

Compression Ignition 6-Stroke Cycle Investigations

D. Ryan Williams, Chad Koci, and Scott Fiveland
Caterpillar

ABSTRACT

Driven by the desire to implement low-cost, high-efficiency NO_x aftertreatment systems, such as Three Way Catalysts (TWC) or Lean NO_x Traps (LNT), a novel 6-Stroke engine cycle was explored to determine the feasibility of implementing such a cycle on a compression ignition engine while continuing to deliver fuel efficiency. Fundamental questions regarding the abilities and trade-offs of a 6-stroke engine cycle were investigated for near-stoichiometric and lean operation. Experiments were performed on a single-cylinder 15-liter (equivalent) research engine equipped with flexible valvetrain and fuel injection systems to allow direct comparison between 4-stroke and 6-stroke performance across multiple hardware configurations. 1-D engine simulations with predictive combustion models were used to support, iterate on, and explore the 6-stroke operation in conjunction with the experiments. Output from the experiments and simulations were then used to perform Availability and Energy balances for a thermodynamic comparison of the two cycles.

Compared to 4-stroke cycle operation, the 6-stroke cycle exhibited lower PM emissions at stoichiometric operation, while higher NO_x/PM emissions were observed under some lean conditions. The ratio of fuel burned in the first combustion event to the second event had a strong impact on performance, heat loss, and emissions. For some 6-stroke strategies engine breathing and airflow management were found to be challenging, and beneficial for other strategies. The thermodynamic analysis showed that under similar boundary conditions, the 6-stroke and 4-stroke engine cycles could attain very similar brake efficiencies, though the detailed availability balance breakdown could differ substantially.

CITATION: Williams, D., Koci, C., and Fiveland, S., "Compression Ignition 6-Stroke Cycle Investigations," *SAE Int. J. Engines* 7(2):2014, doi:10.4271/2014-01-1246.

INTRODUCTION

The three-way catalyst (TWC) has been used in the automotive industry for more than 30 years as a simple and cost-effective means of reducing NO_x, Hydrocarbon, and Carbon Monoxide emissions from spark ignited (SI) engines. Improvements to the technology have led to conversion efficiencies in excess of 99% as emissions regulations have become increasingly stringent. Despite the success in the SI world, the technology has not found its way onto the diesel engine due to the fact that the TWC requires very tightly controlled modulation of the air-fuel ratio around the global stoichiometric point in order to achieve maximum conversion efficiency. Diesel engines have historically stayed far to the lean side of stoichiometric operation because difficulties with mixing and oxygen utilization lead to poor fuel consumption and high particulate matter (PM) emissions.

However, over the past decade there have been significant pushes towards a clean and efficient globally stoichiometric, or rich, diesel combustion system. Light-load (<5 bar BMEP) stoichiometric and rich operation has been shown to be effective for TWC operation and Lean NO_x Trap (LNT) regeneration, [1, 2, 3, 4, 5, 6], where high EGR, switching, and multi-shot strategies all showed benefits in light-duty diesel

engines. Research efforts explained how PM formation mechanisms could be altered through reduced temperature regimes, and where the fundamental differences in emissions and efficiency were with the various stoichiometric strategies. Higher load (>15 bar BMEP) stoichiometric research has been touched on in the published literature in [7, 8], but not expanded in detail. Potential improvements in combustion system performance, like the efforts of [9] suggest, could enable a high-load stoichiometric system to be attractive. One need for improvement stems from an engine system level consideration, where aftertreatment system efficiencies are greatly sensitive to the exhaust temperature and emissions magnitude over the certification cycle or modes. Another need for improvement stems from the reduced efficiency of a stoichiometric diesel combustion system due to poor mixing and increased heat loss.

Thinking about both emissions, mainly PM, and efficiency improvements from a stoichiometric combustion system, a 6-stroke cycle was proposed as an interesting avenue for exploration. It was noted how a 6-stroke cycle had been previously investigated in [10, 11, 12, 13] with an emphasis on a re-oxidation of PM from the first combustion event by the second combustion event. Later a dual-fuel approach was

taken, [14]. The ability for a second combustion event to increase or alter late cycle PM oxidation was an attractive idea when considering stoichiometric diesel combustion. Additionally, the 6-stroke cycle should clearly be evaluated under lean conditions for complete understanding regarding soot oxidation and cycle efficiency.

The overarching ideas spurring the present work were to combine the pursuit of stoichiometric diesel combustion as an avenue for reduced low-emissions achievement by way of a TWC/LNT, and incorporating this with a 6-stroke cycle extended to high-load operation. As will be discussed and reviewed, both the stoichiometric 6-stroke cycle and high-load 6-stroke cycle have non-traditional challenges, strategies, and trade-offs.

Key Cycle Concepts

As there are many facets of the 6-stroke cycle that are somewhat unique to this combustion strategy, some of these key concepts will be introduced here in the hope that it will enhance the understanding of the discussion that is to follow.

6-Stroke Definition

The primary difference between the 4-Stroke and 6-Stroke cycles is the addition of a second compression and combustion stroke immediately following the primary combustion event as shown in [Figure 1](#). This approach is fundamentally different from the traditional 6-stroke approach where water is injected into the cylinder after recompression in order to extract additional expansion work from the hot combustion products. The purpose of the second combustion event is to utilize the excess oxygen in the cylinder by providing an extended period of time in a high temperature environment to promote oxidation of soot particles formed during the first combustion event.

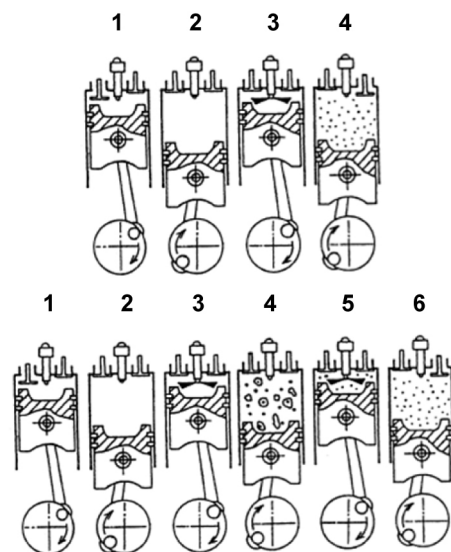


Figure 1. Illustration of 6-Stroke cycle vs. Traditional 4-Stroke cycle. Note the omission of the exhaust event on the fourth stroke in order to recompress the combustion gasses. Adapted from [Figure 1](#) in [12]

Degrees of Freedom

The first thing that should be appreciated when considering the 6-stroke cycle is the number of added degrees of freedom that are introduced by the inclusion of a second combustion event. First and foremost, the quantity of fuel that is injected during each of the combustion events can significantly impact the combustion performance. For the majority of the experiments that will be discussed here, the total fuel rate was held constant for a given operating condition. The percentage of the total fuel that was injected during the first versus the second combustion event, however, was often varied. This ratio of fuel delivery will hereafter be referred to as the 'fuel split' and will be specified simply by the percent of fuel injected during the primary combustion event (i.e. a 70% fuel split will indicate that 70% of the fuel was injected during the first combustion stroke and the remaining 30% was injected during the second combustion stroke).

It should be noted that these injection events are not mutually exclusive. Because much (or all) of the combustion products from the first combustion event are retained in the cylinder, the temperature, pressure and oxygen concentration at the beginning of the second combustion stroke can be impacted by the fuel split. In general, a larger fuel split will consume more oxygen and result in a higher bulk gas temperature and, subsequently, higher pressure on the second stroke. The converse is also generally true; burning less fuel during the first combustion stroke will give lower temperature and pressure for the second stroke. However, because the combustion products from the first combustion are mixed with remaining excess air during the expansion and compression strokes, the local oxygen concentration available to the fuel will not be equivalent to the conditions during the first injection event. Mixing and air entrainment can also be hampered by the elevated temperatures experienced at the start of the second injection event. Thus an attempt to conserve oxygen for the second combustion stroke can be squandered by the inability to utilize the excess oxygen on the second stroke.

The addition of a second combustion event also doubles the flexibility of other standard combustion parameters such as injection timing and number of injections. Even the injection pressure could be independently varied, though with greater difficulty. As mentioned previously, some studies have utilized a second injector and even a second fuel during the second combustion event. There is also the potential for the use of valve events to manipulate combustion. While the sensitivity and optimization of these parameters was not in the immediate scope of this work, to the extent that these other parameters were studied, they will be addressed in the discussion.

Power Density

The power density of the 6-stroke cycle is best considered in relation to a typical 4-stroke configuration. The 4-stroke cycle consists of one power stroke for every two crankshaft rotations. The 6-stroke cycle consists of two power strokes for every three crankshaft rotations meaning, at face value, that a 6-stroke engine should have a power density 4/3x the 4-stroke

power density. The important distinction should be made here that 6-stroke cycle does not have individual exhaust and induction events for each combustion event; instead, all of the air for both combustion events must be inducted and exhausted during a single gas exchange process. Thus, with similar mechanical constraints of the engine components (i.e. boost, cylinder pressure, port flow, metal temperatures, etc.) the 6-stroke engine should be limited to a similar trapped mass per cycle as that of the 4-stroke engine.

Additionally, because the 6-stroke cycle contains an additional revolution of the crankshaft per cycle (1080 deg vs. 720 deg) there are some interesting implications for power and flow rates which are calculated on a basis of time rather than per cycle. First, with only one induction event per cycle, the 6-stroke will have 33% fewer induction events over the same time period as a 4-stroke engine. Thus, with an equal inducted mass per cycle, the 6-stroke engine will require 33% less airflow. At a system level this can have beneficial effects on the pumping work and overall efficiency of the engine system. Second, in order to maintain equal power, the 6-stroke engine must either spin at a rate of 50% faster than the 4-stroke engine or burn 50% more fuel per cycle than the 4-stroke. For this application the latter option was selected. With 50% more fuel per cycle a pure 50% fuel split would represent 75% of the equivalent 4-stroke injection quantity per injection, and a 67% fuel split would result in a first combustion injection equal in quantity and duration to the 4-stroke cycle. At a full load condition, where the injection duration is often limited by the design of the combustion chamber, this could represent the practical limit to the 6-stroke fuel split.

Thus, inasmuch as a combustion system can be designed to deliver the flow rates as described above a 6-stroke engine should be able to deliver a power density that is on the same order of, but not likely substantially better than the 4-stroke engine. This fact will be further corroborated throughout the bulk of in this paper.

Gas Exchange

Maximum engine power output is practically constrained by thermal and mechanical limits of the engine components. As it has been previously explained, the temperature and pressure at the start of the second combustion stroke will be higher because of the energy added by the first combustion event. Because of this, the 6-stroke engine is likely to reach the engine design limits at a much lower load factor than an equivalent 4-stroke engine, unless measures are taken to reduce the temperature and pressure in the cylinder before the start of the second compression stroke. This can be done through a partial exhaust event known as Inter-Cycle Blowdown (ICB) [15]. The increased expansion allowed by the reduction of mass in the cylinder results in lower temperature and pressure; furthermore, the temperature and pressure of the cylinder can be controlled to some extent by the amount of mass that is expelled during the ICB. The timing of the ICB can also be manipulated to take greater advantage of expansion work.

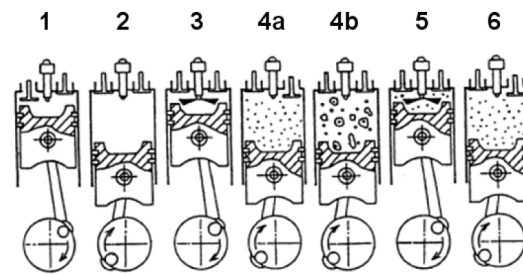


Figure 2. Illustration of 6-Stroke cycle with inter-cycle blowdown. A small portion of the gasses are exhausted in 4a and the remaining gas is compressed in 4b. Adapted from Figure 1 in [12]

While the ICB is a key enabler in achieving higher engine loads with 6-stroke combustion, it also presents some interesting implications to the gas exchange of the cycle. These implications are of particular concern when considering stoichiometric operation, as has been proposed. In order for the TWC to operate properly, the composition of the exhaust gas must be held to a very near stoichiometric equivalence ratio. Because any gases expelled during the ICB will, of necessity, be lean of stoichiometric, the ICB gasses must either be mixed with fuel rich combustion products from the second combustion event or segregated from the rest of the exhaust and recirculated into the cylinder. The latter arrangement would require a split-port head design where independently actuated valves could be used to direct the exhaust products either through the recirculation loop or through the exhaust as pictured in Figure 3. The gas exchange implications of such a design will be discussed in further detail later on.

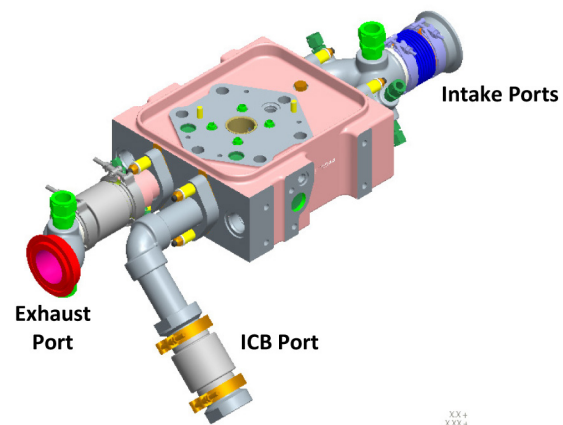


Figure 3. Split port head design for separating ICB flow from exhaust flow during stoichiometric operation.

All of these considerations aside, there is another, more fundamental, tradeoff inherent to the ICB that deserves some attention here. The tradeoff relates to the fuel split and the availability of oxygen particularly during the second combustion stroke. Because all of the air for the entire cycle is inducted prior to the first combustion event, the magnitude of the fuel split will determine how much of the fresh air is left over for the second combustion event. If a portion of the unburned air is expelled during the ICB, however, there will be less oxygen remaining in the cylinder for the second combustion.

stroke. This tradeoff becomes increasingly important at lower fuel splits as a greater proportion of the ICB gas will be made up of fresh air, though it can be reasonably assumed that less ICB will be needed at lower fuel splits to stay within temperature and pressure limits.

EXPERIMENTAL SETUP

Experiments were conducted on a single-cylinder engine that has been adapted from the Caterpillar C-15 (15 liter) engine platform. The test cell is equipped with a Horiba Mexa 7000 for measuring concentrations of gaseous emissions as well as an AVL 415 Smokemeter to give a measure of relative changes in soot production and/or oxidation. The engine was also fitted with an indicating system for crank-angle resolved measurements of cylinder pressure and inlet and exhaust manifold pressures at a resolution of 0.1 CAD. Cylinder pressure was measured using a Kistler 6125C piezoelectric pressure transducer and manifold pressures were measured using Kistler 4045A piezoresistive transducers with a water-cooled adapter used in the exhaust manifold.

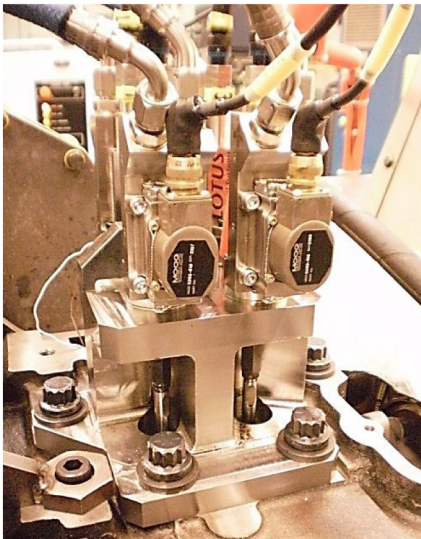


Figure 4. Cylinder head fitted with Lotus AVT system

Several modifications were made to the single cylinder engine to allow for the flexibility of switching seamlessly between 4-stroke and 6-stroke cycles, for comparison of both cycles under similar conditions. The most significant of these modifications involved the actuation of the intake and exhaust valves. Because the valves are normally operated mechanically by a camshaft that is ultimately tied to the rotation of the crankshaft, changing the valve actuation to operate over a 1080 degree cycle versus a 720 degree cycle would entail the removal and redesign of the camshaft and geartrain for 6-stroke operation with additional modifications required to make any changes in the ICB strategy. In light of these hardware challenges the Lotus Active Valve Train (AVT) fully-flexible valve actuation system was determined to be the best option for the testing. The AVT uses hydraulic pressure provided by a separate pump and power source to actuate the

engine valves which decouples the valve actuation from the engine rotation allowing for flexibility in valve timing and switchability from 4-stroke to 6-stroke cycles without any hardware changes.

Similar challenges were also faced in adapting the fuel delivery for the 6-stroke cycle. Once again, in order to decouple the injector operation from the crank and camshaft rotation, the standard Mechanical Electric Unit Injector (MEUI™) was replaced with a high-pressure common rail injector, which was also supplied by a separate pump and power source. The common rail injector was able to provide up to six separate fuel injections per cycle, split in any combination between the first and second combustion strokes.

The final modification required for 6-stroke operation was the adaptation of the Engine Control Module (ECM) to recognize a cycle with three crankshaft rotations rather than two. Modern electronic fuel injection systems rely on a timing signal usually provided by a sensor that 'counts' special teeth that are machined into the crankshaft to determine the rotational speed of the engine, as well as the position of the piston within the desired cycle. The process of computing the speed and position of the engine is often handled by special dedicated circuits within the ECM, because these calculations must be performed at speeds much faster than the ECM's internal processor can handle. While this provides for more robust engine control, it also means that the internal circuitry of the ECM is inherently fixed to a 720 degree cycle, and would need to be physically altered in order to accommodate 1080 degrees per cycle. Here again, a more flexible solution was developed to offer the ability to easily switch between 4-stroke and 6-stroke without changing the ECM. In order to do this the control logic was moved to a more flexible ECM with a larger Field Programmable Gate Array (FPGA), where the dedicated timing circuits could be represented by re-programmable transistor arrays without sacrificing computational speed. This allowed the 720 degree and 1080 degree logic to be placed side-by-side on the chip with a switch to select the appropriate logic for the desired engine cycle. Simplified logic for controlling the rest of the engine functions was also added to the FPGA to keep the number of required ECMs to a minimum.

CYCLE SIMULATION AND ANALYSIS SETUP

The following sub sections describe the simulations and calculations used throughout the 6-stroke cycle and the thermodynamic analysis.

1-D Cycle Simulation

Engine cycle simulations were conducted in Caterpillar Inc.'s proprietary 1-Dimensional (1-D) engine simulation code, DYNASTY™. This code is capable of modeling entire engines and their subsystems with great freedom and flexibility, [16, 17, 18]. The cycle simulations were matched to the single-cylinder experimental test engine, and utilized high and low speed boundary conditions specified from the test data. For the

present work, turbocharger simulations were not included in the 1-D simulations; however, simple turbocharger calculations were performed offline. A schematic of the simulation layout is presented in Figure 5.

In the 6-stroke 1-D engine simulations, compressible flow modeling is handled through discretized duct and orifice components, which include convection and radiation heat losses. Pressure boundary conditions for the inlet and exhaust surge tanks were prescribed using high-speed test data from sensors on the tanks. The model incorporates inlet and exhaust geometry that includes the ports, runners, and sections of the single-cylinder surge tanks. Valve lift and port flow coefficients are used to model engine poppet valve breathing and can easily be modified. When the EGR duct path is not modeled, EGR is simulated as a component of the inlet air, and is controlled with a standard PID feedback loop. Crank-slider calculations are used for piston motion, and friction is incorporated in the mechanical simulation with a user prescribed FMEP value. For the present work, the FMEP was described as a function of engine speed and peak cylinder pressure aligned with 4-stroke engine data. FMEP was cast such that the friction power was preserved between a 6-stroke and 4-stroke engine cycle for the same speed and peak cylinder pressure.

Combustion is handled within a cylinder component in DYNASTY™. A single-zone combustion model was chosen for the compression ignition 6-stroke investigations, with predictive premixed and mixing controlled combustion models. The combustion models predict the gross heat release rate and include ignition delay models and relationships for mixing due to the direct injection of fuel. The fuel injection is modeled through a separate injector component, as a common rail injector. In-cylinder heat transfer was modeled using Caterpillar's proprietary bulk gas heat transfer Nusselt and Reynolds number correlation. NOx predictions are made based on the cylinder thermodynamics and adiabatic flame temperature calculations.

The 1-D engine model was first setup and calibrated to single-cylinder 4-stroke test data. An example 4-stroke matched log-P log-V diagram in Figure 6 shows good results. Compression, combustion, expansion and the pumping loop are matched sufficiently to capture the main thermodynamics and cycle performance. The NOx model was calibrated over full and part load 4-stroke timing swings and had a sufficient $\pm 15\%$ error over a very wide (no EGR to $+25\%$ EGR at full load) range, Figure 7. Subsequent comparison to measured engine data found that this was sufficient for 6-stroke cycle simulation NOx predictions.

To change the 4-stroke 1-D model to simulate a 6-stroke cycle, a few modifications were implemented. The cycle was changed to operate on 1080° (3 revolutions, 6 strokes) instead of 720° , valve lift events were moved to the correct location within the 1080° cycle, and additional fuel injection capability was added for around 360° after the first TDC. All cycle-based calculations (i.e. IMEP, BMEP, etc.) in the simulation results were modified to account for the differences in the cycles in order to compute the correct quantities.

The cycle simulations provide the following results and cycle outputs:

- Brake power, brake torque, BSFC
- Airflow, fuel flow, EGR flow, AFR
- Dynamic intake/exhaust port mass flow
- MEP's, brake and indicated
- In-cylinder heat transfer and heat rejection
- Cylinder surface temperatures
- NOx emissions
- Cylinder pressure, volume, and mass traces
- Cylinder bulk gas temperature trace
- Cylinder apparent and gross heat release traces

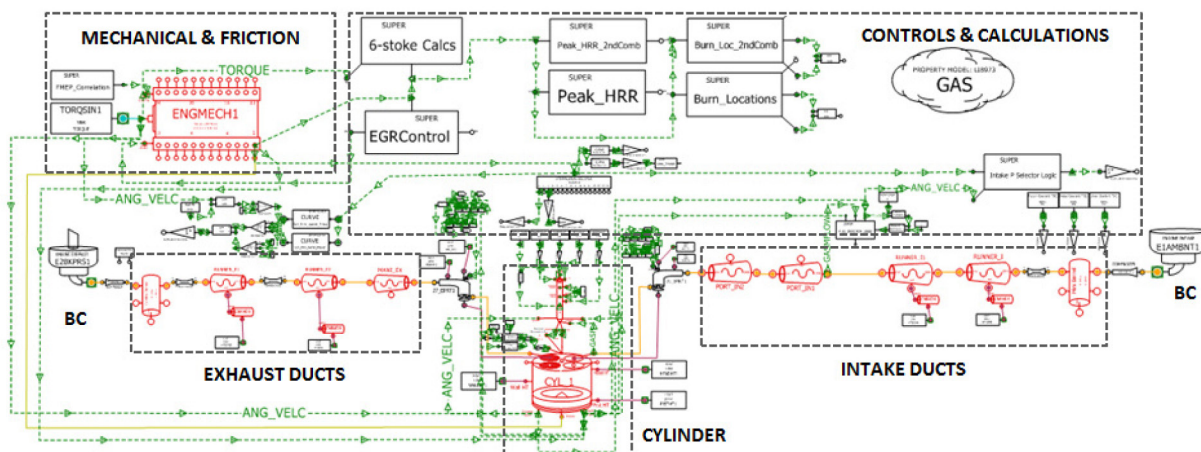


Figure 5. Schematic of the 6-Stroke single cylinder 1-D engine simulation in Caterpillar Inc.'s software, DYNASTY™.

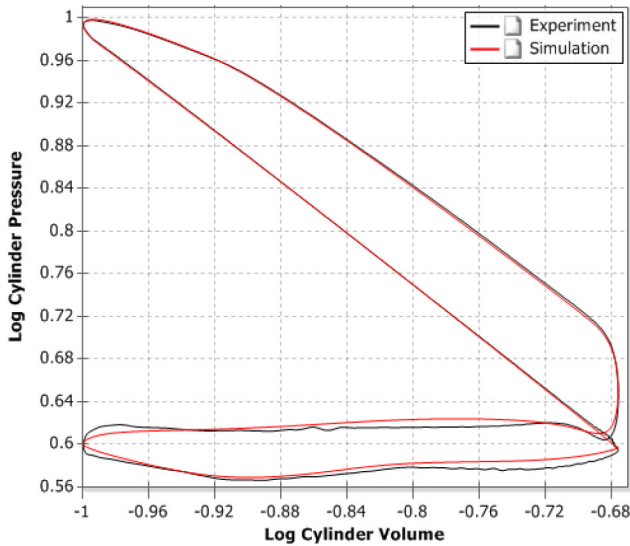


Figure 6. LogP-LogV diagram showing good matching for the 1-D engine simulation to 4-stroke C15-1 data.

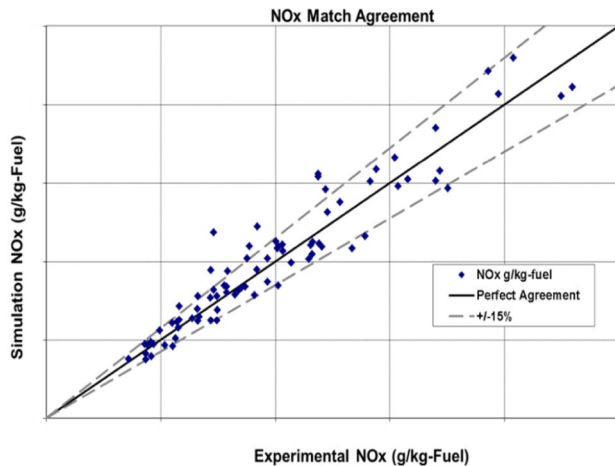


Figure 7. NOx predictions from 1-D simulations showing +/-15% agreement with experimental data.

These results allowed thorough characterization and understanding of the 6-stroke cycle. The simulation results were also the primary input for the first and second law thermodynamic analyses.

Energy and Availability Analyses

For a complete and rigorous comparison between 4-stroke and 6-stroke cycles, a thermodynamic analysis of the engine cylinder was completed. The thermodynamic analysis focused on a control volume surrounding the cylinder as shown in Figure 8. A first law of thermodynamics energy balance was already computed as part of the 1-D simulations, since all of the necessary mass flows, work, and heat transfer quantities were part of the fundamental simulation calculations.

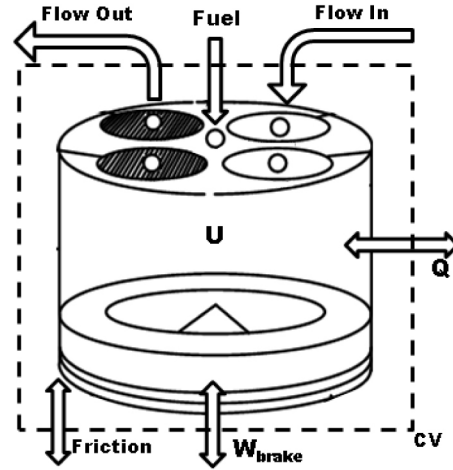


Figure 8. Schematic of the engine cylinder showing the control volume boundary and the crossing mass and energy flows.

A second analysis focused on a system availability (or exergy) analysis. The availability of a system, in relation to a reference environment, is the maximum theoretical work possible as the system and reference environment interact until equilibrium, [19]. The second law analysis utilized the 1-D cycle simulation results listed in the previous subsection. A cylinder control volume methodology similar to the ones presented in Anderson et al. [20] and Edwards et al. [21] is described as follows.

Thermodynamic properties were computed based on ideal gases from the NASA databases [22] and were a function of pressure, temperature, and equivalence ratio. For a control volume the availability rate balance takes the form of equations (1) and (2), where the streams are considered positive when entering the control volume.

$$\dot{A}_{cv} = -\dot{A}_{heat} - \dot{A}_{work} + \dot{A}_{in} - \dot{A}_{out} - \dot{A}_{destroyed} \quad (1)$$

$$\frac{dA_{cv}}{dt} = \sum_j \left(1 - \frac{T_0}{T_j} \right) \dot{Q}_j - \left(\dot{W}_{cv} - p_0 \frac{dV_{cv}}{dt} \right) + \sum_i \dot{m}_i a_{fi} - \sum_e \dot{m}_e a_{fe} - \dot{A}_{destroyed} \quad (2)$$

In equation (1) the control volume availability rate of change is $\dot{A}_{cv}$, availability due to heat loss is $\dot{A}_{heat}$, and availability due to work is $\dot{A}_{work}$. The subscripts 'in' and 'out' define the availability from the mass flows in and out, and the availability rate of change destroyed is denoted as $\dot{A}_{destroyed}$.

At any given time throughout the engine cycle the control volume, or cylinder in the present case, has a level of availability defined by equation (3),