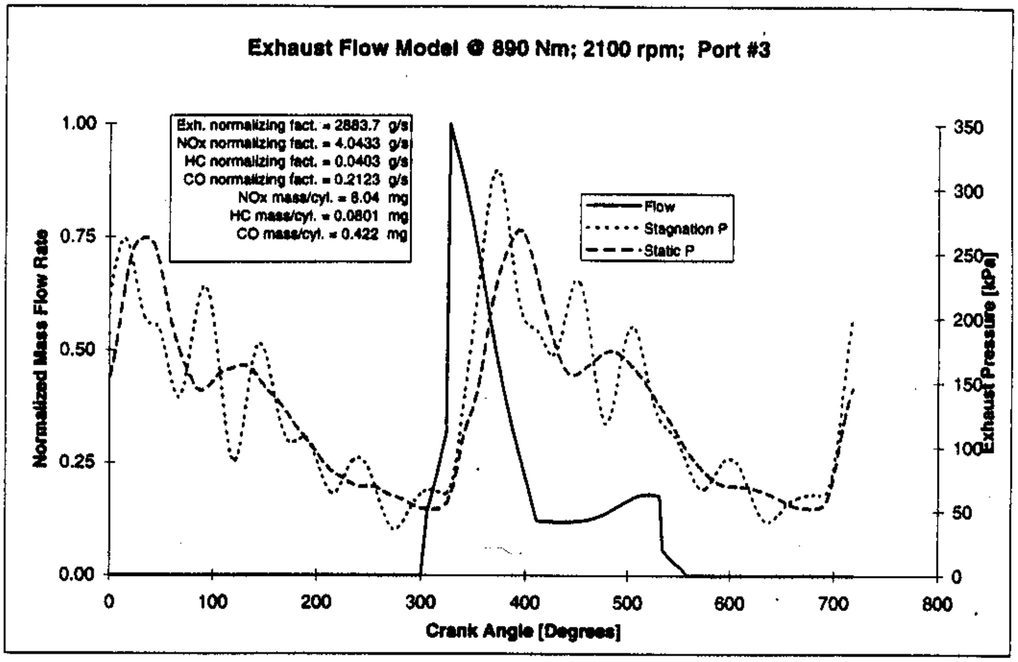
4.2.1 Exhaust Port

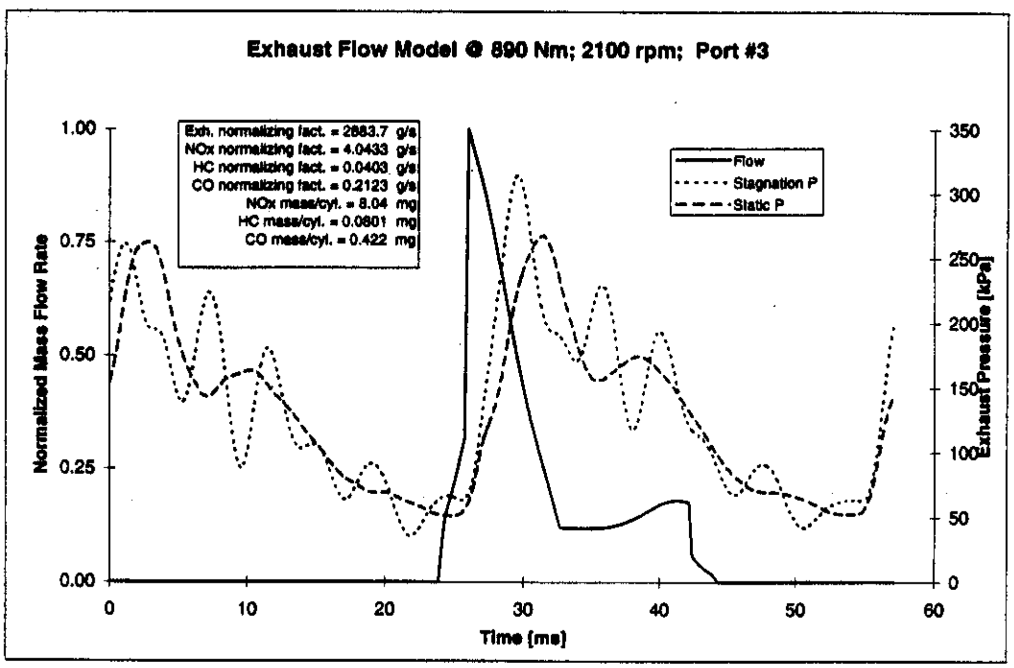
Average exhaust pressures and temperatures at the exhaust port (of cylinder #3) are given in Table 4-4. The dynamic exhaust pressures measured in the port are also plotted in Figure 4-6 for one representative operating condition. The pressure measuring system resonance is particularly evident in the stagnation pressure waveform. See Appendix B for plots of the remaining operating conditions.

Table 4-4 Average Conditions In The Exhaust Port

Test Point	Speed [rpm]	Torque [Nm]	Average Stagnation P [kPa]	Average Static P [kPa]	Average Temperature [Deg. C]
1	2100	890	132.3	129.8	542
2	2100	675	99.1	97.4	477
3	2100	450	72.5	71.3	401
4	2100	185	41.6	41.8	280
5	1800	1050	115	111.9	556
6	1800	790	84.3	83.5	496
7	1800	525	55.9	55.6	423
8	1800	215	31.4	32	275
9	1200	1180	55.2	54.1	628
10	1200	885	36.6	37.6	545
11	1200	595	23.6	24.4	429
12	1200	240	12.8	13.7	255
13	700	710	11	11.6	457
14	700	545	7.47	8.71	392
15	700	365	5.33	6.36	301
16	700	185	3.88	4.72	197

Figure 4-6 Exhaust Flows and Pressures at the Port





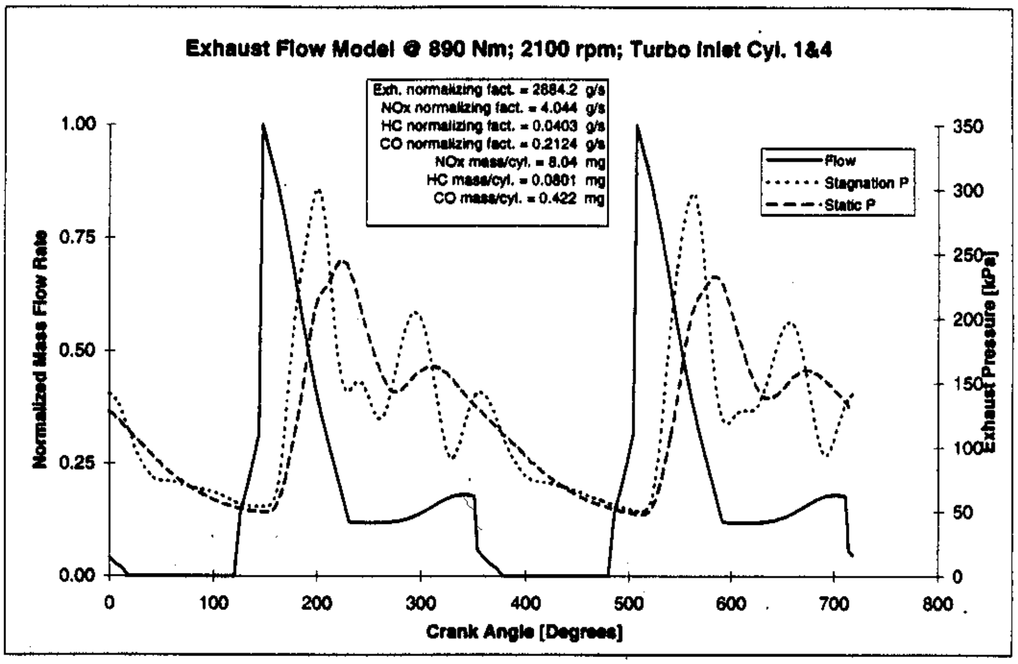
4.2.2 Turbocharger Inlet

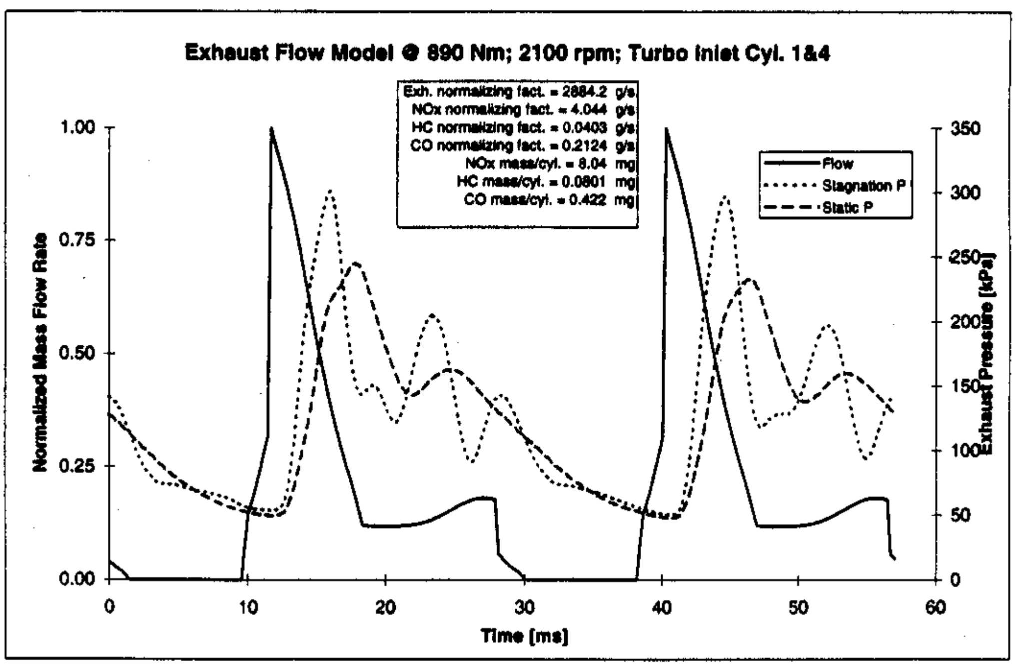
Average exhaust pressures and temperatures at the turbocharger inlet (cylinders #1 and #4) are given in Table 4-5. The dynamic exhaust pressures measured at the turbocharger inlet are also plotted in Figure 4-7 for one representative operating condition. See Appendix B for plots of the remaining operating conditions.

Table 4-5 Average Conditions At The Turbocharger Inlet

Test Point	Speed [rpm]	Torque [Nm]	Average Stagnation P [kPa]	Average Static P [kPa]	Average Temperature [Deg. C]
1	2100	890	124	122.9	561
2	2100	675	90.7	91.6	494
3	2100	450	61.7	63.3	415
4	2100	185	35.9	37.6	293
5	1800	1050	106.8	105.9	578
6	1800	790	77.3	77.2	518
7	1800	525	49.3	50.4	433
8	1800	215	26.7	28.1	290
9	1200	1180	51.5	50.1	651
10	1200	885	33.5	33.2	568
11	1200	595	19.4	19.9	449
12	1200	240	10	10.7	263
13	700	710	9.24	9.22	501
14	700	545	5.43	5.69	423
15	700	365	3.37	3.25	326
16	700	185	5.76	4.96	200

Figure 4-7 Exhaust Flows and Pressures at the Turbocharger Inlet





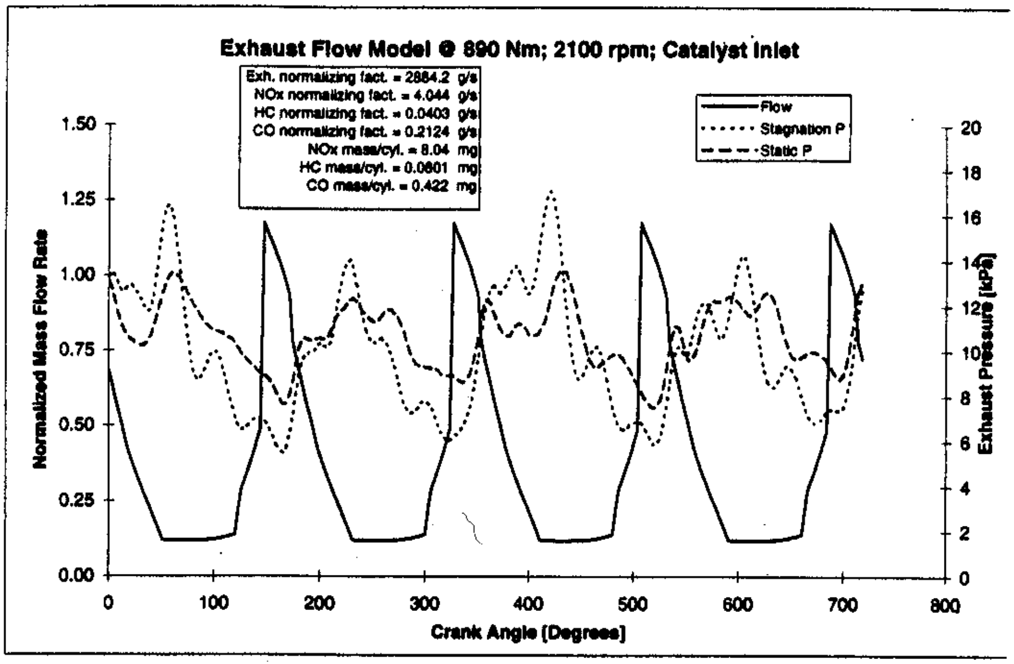
4.2.3 Catalyst Inlet

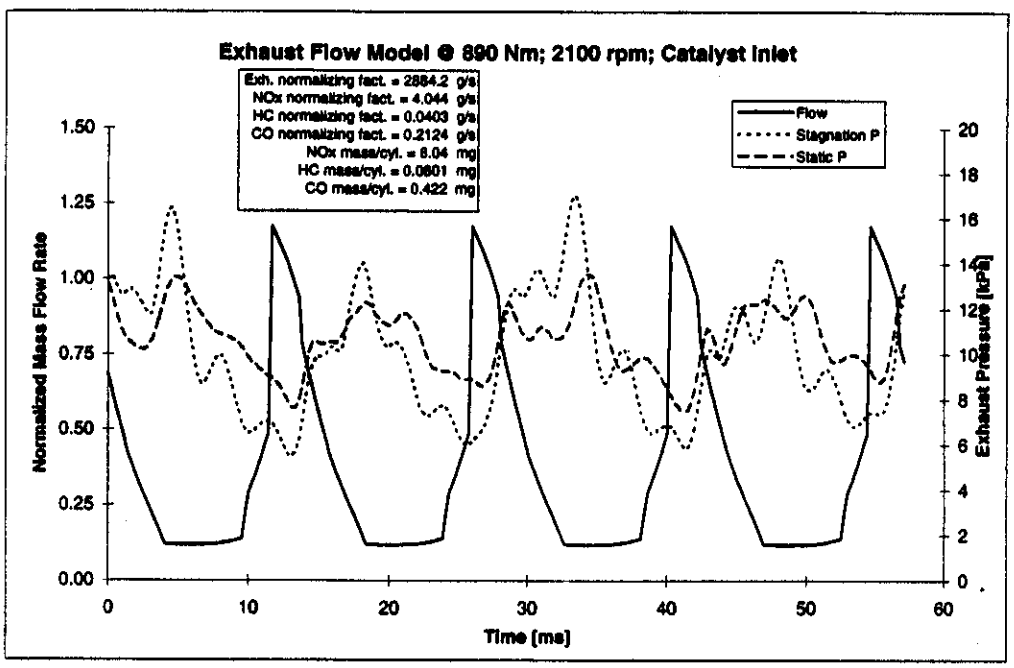
Average exhaust pressures and temperatures at the catalyst inlet are given in Table 4-6. The dynamic exhaust pressures measured at the catalyst inlet are also plotted in Figure 4-8 for one representative operating condition. See Appendix B for plots of the remaining operating conditions.

Table 4-6 Average Conditions At The Catalyst Inlet

Test Point	Speed [rpm]	Torque [Nm]	Average Stagnation P [kPa]	Average Static P [kPa]	Average Temperature [Deg. C]
1	2100	890	9.93	10.62	564
2	2100	675	6.77	7.75	503
3	2100	450	3.01	5.02	430
4	2100	185	2.11	2.83	303
5	1800	1050	7.39	8.27	577
6	1800	790	5.8	6.62	526
7	1800	525	3.57	4.36	447
8	1800	215	1.86	2.37	300
9	1200	1180	3.17	3.94	666
10	1200	885	2.22	2.77	585
11	1200	595	1.52	1.75	468
12	1200	240	0.97	1.04	287
13	700	710	0.78	0.86	503
14	700	545	0.7	0.61	435
15	700	365	0.62	0.48	331
16	700	185	0.66	0.51	224

Figure 4-8 Exhaust Flows and Pressures at the Catalyst Inlet





4.2.4 Exhaust Emissions

Table 4-7 contains the engines' exhaust emissions while operating at each of the steady state points on the test matrix.

Table 4-7 Steady State Exhaust Emissions

Test Point	Speed [rpm]	Torque [Nm]	WET NOx [ppm]	WET THC [ppm]	WET CO [ppm]	WET CO2 [%]	WET O2 [%]
1	2100	890	881	29.1	76.0	6.88	10.6
2	2100	675	826	33.9	71.1	6.12	11.9
3	2100	450	768	40.2	87.1	5.37	13.1
4	2100	185	581	67.5	292	3.76	15.7
5	1800	1050	1160	26.1	89.9	7.44	9.71
6	1800	790	948	26.8	67.3	6.59	11.2
7	1800	525	877	35.3	74.8	5.69	12.7
8	1800	215	754	69.5	246	3.89	15.5
9	1200	1180	1321	26.4	964	9.65	5.81
10	1200	885	1434	31.7	236	8.42	8.05
11	1200	595	1440	30.2	75.6	6.79	10.8
12	1200	240	882	51.0	200	4.02	15.4
13	700	710	2147	25.8	538	9.80	5.65
14	700	545	2717	26.6	110	7.93	8.79
15	700	365	2377	34.9	69.4	5.90	12.2
16	700	185	1313	43.9	170	3.66	15.9

4.3 Discussion

The measured exhaust pressures confirm that exhaust flow pulsations exist at the port, inlet to the turbocharger and inlet to the catalyst. The magnitude of the flow pulsations could be computed from the flow conditions (static and stagnation pressures, and temperature) and the known exhaust system geometry at each point. However, the undesirable resonant harmonics in the stagnation pressure signal prevented this method from being utilized. Knowing the conditions at each of the proposed NOx reductant introduction points will aid in the design of future laboratory NOx reductant metering/introduction systems.

Both the port and turbocharger inlet locations showed zero flow conditions during each engine cycle, offering an opportunity to completely stratify reductant and exhaust at these locations. However, some flow always existed at the catalyst inlet, preventing the possibility for compete reductant/exhaust stratification if reductant were introduced there.

The pressure waveform of Figure 4-6 suggests that two flow pulses occur during each engine cycle at Port #3. However, Port #3 communicates with Port #2 at the turbocharger inlet. Thus, the initial pressure pulse shown in Figure 4-6 is a result of flow in Port #2.

5.0 DYNAMIC EXHAUST FLOW MODEL VERIFICATION

5.1 Impact of Simplifying Assumptions

Exhaust gas stagnation temperature at the inlet to the exhaust valve will in reality be greater than the measured temperature in the port. This will cause the exhaust flow model to overestimate the mass flow of gas from the engine. Also, the temperature varies as a function of time, being greater initially and then decreasing.

Exhaust back pressure (downstream of the valve) does not remain constant during the exhaust process. Initially, as the valve opens, the back pressure will be lower than the average value. However, as flow increases, the back pressure also increases to a value greater than the average. The impact of this on the predicted mass flows is not obvious.

Although this engine does not employ exhaust tuning, there will be some effect at certain operating conditions. Exhaust back pressure at the valve will likely be somewhat lowered by this mechanism during the latter portion of the exhaust process (valve opening), thereby causing the model to slightly underestimate the correct exhaust flow.

Exhaust emissions rates are not constant during the entire exhaust process. This implies that the exhaust pulse from each cylinder will have associated concentration gradients and the flow model will not predict these.

The exhaust gas flow is actually compressible and not adiabatic. Thus, the flow model will not predict the change in the shape of the exhaust flow pulse as it travels through the exhaust system. That is, the model will not predict the damping effects of filling/emptying of volumes along the flow path, nor the effect of friction and flow restrictions.

Due to manufacturing tolerances and design compromises, each cylinder of a four cylinder engine will not contribute exactly one quarter of the engines' total flow throughput. Cylinder to cylinder variations are typically on the order of ±5% The flow model will not predict these variations.

5.2 Refined Exhaust Flow Model

The measured engine test data provided additional information necessary to calibrate the exhaust flow model. It was anticipated that the model would err in the prediction of total exhaust flow due to the simplifying assumptions. Thus, measured engine intake air and fuel provided a means to adjust the model's predicted exhaust flow per cylinder. The refined exhaust mass flow was then:

$$\dot{m}_{er} \quad \frac{\dot{m}_e \overline{\dot{m}}_{em}}{\overline{\dot{m}}_e} \qquad (5\text{-}1)$$

The flow model was developed in EXCEL spreadsheet format. Instructions pertaining to the use of the computer model are given in Appendix C.

As was expected, the predicted flow rates differed from the measured values. The data included actual exhaust pressure measurements at the various locations. These were also incorporated into the model prior to the previous adjustment. Figures 4-6 to 4-8 show representative (rated speed) refined model flow predictions and actual measured dynamic pressures at the exhaust port, turbocharger inlet and catalyst inlet respectively. The flow model predictions for the remaining test points are included in Appendix B. The exhaust flow profiles have not been adjusted for phase delays to allow for visualization of the delay. The predicted phase delays at each of the operating conditions are given in Table 5-1.

5.3 Exhaust Homogeneity/Stratification

One of the purposes of the dynamic exhaust flow model is to help predict the degree of homogeneity or stratification that could be achieved by a reductant metering system when the reductant is introduced at a given location in the exhaust. Table 5-2 summarizes the potential introduction windows at the various locations and the degree of homogeneity or stratification that could be achieved by an appropriately configured synchronous reductant metering system.

Table 5-1 Exhaust Flow Phase Delay

Test Point	Speed [rpm]	Torque [Nm]	Average Mass Flow Per EVO [g/s]	Predicted Phase Delays: Valve to Port [CA Deg.]	Predicted Phase Delays: Valve to Turbo Inlet [CA Deg]	Predicted Phase Delays: Valve to Catalyst Inlet [CA Deg.]
1	2100	890	282.1	8.5	37.6	79.2
2	2100	675	236.7	9.5	41.8	94.2
3	2100	450	188.3	11.5	50.1	121.2
4	2100	185	140.3	15.6	68.6	182.3
5	1800	1050	250.8	7.5	32.9	71.5
6	1800	790	205.9	8.5	37.1	86.7
7	1800	525	160.9	10.2	44.8	113.9
8	1800	215	117.8	14.9	65.7	181.9
9	1200	1180	141.4	5.9	26.0	66.1
10	1200	885	113.8	7.2	31.6	85.5
11	1200	595	89.2	9.7	42.3	121.3
12	1200	240	70.9	14.8	65.5	197.2
13	700	710	45.7	9.5	41.6	126.0
14	700	545	41.7	11.1	48.7	150.3
15	700	365	38.6	13.6	59.6	187.5
16	700	185	37.1	17.0	77.5	242.6

Table 5-2 Potential Opportunities For Exhaust and Reductant Mixing or Stratification

	Time Available to Produce a Homogeneous Reductant/Exhaust Mixture at the Port			Time Available to Produce a Homogeneous Reductant/Exhaust Mixture at the Turbocharger Inlet		
Speed [rpm]	**Time [ms]**	**CA [Degrees]**	**Frequency [Hz]**	**Time [ms]**	**CA [Degrees]**	**Frequency [Hz]**
2100	20.32	256	17.5	20.32	256	35
1800	23.70	256	15	23.70	256	30
1200	35.56	256	10	35.56	256	20
700	60.95	256	5.83	60.95	256	11.67

	Time Available to Produce a Stratified Reductant/Exhaust Mixture at the Port			Time Available to Produce a Stratified Reductant/Exhaust Mixture at the Turbocharger Inlet		
Speed [rpm]	**Time [ms]**	**CA [Degrees]**	**Frequency [Hz]**	**Time [ms]**	**CA [Degrees]**	**Frequency [Hz]**
2100	36.83	464	17.5	16.51	208	35
1800	42.96	464	15	19.26	208	30
1200	64.44	464	10	28.89	208	20
700	110.48	464	5.83	49.52	208	11.67

			Time Available to Produce a Partially Homogeneous Reductant/Exhaust Mixture at the Catalyst Inlet			
Test Point	**Speed [rpm]**	**Torque [Nm]**	**Time [ms]**	**CA [Degrees]**	**Frequency [Hz]**	**% Total Exhaust Flow in This Phase**
1	2100	890	7.94	100	70	74.2
2	2100	675	7.94	100	70	73.7
3	2100	450	7.94	100	70	73.2
4	2100	185	7.94	100	70	72.8
5	1800	1050	9.26	100	60	74.1
6	1800	790	9.26	100	60	73.4
7	1800	525	9.26	100	60	72.9
8	1800	215	9.26	100	60	72.8
9	1200	1180	13.89	100	40	75.2
10	1200	885	13.89	100	40	74
11	1200	595	13.89	100	40	73.1
12	1200	240	13.89	100	40	73
13	700	710	23.81	100	23.33	69.8
14	700	545	23.81	100	23.33	69.8
15	700	365	23.81	100	23.33	69.8
16	700	185	23.81	100	23.33	69.8

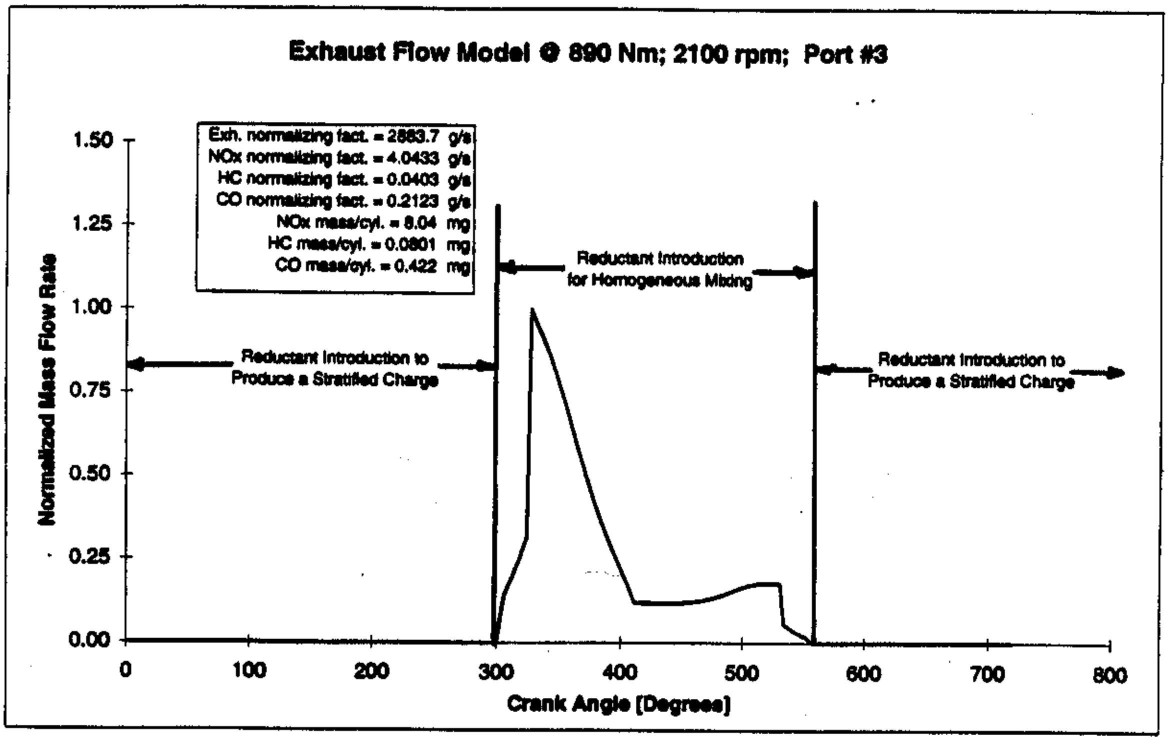
Table 5-2 continued

			Time Available to Produce a Partially Stratified Reductant/Exhaust Mixture at the Catalyst Inlet			
Test Point	Speed [rpm]	Torque [Nm]	Time [ms]	CA [Degrees]	Frequency [Hz]	% Total Exhaust Flow in This Phase
1	2100	890	6.35	80	70	25.8
2	2100	675	6.35	80	70	26.3
3	2100	450	6.35	80	70	26.8
4	2100	185	6.35	80	70	27.2
5	1800	1050	7.41	80	60	25.9
6	1800	790	7.41	80	60	26.6
7	1800	525	7.41	80	60	27.1
8	1800	215	7.41	80	60	27.2
9	1200	1180	11.11	80	40	24.8
10	1200	885	11.11	80	40	26
11	1200	595	11.11	80	40	26.9
12	1200	240	11.11	80	40	27
13	700	710	19.05	80	23.33	30.2
14	700	545	19.05	80	23.33	30.2
15	700	365	19.05	80	23.33	30.2
16	700	185	19.05	80	23.33	30.2

Thus, a properly configured synchronous reductant metering system should be able to produce either a 100% homogeneous reductant/exhaust mixture or a 100% stratified charge at the ports and turbocharger inlet. Figures 5-1 and 5-2 illustrate some of the synchronous reductant metering options at these locations. However, as the exhaust flow never drops to zero, complete mixing or stratification is not possible at the catalyst inlet. Here, approximately 75% of the exhaust flow from each cylinder occurs in a 100° CA window separated by a rarefaction (25% of exhaust flow) lasting 80° CA. Figure 5-3 shows some of the reductant metering options at the catalyst inlet.

The reductant introduction configurations depicted in Figures 5-1 to 5-3 are not the only strategies that could be employed in a lean NOx reduction system. The dynamic exhaust flow model will aid in the development of synchronous reductant metering strategies which provide the catalyst with the correctly prepared and phased feedgas mixture for optimum NOx reduction performance.

Figure 5-1 Reductant Metering Options at the Exhaust Port



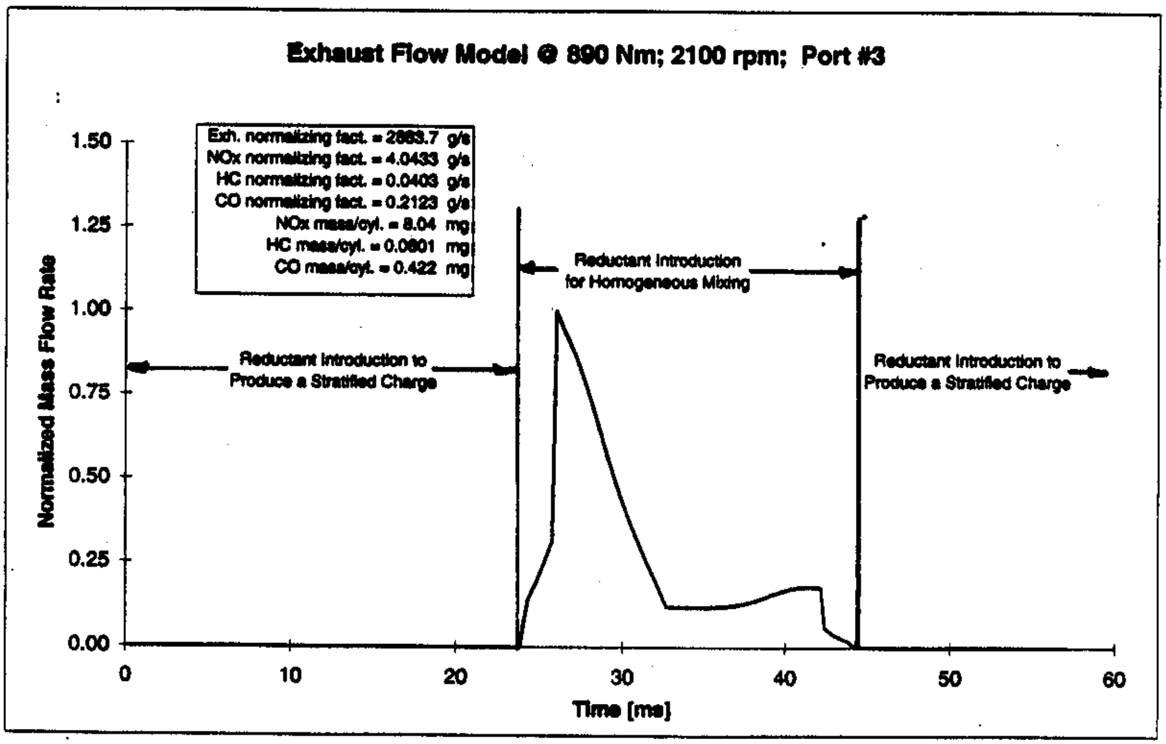
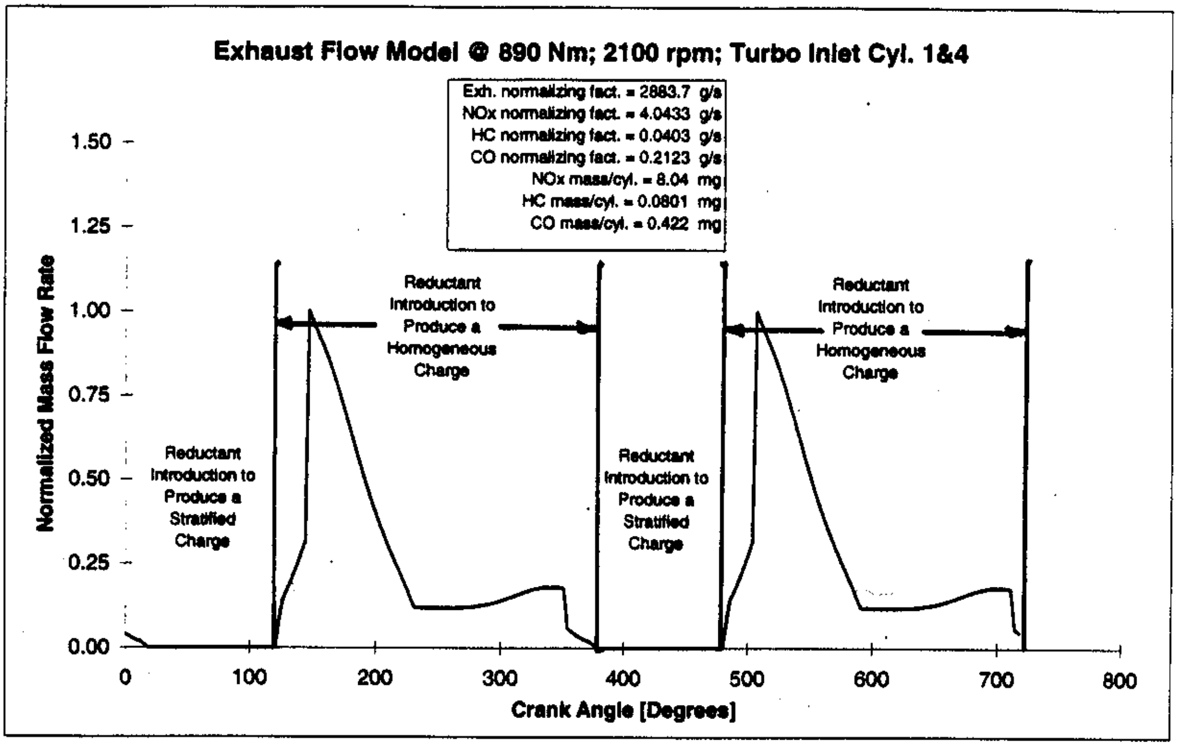


Figure 5-2 Reductant Metering Options at the Turbocharger Inlet



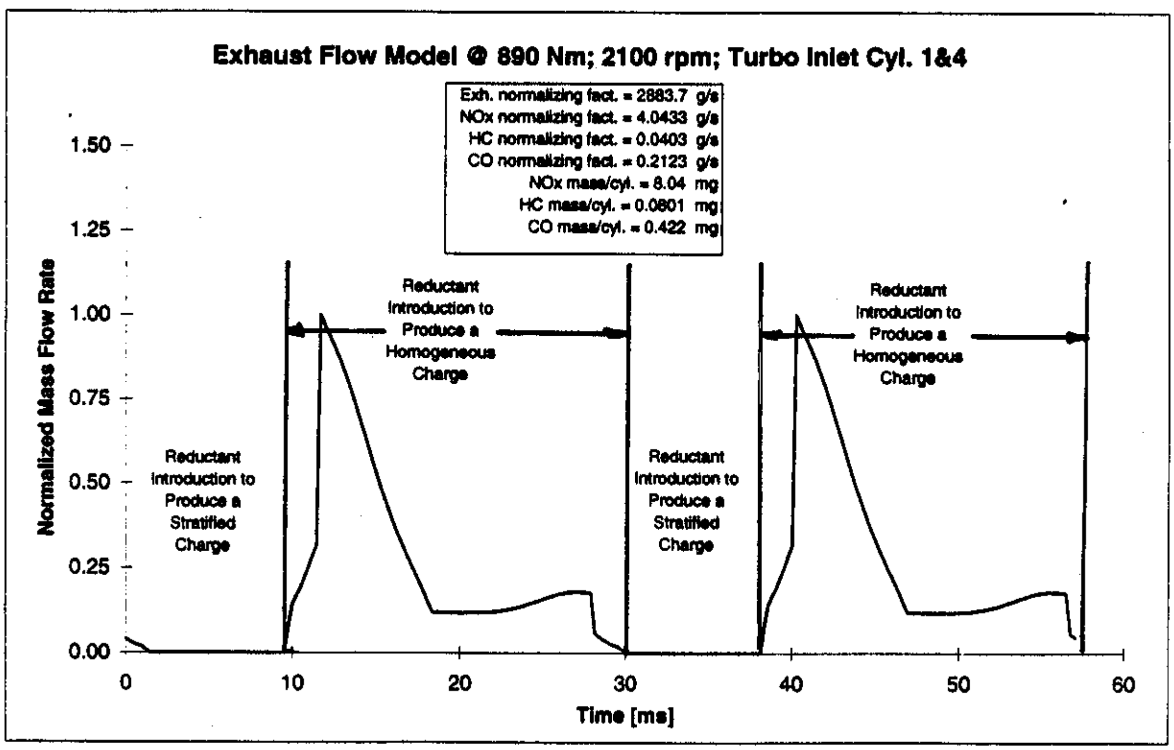
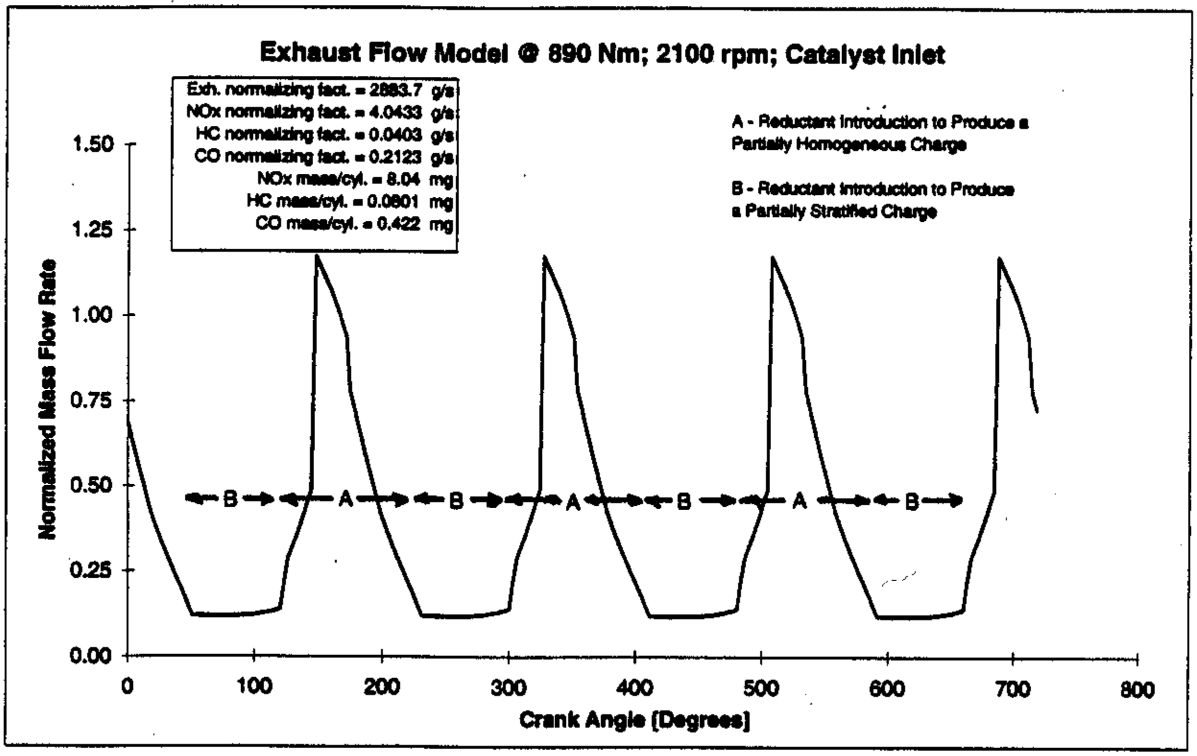
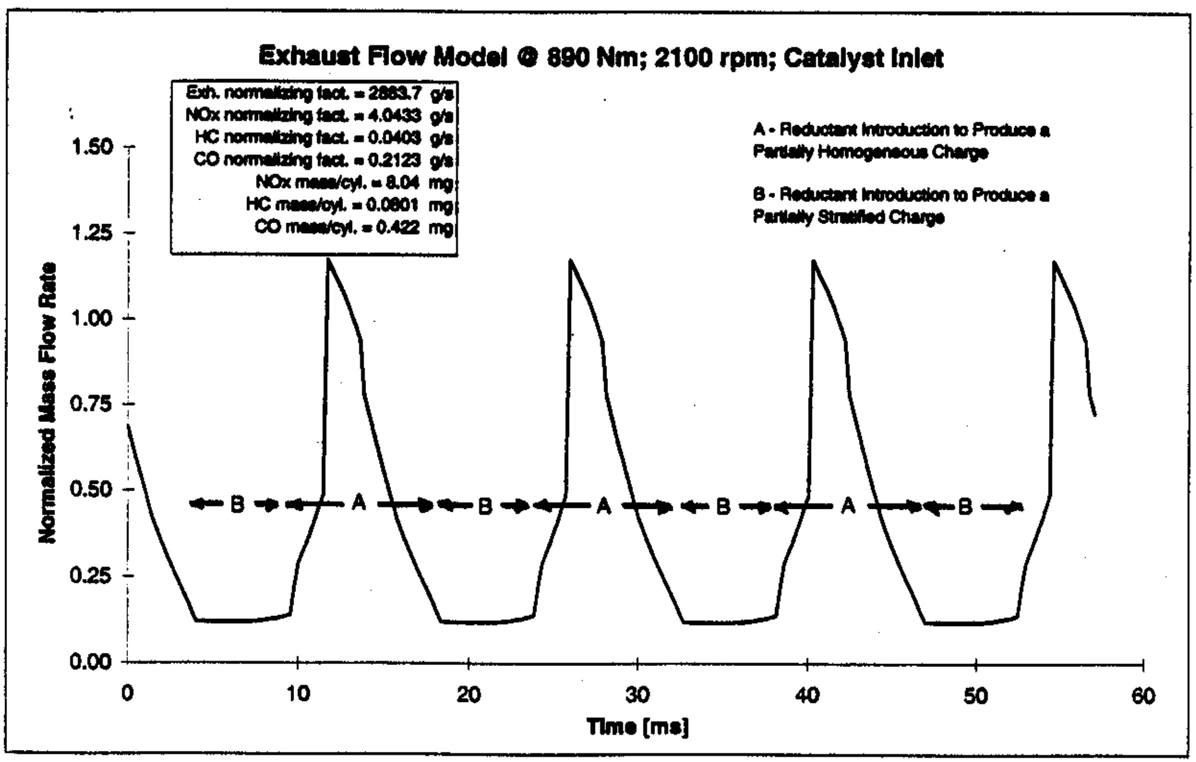


Figure 5-3 Reductant Metering Options at the Catalyst Inlet





6.0 CONCLUSIONS AND RECOMMENDATIONS

Diesel exhaust aftertreatment in the form of lean NOx catalysis requires that the exhaust and reductant feedgas to the catalyst be properly prepared. The dynamic exhaust flow model and engine test data developed during this phase of a multi-phase program predicts exhaust flow rates, pressures and temperatures. Future phases will use this information to develop and optimize synchronous reductant metering systems and strategies (homogeneous/stratified, phasing, quantity of reductant), and to evaluate reductant introduction methods (location, mixing, liquid/gas, particle size).

The specific outcomes of this program were:

- completion of a dynamic exhaust flow model to predict instantaneous flow rates in the exhaust of a turbocharged, 4-cylinder diesel engine;
- refinement and calibration of the flow model using engine test data;
- measurement of exhaust pressures and temperatures at several locations;
- calibration of the model to predict instantaneous mass emissions flow rates of NOx, HC and CO at the exhaust ports, turbocharger inlets and catalyst inlet;
- model predictions that 100% exhaust/reductant stratification or 100% homogeneity are possible when the reductant is introduced synchronously at the exhaust ports or turbocharger inlets;
- confirmation that zero exhaust flow conditions do not exist at the catalyst inlet, and hence, complete stratification or homogeneous mixing is not possible when reductant is introduced there.

Future work should include a “Lean NOx Parametric Study”, with continuing participation of engine and catalyst manufacturers, to investigate system issues and to identify:

- the optimal phasing of reductant and exhaust;
- the optimal reductant mixing state (homogenous or stratified);
- the optimal reductant introduction state (liquid/gas, temperature, particle size distribution, momentum);
- the optimal reductant introduction site (port, turbocharger inlet, catalyst inlet);
- the optimal quantity of reductant.

In order to have the greatest utility, the dynamic flow model would also be adapted to 6 cylinder and larger engines.

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REFERENCES

1. Heywood, J.B., Internal Combustion Engine Fundamentals, McGraw-Hill Book Company, 1988

APPENDIX A

ENGINE EXHAUST PRESSURE MEASUREMENT DATA

(3 Pages)

(Not available in electronic format /
Non disponible en format électronique)

APPENDIX B

REFINED DYNAMIC EXHAUST FLOW MODEL

(48 Pages)

(Not available in electronic format /
Non disponible en format électronique)

APPENDIX C

INSTRUCTIONS FOR THE USE OF THE EXHAUST FLOW MODEL

(5 Pages)

(Not available in electronic format /
Non disponible en format électronique)