

An Experimental Investigation into Diesel Engine Size-Scaling Parameters

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ABSTRACT

With recent increases in global fuel prices there has become a growing interest in expanding the use of diesel engines in the transportation industry. However, new engine development is costly and time intensive, requiring many hours of expensive engine tests. The ability to accurately predict an engine's performance based on existing models would reduce the expense involved in creating a new engine of different size. In the present study experimental results from two single-cylinder direct injection diesel engines were used to examine previously developed engine scaling models. The first scaling model was based on an equal spray penetration correlation. The second model considered both equal spray penetration and flame lift-off length. The engines used were a heavy-duty Caterpillar engine with a 2.44L displacement and a light-duty GM engine with a 0.48L displacement. Several modifications were performed to the Caterpillar engine before testing in order to be consistent with the scaling models. These modifications included: a common geometry piston, a scaled injector nozzle, a scaled rate of injection profile by utilizing a flexible Hydraulic Electronic Unit Injector (HEUI), and an increased swirl ratio. This paper compares these engines operating across a conventional diesel combustion regime to examine the validity of the scaling arguments and to test the effects of engine speed and swirl ratio.

INTRODUCTION

Known for fuel efficiency, dependability, and ability to operate continually in high load conditions, the diesel

engine has become the mainstay of the heavy-duty transportation industry over the past century. However, recent increases in global fuel prices have placed additional emphasis on fuel economy. With their higher efficiency, a great interest in expanding use of diesel engines has resulted.

Previous researchers have studied methods for reducing both NO_x and PM to meet future government standards through the use of advanced technologies. Many of the strategies incorporate methods to increase in-cylinder turbulence, therefore raising the rate of PM oxidation. This can be accomplished through the use of direct fuel injection with a multi-hole nozzle utilizing high injection pressures [1,2]. These methods decrease burn durations and increase efficiency while reducing PM to allow for an increased amount of EGR to reduce NO_x formation. Increased in-cylinder turbulence or mixing can also be generated with the use of high levels of swirl [2] or multiple injection events [3,4,5,6]. Furthermore, an increased amount of in-cylinder air through the use of high levels of boost has been shown to reduce PM emissions and Brake Specific Fuel Consumption (BSFC) [7]. Variable IVC timing combined with flexible boost pressures has been shown to aid in the reduction of both PM and NO_x by reducing the compression ratio and thus decreasing the peak cylinder temperatures [8]. Despite the advancement of in-cylinder emission reduction it is likely that the use of exhaust after-treatment will be required in meeting future emissions standards on diesel engines.

The implementation of these advanced technologies has allowed the diesel engine to become more refined, not only with respect to emissions, but in terms of noise,

power and economy as well. Additionally, it has given engine developers many different design variables. While considered positive in terms of refinement, additional variables can potentially complicate engine development as more design and calibration work is required. Engine development and calibration are both costly and time consuming. Additionally, this work needs to be repeated when producing two engines that differ in displacement. Therefore, it would be greatly beneficial to use the strategy learned on one platform and apply it to produce engines of different sizes and applications.

The idea of diesel engine scaling was mentioned by Bergin et al. [9] as a means for testing a model of their “spin spray” combustion concept that explored two injections staggered in time. Their method of scaling the engine speed and injector parameters for a constant equivalence ratio was based on ensuring similarities in the in-cylinder flow and temperature distributions for diesel engines of different size. Bergin et al. considered a range of engine displacements that varied by a factor greater than 30. This wide range incorporated applications from high speed automotive to commercial power generation diesel engines. Stager et al. [10] extended the work of Bergin and proposed an additional or extended scaling model that considered both fuel spray and combustion characteristics. They applied these rules to developed models of two perfectly scaled engines with a length-scaling factor of two from the small to large engine. They concluded that their extended scaling model better predicted the combustion characteristics due to the addition of a flame lift-off length scaling parameter. Additionally, three regions were identified with varying levels of influence and turbulence and chemical timescales. Continuing the work of Stager, Shi et al. [11] applied the extended scaling model to two production engine platforms for a range of operating conditions. They found that in-cylinder pressure trace and heat release rate results could be well predicted based on the scaling laws. Emission results were well captured in combustion regimes controlled by turbulent time scales.

The present research follows the computer modeling work of Shi and Stager by experimentally testing their simple and extended engine size-scaling models. The focus was to evaluate the scaling models when applied to conventional compression ignition engines operating on diesel fuel. This work was performed on two single-cylinder diesel engines. The first is a heavy-duty 2.44L Caterpillar Single Cylinder Oil Test Engine (SCOTE). The second engine is a 0.48L GM HSDI diesel.

EXPERIMENTAL SETUP

CATERPILLAR ENGINE LAB - The larger engine was a Caterpillar 3401E Single Cylinder Oil Test Engine (SCOTE). This engine is a single cylinder version of the Caterpillar 500HP 3406E Heavy-Duty On-Highway truck engine. The engine’s base configuration and the final (Scaled) configuration that was used for the present research are both summarized in Table 1 below. A

schematic of the instrumented laboratory layout with the recently updated exhaust and EGR systems is given in Figure 1.

The gaseous emissions were measured on a concentration basis. The gaseous measurements include exhaust CO₂, CO, HC, NO_x and intake CO₂. The samples are acquired with a sample probe in the center of the exhaust and intake surge tanks. The particulate emissions (PM) were measured using an AVL 415s Smoke Meter. The smoke meter used its own heated sampling line placed in the center of the low pressure side of the exhaust stream. The meter was set to take five samples and average the resulting PM concentration. Sample volume was set to be a constant 1000ml throughout testing.

The fuel injector used in the large engine (CAT engine) for this research was a Caterpillar 315B Hydraulic Electronic Fuel injector (HEUI). The injector uses high-pressure engine oil to pressurize the fuel. This is done by means of a hydraulic amplifier which has a 6:1 amplification ratio. In the present research an oil pressure up to 26 MPa was utilized for a maximum injection pressure of 156 MPa.

Table 1: Caterpillar SCOTE specifications

Caterpillar 3401 SCOTE	
Displacement	2.44 [L]
Compression Ratio	Scaled: 15.5:1 SCOTE: 16.1:1
Squish Height	Scaled: 2.22 [mm] SCOTE: 1.57 [mm]
Bowl Volume	Scaled: 110.8 [ml] SCOTE: 115 [ml]
Connecting Rod Length	261.6 [mm]
Bore x Stroke	137.2 [mm] x 165.1 [mm]
Piston	Scaled Geometry Standard SCOTE - Mexican Hat Bowl
Valvetrain (4 valve)	EVC = -355 deg [ATDC] IVC = -143 (stock), 85 deg [ATDC] EVO =130 deg [ATDC] IVO =335 deg [ATDC]

An advantage of a HEUI-type injector is the decoupling of the fuel pressurization event from the piston location, which is a limitation of mechanical unit injectors. The HEUI is capable of providing an injection event anywhere in the engine cycle. Furthermore, the HEUI 315B allows for a flexible rate-of-injection (ROI). This flexibility in ROI rise makes the HEUI an optimum choice for engine size-scaling research, because it facilitates a match of the fuel delivery profile between the engines.

The specifications of the custom scaled HEUI 315B injector tips are given in Table 2. The tip geometries were chosen to scale with two different injector nozzles that were available on the small engine used in the present research.

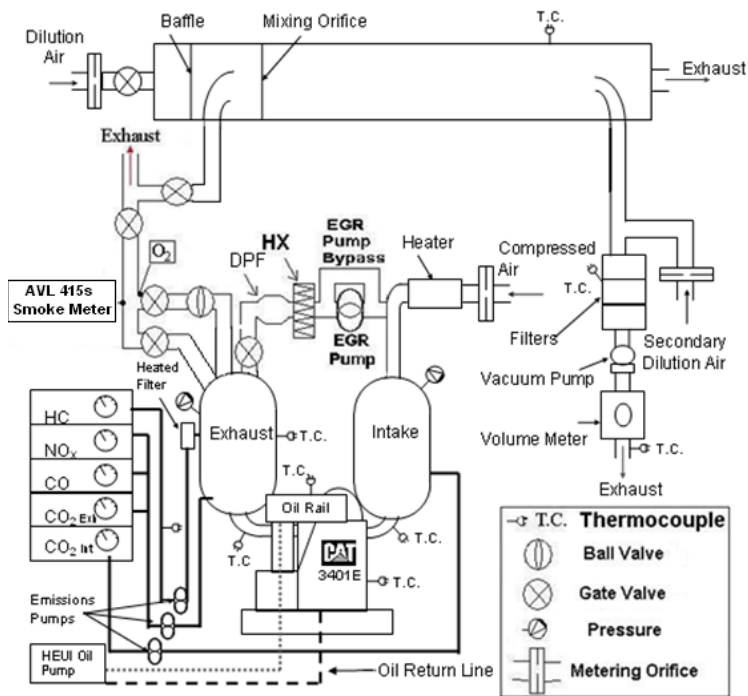


Figure 1: Caterpillar engine laboratory setup

Table 2: HEUI 315B Injector Specifications

Tip Configuration	Scaled Tips
Injection Pressure (max)	156 MPa
Number of Nozzle Holes	8
Nozzle Hole Diameter	229 μm
Spray Angle (included)	130°, 154°
Amplification Ratio (Injection Pressure to Oil Supply Pressure)	6:1

GM ENGINE LAB - The smaller engine used for the present research was a single cylinder direct injection diesel engine typical of a passenger car application. This engine used a Ricardo Hydra engine block and an engine head from a GM/FIAT 1.9L 4-cylinder HSDI engine. Table 3 summarizes the engine's specifications for its final scaled configuration. A schematic of the instrumented laboratory setup is given in Figure 2.

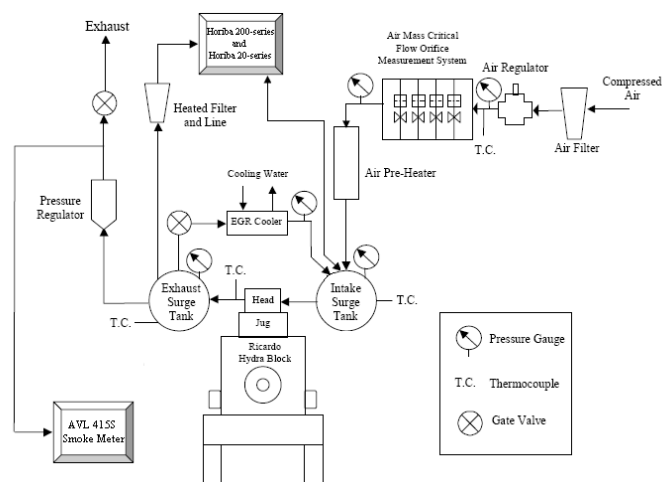


Figure 2: GM engine laboratory setup

Table 3: GM HSDI engine specifications

GM DOHC HSDI Diesel Engine	
Displacement	0.477 [L]
Compression Ratio	14.3:1
Squish Height	1.33 [mm]
Bowl Volume	23.85 [ml]
Connecting Rod Length	145.54 [mm]
Bore x Stroke	82.0 [mm] x 90.4 [mm]
Piston	Open-Crater-bowl (Scaled Geometry)
Intake Ports	1 Helical, 1 Tangential
Valvetrain (4 valve)	EVC = 388 deg [ATDC] IVC = -132 deg [ATDC] EVO = 112 deg [ATDC] IVO = 344 deg [ATDC]

The GM engine's injection system was a Bosch high pressure common rail injection system. This system allows flexible injection pressure and timing. The Injector specifications are shown in Table 4. Note that the 8 hole nozzles with both 130 and 154 degree spray angles correctly scale with the custom nozzles for the HEUI 315B from Table 2.

Table 4: Bosch Injector Specifications

Injection Pressure	Scaled Tips
# of injections per cycle	1 ~ 8 injections
Number of Nozzle Holes	8
Nozzle Hole Diameter	133 μm
Spray Angle (included)	120°, 130°, 140°, 154°
Flow Number (30sec steady 10MPa)	400cc

ENGINE SIZE-SCALING MODELS

SPRAY PENETRATION SCALING - An experimental investigation of fuel sprays was performed by Hiroyasu et al. [12] to develop correlations that have proven to be very useful for computational combustion analysis. Among these correlations are relationships for the spray tip penetration as a function of injection parameters and time. They determined two correlations for spray penetration. The first considers the jet tip penetration prior to liquid breakup. The liquid breakup is characterized by a breakup time, t_{BK} , and the breakup length, L_{BK} . The second correlation considers the penetration of the atomized spray, or in the case of injections into high temperature ambient gases, vapor phase fuel. These relationships are given by

$$s = 0.39 \left(\frac{2\Delta P}{\rho_F} \right)^{\frac{1}{2}} t \quad 0 < t < t_{BK} \quad \text{Equation 1}$$



$$s = 2.95 \left(\frac{\Delta P}{\rho_A} \right)^{\frac{1}{4}} \sqrt{d_o t} \quad t \geq t_{BK} \quad \text{Equation 2}$$

$$t_{BK} = 28.65 \frac{\rho_F d_o}{\sqrt{\rho_A \Delta P}} \quad \text{Equation 3}$$

s = Spray penetration length
 ρ_F = Liquid fuel density
 ρ_A = Air density
 ΔP = Pressure across injector
 t = Time
 d_o = Injector nozzle diameter
 t_{BK} = Time until jet breakup.

FLAME LIFT-OFF LENGTH SCALING - Due to high injection velocities the flame of a diesel injection diffusion plume anchors at a certain distance downstream of the injector nozzle [13]. This position is known as the flame lift-off length.

Siebers and Higgins [14] investigated the effects of ambient conditions, injection pressure and nozzle diameter on flame lift-off length in a constant volume combustion vessel. They found that increases in ambient temperature and density have a strong inverse effect on the lift-off length. Also, lift-off length decreases with reduced nozzle diameter and injection pressure. The amount of air entrained prior to the flame lift-off location was not directly proportional to the lift-off length. They found that even though a smaller nozzle diameter decreased lift-off length there was an increase in the percentage of air entrainment.

Siebers et al. [15] studied the effect of ambient oxygen concentration on flame lift-off length. Oxygen concentrations between 15 and 21 percent were considered to represent an EGR range of approximately 0 to 40 percent under high load conditions. They found the lift-off length to be inversely proportional to changes in the ambient oxygen concentration. However, the amount of oxygen entrainment was unchanged due to the lower oxygen concentration in the ambient air since the lift-off length increases correspondingly. The combustion and PM formation regions are pushed downstream with reduced oxygen concentration as the combustion rates are slowed, increasing the time required for the fuel preparation process to occur.

Pickett et al. [16] summarized the work of Siebers et al. [14,15] to propose a power law relationship for the flame lift-off length, H , based on several important parameters as

$$H = C T_A^{-3.74} \rho_A^{-0.85} d_o^{0.34} U^1 Z_{ST}^{-1} \quad \text{Equation 4}$$

H = Flame lift-off length
 C = Constant of Proportionality
 T_A = Ambient temperature
 ρ_A = Ambient gas density
 U = Injection velocity
 Z_{ST} = Stoichiometric mixture fraction.

They then continued to investigate the relationship of lift-off length to ignition delay time for several different diesel fuels, and found a strong correlation between the lift-off length and ignition delay for the conditions tested.

ENGINE TORQUE SCALING - In previous work with diesel engine scaling [10,11] a conservation of mass relationship was incorporated into the scaling model as

$$m = \rho_f A_{noz} U \Delta t \quad \text{Equation 5}$$

m = Mass of fuel injected
 ρ_f = Fuel Density
 A_{noz} = Nozzle area
 U = Injection velocity
 Δt = Injection duration (time)

The use of Equation 5 scales the mass of fuel injected in the cylinder as a function of engine size. Shi and Reitz [11] note that if the engines exhibit the same thermal and combustion efficiency this will result in a scaled engine torque or constant mean effective pressure (MEP).

SIMPLE SCALING MODEL - The simple scaling model is based on both conservation of mass and equal spray penetration correlations as defined by Equations 2 and 5. For this scaling model the engine speed is free to be chosen as the two correlations do not form a closed equation set. As with the previous work by Stager [10] the engine speed is chosen to be inversely proportional to the length scaling factor (L) and the injection duration (Δt) is set to scale as the inverse of the engine speed. This results in an injection duration that is equal in engine crank angle degrees between the engines. The mass of fuel injected (m) is scaled by the TDC volume (L^3) to achieve a constant equivalence ratio. The spray penetration is set to scale proportional to the length scaling factor to place the fuel plume in the same location of the combustion chamber. Then Equation 2 and 5 are solved to determine the nozzle area and injection velocity. Table 5 shows the resulting values of the scaling parameters.

Table 5: Simple scaling model scaling factors [10]

Parameter	Scaling Factor Length	Scaling Factor Volume
m	L^3	V
s	L	$V^{1/3}$
d	L	$V^{1/3}$
A	L^2	$V^{2/3}$
Δt	L	$V^{1/3}$
U	L^0	V^0
N	L^{-1}	$V^{-1/3}$

EXTENDED SCALING MODEL - The extended scaling model also uses the conservation of mass and spray penetration correlations as the simple scaling model. The extended model expands on this to scale the combustion characteristics by the addition of a flame lift-off length.

off correlation with Equation 4. Below these correlations are simplified by eliminating the in-cylinder air and injected fuel densities [10]. During testing the attempt was made to keep these properties consistent between the engines.

$$m \approx d_o^2 U \Delta t \quad \rightarrow \quad L^3 \approx (L)^2 L^{2/3} L^{1/3}$$

$$s \approx d_o^{1/2} \Delta t^{1/2} U^{1/2} \quad \rightarrow \quad L \approx (L)^{1/2} (L^{1/3})^{1/2} (L^{2/3})^{1/2}$$

$$H \approx d_o^{0.34} U^1 \quad \rightarrow \quad L \approx (L)^{0.34} L^{2/3}$$

The mass of fuel injected again scales linearly with TDC volume for proper torque scaling also; the spray penetration and flame lift-off length scale linearly with the length scaling factor. With these restrictions the three correlations above form a closed equation set with a unique solution for the nozzle area, injection duration and injection velocity. Table 6 shows the resulting scaling parameters from the solution to equation set from above.

Table 6: Extended scaling model scaling factors [10]

Parameter	Scaling Factor Length	Scaling Factor Volume
m	L^3	V
s	L	$V^{1/3}$
H	L	$V^{1/3}$
d	L	$V^{1/3}$
A	L^2	$V^{2/3}$
Δt	$L^{1/3}$	$V^{1/9}$
U	$L^{2/3}$	$V^{2/9}$
N	$L^{-1/3}$	$V^{-1/9}$

From comparison of Tables 5 and 6 it is seen that to properly scale all of the correlations in the extended model requires that the engines operate at a more similar engine speed. Also, to conserve the scaling of the fuel mass injected the injection velocity is increased in the larger engine. Note that the nozzle diameter scales linearly with the length scaling factor for both the simple and extended scaling models.

SWIRL RATIO SCALING – Hiroyasu [12] developed two dimensionless correlations to characterize the influence the rotating air in the cylinder (i.e., swirl) has on the fuel spray. The first correlation characterizes the effect on spray penetration, where the parameter C_s is proportional to the decrease in spray penetration; this is shown in Equation 6. The second parameter C_θ is proportional to the azimuthal deflection of the spray as shown in Equation 7.

$$C_s = \left(1 + \frac{\pi R_s N S}{30 U_0} \right)^{-1} \quad \text{Equation 6}$$

$$C_\theta = \left(1 + \frac{\pi R_s N S}{30 U_0} \right)^2 = \frac{1}{C_s^2} \quad \text{Equation 7}$$

R_s = Swirl Ratio
 N = Engine Speed
 S = Spray Penetration
 U_0 = Injection Velocity

The value of C_s and C_θ for the two engines should be equal such that the spray deflection scales proportional to the length scaling factor. Solving for the swirl ratio scaling factor with the simple scaling model gives

$$C_s \approx \left(\frac{R_s N S}{U_0} \right)^{-1} \Rightarrow L^0 \approx \left(\frac{R_s L^{-1/3} L^1}{L^0} \right) \Rightarrow R_s \approx L^0$$

And the extended scaling model gives

$$C_s \approx \left(\frac{R_s N S}{U_0} \right)^{-1} \Rightarrow L^0 \approx \left(\frac{R_s L^{-1/3} L^1}{L^{2/3}} \right) \Rightarrow R_s \approx L^0$$

From above it is shown that the swirl ratio should be the same numeric value for each engine when using either the simple or extended scaling model.

ENGINE SIZE-SCALING FACTOR - For the studies of Stager [10], the engines were perfectly scaled. That is every length dimension between the engines was proportional to a length scaling factor (L) of two. Also, the engine's displacement and other volumes were proportional to a volume scaling factor (V) equal to the cube of the length scaling factor ($V = L^3$).

When applying the scaling models to real production engines finding two engines that share the same relative dimensions would be challenging. Large engines typically have a smaller bore-to-stroke ratio because they run at lower engine speeds. Low engine speeds reduce the frictional losses and high piston speeds caused by a long stroke. Shi et al. [11] used a computational model to compare two different scaling factors for the same engines used in the current research. The first, termed Displacement Volume Scaling was to use the ratio of the engine's swept volume or displacement $V=2.439/0.477$. With this method the squish region does not have the same relative dimensions due to the CAT engine's smaller relative bore diameter. Importance of the TDC flow fields on the combustion process led to a second method of size-scaling based on the engine's bores $L = 137.2/82.0 \rightarrow L=1.67$. This method termed TDC Volume Scaling resulted in the same relative dimensions at TDC. TDC Volume Scaling was the method of scaling pursued for the current research.

The base SCOTE piston geometry is a shallow bowl Mexican hat design. This design was not used for the scaled engines because the wide bowl caused some potential interference problems with the GM piston's ring groove. New pistons with profiles based on the GM laboratory's experimental bowl design were produced for both engines. The pistons were designed to scaled bowl

dimensions and squish height. The profiles of the two scaled piston bowls as well as the original Caterpillar SCOTE piston geometry are shown in Figure 3.

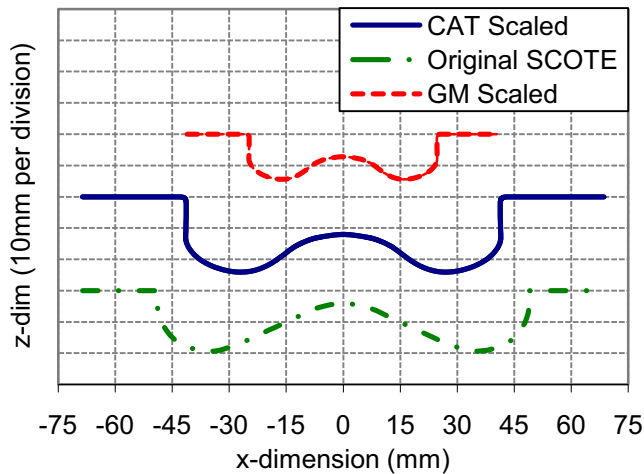


Figure 3: Piston geometries, Caterpillar Original SCOTE and Scaled, GM Scaled.

Table 7 shows the resulting engine dimensions with the scaled geometry pistons installed and the desired versus actual scaling factors. The values are as manufactured not designed. For this reason there is some small discrepancy between the desired and actual scaling of TDC dimensions.

Table 7: Measured CAT and GM engine dimensions with scaled pistons installed

	CAT Scaled	GM Scaled	Desired Scaling	Actual Scaling
Bore (mm)	137.2	82.0	L = 1.67	1.67
Stroke (mm)	165.1	90.4	L = 1.67	1.83
Connecting Rod (mm)	26.2	14.5	L = 1.67	1.80
Squish Height (mm)	2.2	1.3	L = 1.67	1.65
Squish Width (mm)	25.7	15.4	L = 1.67	1.67
Bowl Volume (cm^3)	110.8	23.85	L^3= 4.68	4.65
Bowl Diameter (mm)	85.9	51.3	L = 1.67	1.67
Displacement (L)	2.44	0.477	L^3= 4.68	5.12
TDC Volume (cm^3)	167.7	35.46	L^3= 4.68	4.73
Crevise Volume (cm^3)	24.4	4.59	L^3= 4.68	5.32
* L = CAT _{bore} / GM _{bore} = 137.2 / 82.0 = 1.67				

METHODS

RATE OF INJECTION - The importance of the ROI shape lies in the scaling models. Not only should the peak value of the injection velocity scale but the injection

event should exhibit a scaled velocity profile. Then if the nozzles' effective areas scale with L^2 the injection mass and duration will be correct. However, the effective area of the nozzle is dependent on both, manufacturing tolerances and the discharge coefficient of the holes. Plus, the injection velocity cannot be directly measured with the ROI meter. Therefore, a higher priority was placed on scaling the fuel mass delivery profile for the present research. With the mass profile scaled then scaling parameters of injection duration and fuel mass per injection will be met. This will ensure similar torque scaling and simplify the comparison between the fired engine cases. Also, because the scaled HEUI 315B nozzles were made with an area target based on the scaling models the resulting injection velocity profile will be close to the desired scaled value. Minor differences may occur due to differences in discharge coefficient or manufacturing.

The Bosch injector exhibits a faster rising ROI than the HEUI's ramp ROI. For this reason the flexible rate shaping of the HEUI needed to be implemented. The rate shaping of the HEUI 315B is done by incorporating a small pilot command shortly before the main command. The pilot command is characterized by the command duration and dwell duration. The dwell duration is defined from the end of the pilot to the beginning of the main command.

The command for a square ROI was previously defined as a 550μs pilot duration and 800μs dwell (550-800 naming convention). The resulting square ROI injection profile is shown in Figure 4 below for the 229μm nozzle.

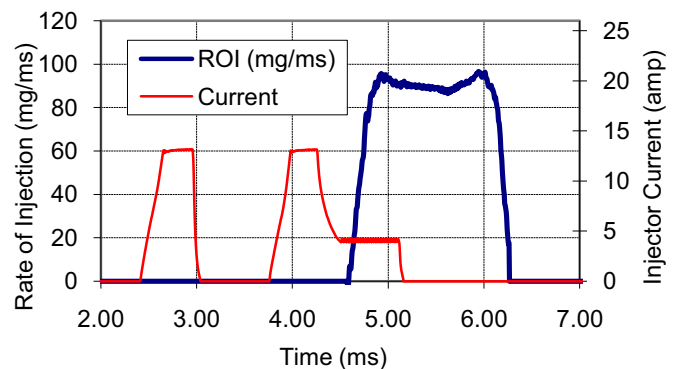


Figure 4: HEUI 315B Square ROI (550-800)

For ROI's slower than square Coldren et al. [17] describe an ERS injection (Electronic Rate Shapes) that is between the square and ramp ROI's. This method involves a shortened cc-pilot dwell and slightly longer cc-pilot duration. Initially this was attempted with the scaled injector. However, the ROI's initial rise was not able to be lowered enough while remaining stable before reverting back to a ramp like ROI. The method used to develop the scaled ROI was to increase the cc-pilot duration and cc-pilot dwell. With this method the fuel inside the injector is pressurized as with the square ROI, however the additional time in the dwell allows the pressure to subside. The result is ROI profiles that exhibit a "fast ramp" leading edge. Several different