

Diesel Engine Size Scaling at Medium Load without EGR

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ABSTRACT

Several diffusion combustion scaling models were experimentally tested in two geometrically similar single-cylinder diesel engines with a bore diameter ratio of 1.7. Assuming that the engines have the same in-cylinder thermodynamic conditions and equivalence ratio, the combustion models primarily change the fuel injection pressure and engine speed in order to attain similar performance and emissions. The models tested include an extended scaling model, which scales diffusion flame lift-off length and jet spray penetration; a simple scaling model, which only scales spray penetration at equal mean piston speed; and a same speed scaling model, which holds crankshaft rotational velocity constant while also scaling spray penetration.

Successfully scaling diffusion combustion proved difficult to accomplish because of apparent differences that remained in the fuel-air mixing and heat transfer processes. A computational investigation revealed that the larger nozzle diameter (relative to the ideal scaled value) used in the small engine experiments caused spray-wall impingement, which altered the mixing and subsequent combustion event. Similar pressure profiles and combustion could be achieved by increasing the injection pressure of the small engine relative to the injection pressure predicted by the scaling models. Large differences in soot and carbon monoxide emissions suggest dissimilarities in fuel-air mixing, late-cycle oxidation, and heat transfer. Computational and experimental results showed that swirl ratio had a large effect on mixing and emissions, and is an important parameter in engine scaling. Finally, a new scaling relation was derived in which engine speed, injection pressure, and injection duration are held constant.

INTRODUCTION

In fluid mechanics, similitude generally applies to scale-size models inserted in a flow to predict the performance of an actual-size prototype. Two systems are judged physically similar when the conditions for geometric, kinematic, and dynamic similarity have been met [1]. Engineers have considered how to apply the theory of similitude to engine design for over a century [2]. Ideally, the key design features from one engine platform could be scaled up or down to meet the requirements of other engine sizes. Unfortunately, there are many challenges to achieving similarity in engine systems. Inertial forces and material limitations impose size and rotational speed constraints. Small engines suffer from proportionally larger heat transfer losses, which limit efficiency. Scaling of diesel engines further confounds the problem because power is proportional to the combustion rate, combustion is limited by the fuel-oxidizer mixing rate, and mixing does not scale simply with engine speed. Successful scaling of combustion in diesel engines would accurately predict performance and emissions, guiding design for a wide range of engine sizes.

DIFFUSION FLAME STRUCTURE

Understanding of diesel engine combustion and emission formation has significantly advanced in recent years due to the use of optical diagnostic techniques. Synthesizing the work of many studies, Dec [3] proposed a time-sequenced phenomenological model of direct-injection diesel combustion covering the duration of an injection event. Particularly noteworthy was Dec's hypothesis that a standing fuel-rich premixed flame is established downstream of a diesel jet's liquid length. Dec argued that the products from the premixed flame provide the fuel species for the lifted diffusion flame at the stoichiometric location surrounding the

jet, and that the premixed flame products are the inception point for soot at the head vortex.

Pickett *et al.* [4] refined Dec's conceptual model by investigating soot formation in diesel flames in an optically accessible constant-volume combustion chamber. They verified the presence of a standing partially premixed reaction zone, but demonstrated that it occurs well upstream of the location of first soot formation. Furthermore, they reported that enhanced air entrainment upstream of the diffusion flame lift-off length can minimize or even eliminate soot formation inside the diesel jet. Advanced technology diesel engines accomplish this by using smaller orifice diameter injector nozzles and higher fuel injection pressures. With these insights, Pickett *et al.* updated Dec's conceptual model (Figure 1) to show how fuel properties and air entrainment upstream of the lift-off length affect combustion and soot formation.

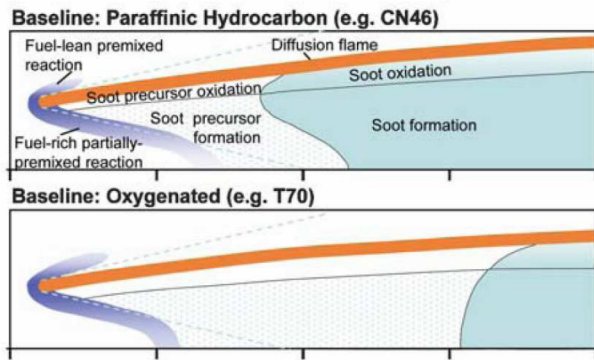


Figure 1. Pickett *et al.*'s [4] revised diesel diffusion flame model

DIESEL ENGINE SCALING

The diffusion flame concepts introduced above highlight the key physical features of the flame that the combustion scaling models attempt to scale: spray penetration, liquid length, and lift-off length. As explained in several sources [5, 6, 7, 8], dimensional analysis of the continuity equation and correlations for spray penetration and lift-off length yield scaling parameters for the operating conditions predicted to achieve scaled combustion. (The reader is referred to the aforementioned sources for more detailed information on deriving the scaling models.) The simple scaling model scales fuel mass and spray penetration, while the extended scaling model further scales lift-off length. The resulting scaling parameters are listed in Table 1. Consistent with literature [5, 6, 7, 8, 11], the bore ratio is defined as the ratio 'L' by which all design parameters should scale. For the present study, $L = 137.2 \text{ mm}/82.0 \text{ mm} = 1.7$.

Table 1. Scaling parameters

Parameter	Simple Scaling	Extended Scaling	Same Speed Scaling
Fuel mass (m_f)	L^3	L^3	L^3
Spray penetration (s)	L^1	L^1	L^1
Lift-off length (H)	N/A	L^1	N/A
Injector orifice diameter (d_o)	L^1	L^1	L^1
Injection pressure (Δp_{inj})	L^0	$L^{4/3}$	L^2
Engine rotational speed (N)	L^{-1}	$L^{-1/3}$	L^0
Time-based injection duration (t)	L^1	$L^{1/3}$	L^0
Crank-angle injection duration	L^0	L^0	L^0
Mean piston speed (S_p)	L^0	$L^{2/3}$	L^1

According to the correlation (Equation (1)) proposed by Siebers [9], the liquid length (LL) is predicted to scale linearly with injector orifice diameter (d_o), assuming all other parameters are equal. Thus the liquid length scales by the scaling factor L for the simple and extended scaling models as well.

$$LL = c_2 \sqrt{\frac{\rho_f}{\rho_a} \frac{\sqrt{C_d} d_o}{\tan(\theta_s/2)}} \sqrt{\left(\frac{2}{(F/A)_{liq}} + 1\right)^2 - 1} \propto d_o \approx L \quad (1)$$

Much research [5, 6, 7, 8, 10, 11] has been published on diesel engine size scaling, particularly exploring the simple and extended scaling models proposed by Stager and Reitz [5]. Staples *et al.* [7] executed an experimental investigation of the extended scaling parameters upon which the current work is based. In their research, performance metrics such as indicated mean effective pressure (IMEP) and indicated specific fuel consumption (ISFC) showed good agreement across a range of speeds, injection timings, and initial thermodynamic conditions. Engine-out soot and nitrogen oxides (NO_x) emissions also agreed fairly well, despite bulk gas temperature differences and scaling constraints on engine kinematic timescales. Cylinder pressure was greater in the small engine, however, signifying that scaling of conventional diesel diffusion-type combustion had not been fully realized. Staples *et al.* [7] hypothesized that discrepancies in a spray parameter caused faster fuel vaporization in the small engine, giving faster combustion and higher cylinder pressure.

Lee *et al.* [8] performed a computational study investigating the sensitivity of various scaling parameters (engine speed, fuel injection pressure, nozzle orifice diameter, injection duration, compression ratio, intake valve closing, squish height, and effective compression ratio) to engine out emissions. For their cases the extended scaling model very

accurately predicted combustion and emissions trends for an up-scaled engine (scaling ratio ~ 1.2) operating in a conventional diesel regime. Furthermore, Lee *et al.* [8] proposed a new scaling correlation in which engine speed remained constant and injection velocity scaled by L^1 . They concluded that injection duration was the most sensitive parameter for scaling combustion and emissions.

Puntambekar [10] conducted a combustion and spray visualization study in an optical engine with similar geometry to the small engine used in the work of Staples *et al.* [7]. Puntambekar did not observe strong agreement between his results and the Hiroyasu [12] spray penetration correlation used in the scaling models, hypothesizing that the reason for the discrepancy was due to a ramped rate of fuel injection. Additionally, for the conditions that Puntambekar investigated, he observed that spray penetration was controlled by injection conditions for a longer time period than predicted by Hiroyasu's expression for initial-stage penetration. This may have implications for the scaling models because they assume the break-up time is negligible compared to the injection duration.

In summary, the current research was motivated by the goal of continuing the experimental study on diesel engine size scaling started by Staples *et al.* [7] in order to further evaluate the proposed scaling laws at low speed, low injection pressure operating conditions. Towards that end, this research implemented similar common-rail diesel injector technology in both engines to eliminate variability in the injection systems.

EXPERIMENTAL AND COMPUTATIONAL SETUP

ENGINES

The small engine used in this experiment is a 4-valve, dual overhead cam, direct injection single-cylinder research engine. The major components consist of a Ricardo Hydra crankshaft and crankcase, a custom cylinder block, and an aluminum cylinder head from a General Motors (GM) 1.9 liter 4-cylinder high-speed direct-injection diesel engine. Note that the piston material is aluminum and cooled by an oil jet. The swirl ratio was lowered by use of adjustable vanes upstream from the intake ports. Pressurized and temperature-controlled inlet air was used to simulate the boost capabilities of a turbocharger. Cylinder pressure measurements were sampled every 0.1 crank angle with a Kistler model 6125A cylinder pressure transducer, and averaged for 499 cycles to generate a raw pressure trace. Engine exhaust emissions were sampled from the exhaust surge tank and measured with a Horiba emissions bench. Soot measurements were obtained from the classic filter paper method of the AVL-415S smoke meter. The GM engine was the base engine from which to

scale up to the large engine, thus no major modifications were made for the scaling experiments.

The large engine is a Caterpillar (Cat) 4-valve, single overhead cam, direct injection single-cylinder oil test engine. Several modifications were made to the Cat engine so it would be similar to the small engine. Modifications from the production configuration include a shroud over the intake valves [7] to increase swirl, custom camshaft with late intake valve closing (IVC) timing, custom piston with scaled bowl geometry [7], and a Bosch common rail fuel injection system. The Cat engine's IVC timing was delayed to compensate for its higher geometric compression ratio, and to be consistent with previous research [7]. Similar to the GM engine, pressurized and temperature-controlled inlet air was used to simulate the boost capabilities of a turbocharger. Cylinder pressure measurements were sampled every 0.25 crank angle with a water-cooled Kistler 6067C1 pressure transducer, and averaged for 499 cycles to generate a raw pressure trace. Also similar to the GM engine laboratory, engine exhaust emissions were sampled from the exhaust surge tank and measured with a Horiba emissions bench; and soot was measured with an AVL-415S smoke meter.

Ideally, all geometric parameters, such as displacement, compression ratio, and piston bowl shape, would be perfectly scaled. In this research, however, some of the dimensions, such as stroke and displacement, are constrained. Table 2 lists the dimensions of the Cat and GM engines, and compares the ideally-scaled geometric scaling factor to the actual scaling factor (using the 1.7 bore ratio as the length scaling factor L) calculated from the engines' relevant dimensions. The computational study by Shi and Reitz [6] showed the importance of scaling the top dead center (TDC) geometry for conventional diesel operation, which was achieved in the current study by using scaled-geometry pistons.

Table 2. Geometric similarity of scaled engines

Dimension	Unit	Geometric Scaling Factor	Large Engine (Cat)	Small Engine (GM)	Actual Scaling Factor
Bore	mm	L^1	137.2	82.0	L^1
Stroke	mm	L^1	165.1	90.4	$L^{1.2}$
Connecting rod	mm	L^1	261.6	145.5	$L^{1.1}$
Displacement	liter	L^3	2.44	0.48	$L^{3.2}$
TDC volume	cc	L^3	168.2	33.5	$L^{3.1}$
Bowl volume	cc	L^3	110.8	23.9	L^3
Squish height	mm	L^1	2.2	1.3	L^1
Swirl ratio	-	L^0	1.38	1.55	$L^{-0.2}$
IVC (aTDC)	deg	L^0	-85	-132	$L^{-0.9}$
EVO (aTDC)	deg	L^0	130	112	$L^{0.3}$

Significant pressure trace oscillations were observed in the Cat engine when using the Bosch injector in a conventional diesel combustion regime. Raw pressure traces were examined at various conditions to further analyze and isolate the problem. It was determined that the oscillations were due to combustion chamber resonance [13] excited by the higher pressure rise rates from the fast initial rate of injection (ROI) of the Bosch injector. Therefore, the raw pressure traces were filtered with a low-pass finite impulse response filter using 101 coefficients. In the interest of consistency in comparing pressure traces, the same filtering technique was applied to the GM engine pressure data. Heat release rates (HRR) were calculated from Equation (2) using the filtered pressure data, modeling the cylinder gases as a single, uniform zone. Assumptions in deriving the heat release equation from an energy balance on the cylinder contents include constant mass, molecular weight, and wall temperature (400 K); ideal gas; and a correlation for temperature-dependent ratio of specific heats (γ) from Gatowski *et al.* [14]. Contributions to trapped mass by the exhaust gas residual were estimated using the correlation of Yun and Mirsky [15]. The heat transfer coefficient (h) was estimated using Anand's convective heat transfer correlation relating the Nusselt number to the Reynolds number [16].

$$\dot{Q}_{ch} = \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + h A_s (T - T_w) \frac{dt}{d\theta} \quad (2)$$

INJECTORS

The current study utilized Bosch common rail injectors with similar nozzle geometry in order to minimize the influence of geometry on fuel vaporization, which was one of the factors noted by Staples *et al.* [7] as a source of potential dissimilarity in their work. Table 3 lists the specifications for the Cat and GM engine injectors. Note that with each of the three scaling models proposed thus far (simple, extended, and same speed), orifice diameter should scale by L . Although the orifice diameter of the GM injector was slightly oversized for scaling purposes (scaling down by a factor L from the Cat injector yields a 0.149 mm orifice diameter versus 0.164 mm for the injector used), research [17] suggests that the vco configuration helped compensate for this by increasing cavitation, thus lowering the discharge coefficient.

Table 3. Injector specifications

Parameter	Cat	GM
Injector Holder	Bosch CRIN 2	Bosch CRIN 1
Nozzle Type	sac	vco
Sac Volume (mm ³)	0.7	0.0
Number of Orifices	6	6
Included Angle (deg)	145	144
Nozzle orifice diameter, d_o (μm)	250	164
Discharge Coefficient, C_d	0.69	0.67
Hydraulic Flow (cc per 30 sec @ 100 bar)	1000	440
Aspect ratio (length/ d_o)	3.2	5.6
K factor ($(d_{o,inlet} - d_{o,exit}) / 10 \mu m$)	-0.2	-0.4

Conventional diesel combustion is mixing-controlled, and the mixing energy predominantly comes from the kinetic energy of the fuel injection event. Therefore, the ROI profiles must scale between the Cat and GM engines for combustion scaling to be successful. In other words, a proportional amount of fuel should be injected at the same relative crank angle interval of the engine cycle, with scaled velocity and spray penetration. For all scaling models the injection duration on a crank angle basis is the same for both engines. A Bosch ROI meter [18] was used to characterize and evaluate the ROIs before any experimental engine data was taken. Figure 2 shows a sample of the experimentally-measured ROI profiles for the Cat and GM injectors for simple scaling operating conditions.

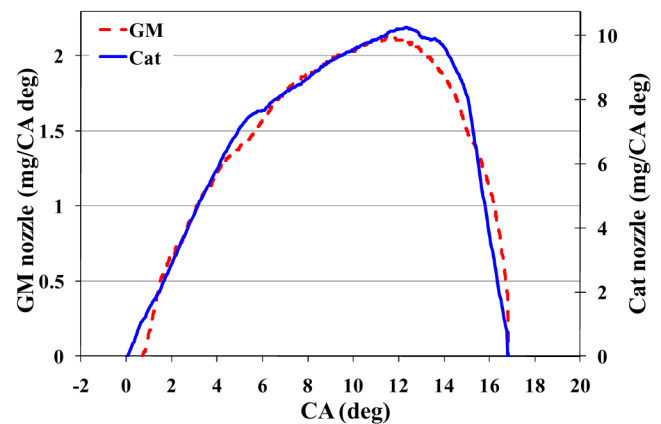


Figure 2. Simple scaling rates of injection for Cat and GM injector nozzles

Table 4 compares key injection parameters for the rate shapes shown in Figure 2. Consistent among all of the ROI comparisons for each of the scaling models is that the GM engine's 6-hole, 0.164 mm orifice diameter nozzle had a slightly shorter than ideal injection duration on a crank angle basis, perhaps because the orifice diameter and hence total orifice area was slightly oversized. Another interesting trend

is the comparatively faster ramp up at the start of injection for the 6-hole, 0.164 mm orifice diameter nozzle; this would suggest better mixing and faster combustion for the GM engine. Overall, the agreement was judged satisfactory for engine testing.

Table 4. Simple scaling injection metrics corresponding to rates of injection shown in Figure 2

Parameter	Unit	Simple Scaling Factor	Cat	GM	Actual Scaling Factor
Injection Pressure	bar	L^0	755	755	L^0
Fuel Mass	mg	L^3	117.1	24.6	L^3
Injection Duration	μ s	L^1	2320	1328	$L^{1.1}$
Engine Speed	rpm	L^{-1}	1208	2018	L^{-1}
Injection Duration	CA	L^0	16.8	16.1	$L^{0.1}$
Peak Flow Rate	mg/CA	L^3	10.3	2.1	$L^{3.1}$

COMPUTATIONAL MODEL

The current experimental set-up was modeled using the KIVA code [19] with sub-models developed at the Engine Research Center. A 1/6th 3-D mesh was used for each engine assuming axi-symmetric conditions. The number of cells at bottom dead center (BDC) is approximately 18,000 for the GM and 25,000 for the Cat engine. These meshes have been validated in a previous study [6]. Spray break-up was modeled with the Kelvin- Helmholtz (KH) model combined with the Rayleigh-Taylor (RT) model [20]. The gas jet model [21] was employed to suppress mesh dependency of the solutions. The physical properties of the diesel fuel were modeled as those of tetradecane ($C_{14}H_{30}$). The turbulent flow field was solved using the RNG $k-\varepsilon$ model [22]. Finally, chemical reactions were calculated using the CHEMKIN library, which was coupled with a reduced n-heptane mechanism containing 41 species and 130 reactions [23].

RESULTS

Comparisons of fired cylinder pressure data from the GM and Cat engines were one of the main tools used in evaluating the efficacy of the scaling laws. The extended scaling model was explored first, based on the results of previous research [6, 7, 8]. Ideally, the pressure traces would fall on top of each other for perfectly scaled combustion. Before presenting the fired cylinder pressure data, it is useful to analyze the motored pressure data through the lens of scaling, without the complications of combustion.

MOTORED CYLINDER PRESSURE

Figure 3 shows a motored cylinder pressure comparison of the GM and Cat engines operating at low speed extended

scaling conditions (Table 6). Additionally, a motored pressure trace of the Cat engine in stock configuration (16.1 compression ratio, -143 degrees aTDC IVC timing) is shown to highlight compression ratio effects. As in the fired cases, boost pressure was adjusted in both engines to obtain the same peak cylinder pressure (75 bar). The traces most noticeably deviate on the compression side from -70 to -15 crank degrees, which is a result of the gradient differences in the volume derivative. The scaled configuration of the Cat engine adequately matches the GM engine, thereby validating the Cat engine's modifications.

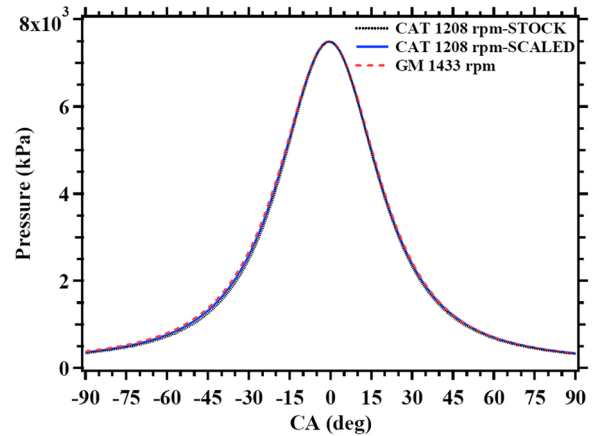


Figure 3. Motored cylinder pressure comparison at low speed, extended scaling operating conditions

Pressure versus volume curves are plotted in Figure 4 for low speed extended scaling conditions. The GM and scaled Cat compression and expansion curves matched reasonably well as predicted by Figure 3. The inset plot clearly shows the impact of the delayed IVC timing on the Cat engine: greater boost requirements (compared to the stock engine configuration) to reach the same peak pressure, and the convex curvature in the compression stroke near BDC.

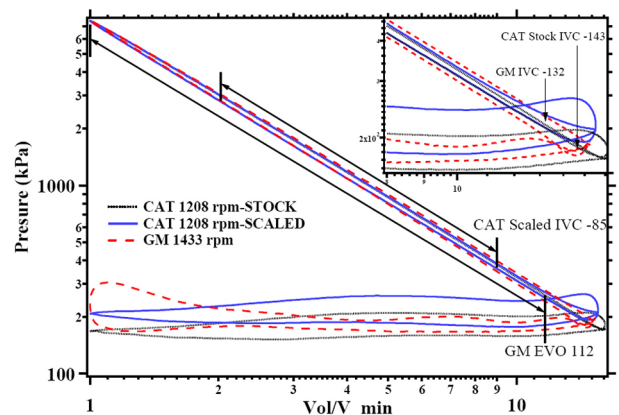


Figure 4. Comparison of pressure vs. volume traces at low speed, extended scaling operating conditions

The closed-valve period of a motored engine cycle can be modeled as an isentropic process, viz., $pV^n = \text{constant}$. An isentropic pressure versus volume curve would then result in a single, constant slope throughout the compression and expansion strokes. Assuming that heat transfer accounts for a majority of the irreversibilities in an actual engine cycle, then the change in slope between the compression and expansion strokes of the traces in [Figure 4](#) gives a qualitative measure of the heat transfer losses in the GM and Cat engines.

To test this hypothesis, the polytropic coefficients of the compression and expansion curves were compared. One method widely used to calculate the polytropic coefficient of the compression stroke is to consider the entire compression stroke, from BDC to TDC. The authors decided to use a narrowed region to improve the linear fit regression analysis of the particular engine configurations used in the study, specifically because of nonlinearities introduced at BDC from the delayed IVC timing of the Cat engine, and the noticeable curvature in the compression stroke for the Cat engine between the volume ratios from 1 to 2. Again in the interest of consistency, the same method was used for calculating the GM engine coefficients. The arrowed regions in [Figure 4](#) show the selected regions of the compression and expansion curves; their slopes are compared in [Table 5](#). At 1208 rpm the Cat engine's compression and expansion polytropic coefficients were nearly equal, whereas the GM engine at 1433 rpm (extended scaling) and 2018 rpm (simple scaling) exhibited a larger change in slope. Given that the GM engine has a larger difference in the polytropic coefficients for the compression and expansion curves, it suggests that the Cat engine has fewer losses than the GM engine.

Furthermore, based on a scaling analysis of [Equation \(2\)](#) [24], it can be theorized that normalized heat transfer is independent of engine size ($Q_w/m_f \propto L^3/L^3=1$) if engine speed scales by L^{-2} . Thus, the heat transfer losses from the Cat engine operating at a slow speed should scale (by L^3) the heat transfer losses from the GM engine operating at a fast speed. The data in [Table 5](#) supports this argument, in which the comparison between the most disparate speeds tested in this study (Cat 785 rpm/GM 2018 rpm = $L^{-1.83}$) shows the engines' losses to be on the same order of magnitude, although the GM engine's delta k was still larger by a factor of two.

Table 5. Comparisons of compression and expansion polytropic coefficients for Cat and GM engines

		k_{exp}	k_{comp}	Δk
Extended Scaling	Cat 1208 rpm	1.378	1.376	0.002
	GM 1433 rpm	1.402	1.362	0.040
Simple Scaling	Cat 1208 rpm	1.378	1.376	0.002
	GM 2018 rpm	1.393	1.356	0.037
Compensate for Heat Transfer	Cat 785 rpm	1.383	1.368	0.015
	GM 2018 rpm	1.393	1.356	0.037

EXTENDED SCALING - EXPERIMENT

[Table 6](#) lists the operating conditions for low speed extended scaling. Injection timing was swept from -10 to $+5$ crank angle degrees aTDC while holding the fuel rate constant, thus the engine load was not constant but ranged from 9.1-10.1 bar for the large engine and 8.6-10.2 bar for the small engine. The intake surge tank pressure was adjusted so that peak cylinder pressure equaled 75 bar when operating at an injection timing of $+5$ degrees aTDC. This boost pressure was then used for all cases in the injection timing sweep. The late SOI was chosen to determine boost requirements because peak cylinder pressure near TDC was not influenced by combustion (see [Figure 5](#)). The large engine required more boost because of the late IVC camshaft-with the stock camshaft boost pressure equaled 172 kPa. Injector shim height was adjusted to scale nozzle tip protrusion by the bore ratio. Consistent with the literature [5, 6, 7], the intake temperature of the GM engine was increased by ten degrees Celsius compared to the Cat engine to help compensate for heat transfer.

Table 6. Extended scaling engine operating conditions

	Cat	GM
Speed (rev/min)	1208	1433
Fuel rail pressure (bar)	755	380
Fuel mass (mg)	117	25
SOI (deg aTDC)	-10, -5, 0, +5	-10, -5, 0, +5
Nozzle tip protrusion (mm)	2.3	1.4
Intake Pressure (kPa)	205	189
Exhaust Pressure (kPa)	211	202
Intake Temperature (°C)	40	50
EGR (%)	0	0
IMEP_g (bar)	9.1—10.1	8.6—10.2

[Figure 5](#) compares fired cylinder pressure traces from the GM and Cat engines operating at extended scaling conditions. The general characteristics of the pressure traces agree reasonably well between the engines, specifically the start of combustion (SOC) timing, qualitative shape (*i.e.*, peaks and valleys corresponding to premixed and diffusion combustion events), and expansion strokes. The crank angle location of the GM