The Radical Novelties Engine Simulation:
An Example of its Operation

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This paper presents results generated by the Radical Novelties engine simulation. The purpose of the paper is to give the reader an insight into the unseen physics that occur within a piston engine. It also provides an understanding of what the Radical Novelties engine simulation actually models and simulates, and how the simulation can aid in the design of engines, intake systems and exhaust systems.

Piston engines have been in use for about 100 years. But even at this late stage of their existence, they are still developed and perfected using a trial and error approach. When a new engine is designed, a prototype is fabricated and then tested on a dynamometer. The designers then consider the results of the test and make educated guesses on what parts of the engine they should modify to bring its performance in line with its design requirements. Once they have decided on what to change, new components are drawn up and fabricated. Then the engine is reassembled and tested again. The cycle is repeated until the engine finally complies with its intended design. This process can take thousands of hours of testing, thousands of hours of design and fabrication work, and many calendar months. The goal of the engine simulation is to provide a tool that can look inside of an operating engine and allow a designer to solidly determine what must be changed and what should be left alone. In addition, it provides for the rapid setup of an engine and rapid modification of any of its parts. The delay of drawing and fabricating new parts is eliminated, making for fast modifying/testing cycles.

The simulation employs Computational Fluid Dynamics (CFD) to model the flows through an engine's intake and exhaust systems, through its intake and exhaust valves, through plenums, pipe connections, and mufflers, and also inside of the cylinders. The fluid flows are modeled using a compressible, time-accurate approach. An example of the manner in which the flows are modeled is given in Figure 1, illustrating the flow through a "collector", where three pipes come together. Two exhaust header pipes from cylinders connect at the top and left sides of the collector, and an exhaust pipe leading to the atmosphere connects at the right. Flow is coming

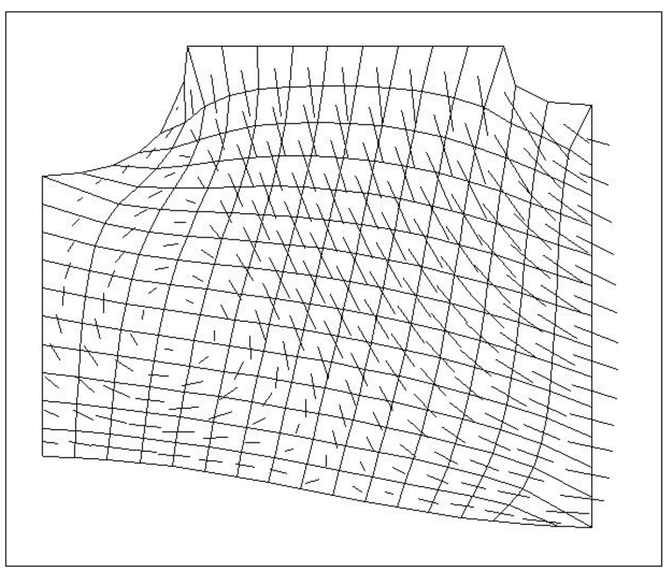

Figure 1: Exhaust Flow Through a Collector

from the top pipe and exiting to the right toward the atmosphere. Engine models are built in the simulation by assembling combinations of cylinders, pipes and collectors. The simulation can model an engine comprised of any number of cylinders in any configuration.

For testing the simulation itself, I often use a single-cylinder engine model. The single-cylinder model, with no intake or exhaust connections to other cylinders, has simple, distinguishable dynamics that make it useful as a testing and teaching device. It also offers fast test runs. The simulation can generate this engine's power curve, from 1,000 to 9,000 rpm, in about 12 minutes. In the following pages, I'll expose the physics and dynamics that occur inside of the single cylinder engine while it's operating at full throttle over a full range of rotational speeds. While the details are particular to this engine, the physics and dynamics you see described occur in all piston engines. In multi-cylinder engines, the dynamics are more complex, since the dynamics of one cylinder affect, and are affected by, the dynamics of the other cylinders.

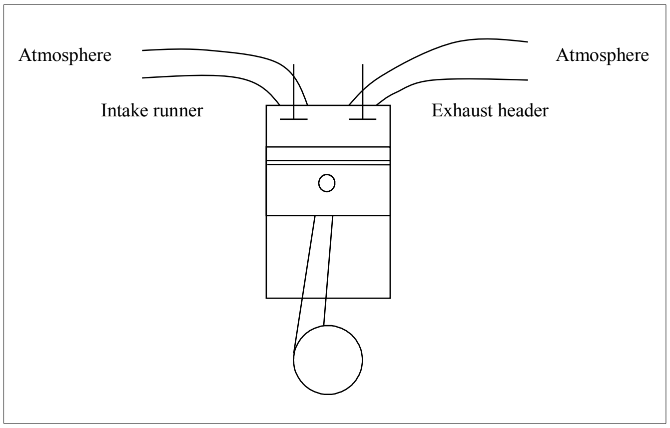

Figure 2: Single-Cylinder Engine Layout

Figure 2 shows the basic layout of the engine. It has a single intake and a single exhaust valve. These are connected to a single intake runner and single exhaust header, both of which are open to the atmosphere. The engine's detailed specifications are presented in Table 1.

Bore	3.46"	88.0 mm
Stroke	2.69"	68.0 mm
Displacement	25.29 in^3	413.6 cc
Compression Ratio	9.0:1	
Connecting Rod Length	5.46"	138.7 mm
Blowby Area	0.007 in^2	4.5 mm^2
Intake Valve Diameter	1.61"	41.0 mm
Intake Valve Lift	0.354"	9.0 mm
Intake Opens	323.1° past TDC of Compression	
Intake Closes	600.8° past TDC of Compression	
Intake Cam Profile	Catenoidal	
Exhaust Valve Diameter	1.44"	36.5 mm
Exhaust Valve Lift	0.252"	6.4 mm
Exhaust Opens	120.0° past TDC of Compression	
Exhaust Closes	383.9° past TDC of compression	
Exhaust Cam Profile	Catenoidal	
Intake Runner Diameter	1.51"	38.4 mm
Intake Runner Length	8.5"	215.9 mm
Exhaust Header Diameter	1.20"	30.6 mm
Exhaust Header Length	12.0"	304.8 mm

Table 1: Single-Cylinder Engine Specifications

The simulation's predicted power and torque curves for this engine are presented in Figure 3. The engine produces about 26 hp at 5,900 rpm. Torque reaches a peak of 27 ft-lb at 4,000 rpm. The torque curve is far from flat, which is typical for an engine with an open intake runner. The resulting power curve has a slightly concave front side, also typical for an engine in this configuration. (Adding an air box or air cleaner will typically flatten the torque curve and result in a more linear front to the power curve.)

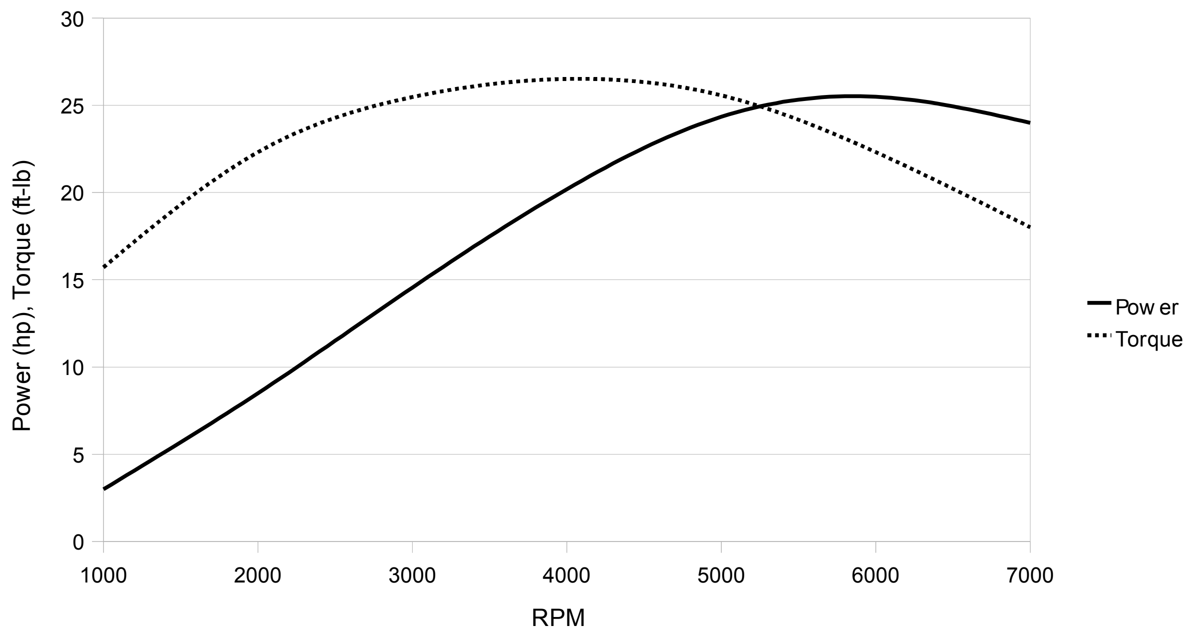

Figure 3: Predicted Power and Torque for the Single-Cylinder Engine

The shape of the torque curve is a direct result of the engine's intake and exhaust dynamics. If the dynamics of the intake and exhaust airflows could be eliminated from the engine, the torque curve would be perfectly flat, and the power curve would be linear. The power curve drops after reaching a peak. This happens on all engines, and is also a result of the intake and exhaust dynamics. In the case of this engine, the flow through the gap between the intake valve and its seat chokes (reaches Mach 1) at 4,000 rpm. This limits the mass flow rate through the gap. As rpm increases, the valve is open a shorter and shorter length of time, and so less and less total mass can flow through over the course of the intake stroke. This causes the torque to drop off above 4,000 rpm.

Figure 4 illustrates the density of the charge inside of the cylinder over the course of a single cycle (two crankshaft revolutions). Crankshaft rotation is from left to right, and is measured from Top Dead Center (TDC) of the compression stroke. (To make the curves easier to see, the figure starts at the beginning of the intake stroke, placing TDC of compression in the center.) The engine's four strokes are indicated across the top of the figure. The overall shape of the curves is a reflection of the compression of the charge after both valves have closed, followed by expansion of the charge through the power stroke.

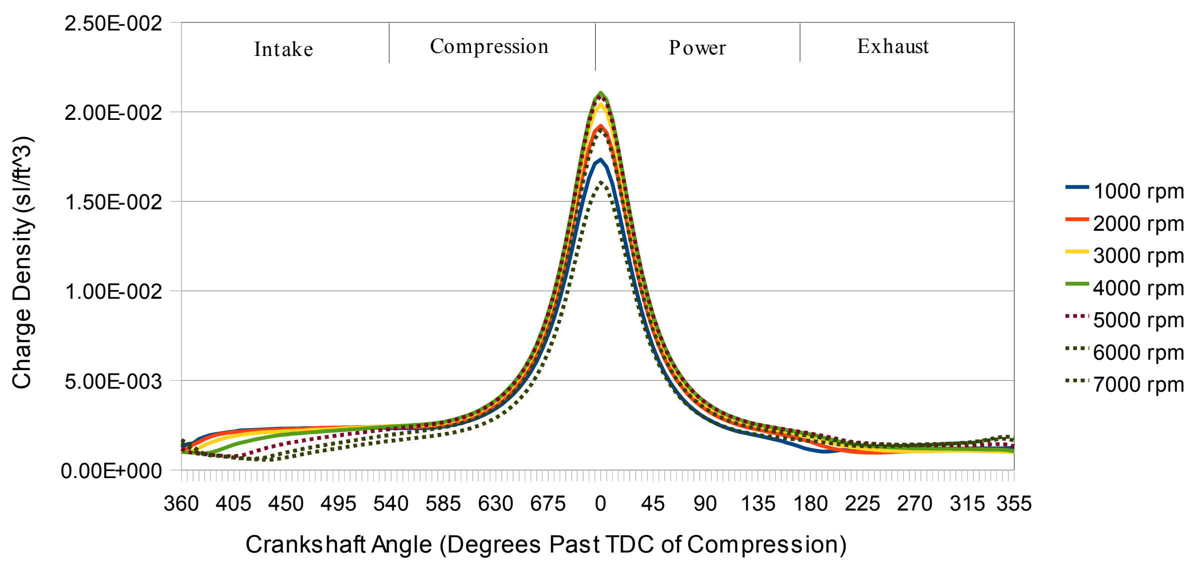

Figure 4: Density of the Charge Inside of the Cylinder

Note how the maximum density of the charge rises from 1,000 rpm to 4,000 rpm, and then drops again as the engine speed increases. These changes in maximum density with engine speed are caused by the intake and exhaust dynamics. The intake dynamics drive more air/fuel mixture into the cylinder at 4,000 rpm than at any other operating speed. If there were no intake and exhaust dynamics, we would see a single curve, the cylinder being filled exactly the same at every engine speed. That would also yield a perfectly flat torque curve, and a linear power curve with no peak. This shows how intake and exhaust dynamics dominate the performance and operating character of any real engine.

Figure 5 illustrates the total mass of fuel/air mixture drawn into the cylinder over the course of the intake cycle. The intake valve opens at the left end of the figure and closes at the right. Mass flow *into* the cylinder is *negative*. Note how, at 4,000 rpm, more mixture is drawn into the cylinder than at any other rpm. This results in a higher density of charge, creating the torque peak at this rpm.

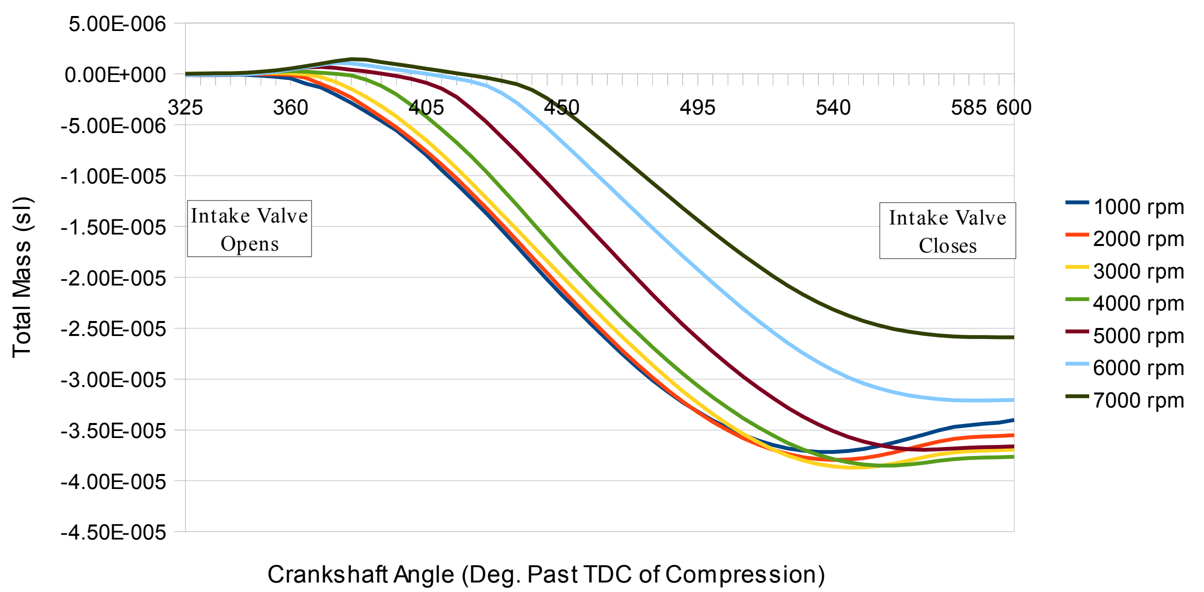

Figure 5: Total Mass of Fuel/Air Mixture Drawn into the Cylinder

Also note how, at many rpms, some of the intake charge flows back out of the cylinder before the valve closes. And how, at some rpms, some residual charge from the previous combustion cycle flows out of the cylinder and into the intake system right after the valve opens. This is the reason engines develop carbon deposits in their intake systems.

Figure 6 illustrates the temperature of the charge inside of the cylinder over a two-revolution cycle. Crankshaft rotation is again from left to right, with TDC of compression in the center. The sharp jump in temperature is ignition of the charge. One thing to note here is the general affect of increasing rotational speed. As the rpm rises, each stroke of the engine occurs over a shorter length of time. And so as engine speed rises, there is less time for heat to transfer from the charge to the cylinder walls and vice-versa. As engine speed rises, the charge stays hotter and the engine is more efficient. At TDC of the compression stroke, the temperature of the charge rises from 938°R at 1,000 rpm to 1,122°R at 7,000 rpm. The increase in temperature before ignition leads to an even larger increase in flame temperature, from 4,260°R at 1,000 rpm to 4,619°R at 7,000 rpm. Interestingly, engine speed seems to dominate here. We don't see the temperatures rising with rpm and then dropping, as the power curve does. It appears that thermal loads and stresses are dominated by engine speed, not intake and exhaust dynamics. The rpm-to-rpm variation in temperature over the power stroke is also related to the timescale of the strokes. With the high temperatures of combustion, heat transfers faster, and the difference in temperature with engine speed is even more pronounced. Note how the curves fall nicely, one above the other through the power stroke. The figure is periodic, so you can follow each curve around, from the right side of the curve to the left. On the left side, note how the charge temperature drops as cool, ambient air at 531° R is drawn in during the intake stroke.

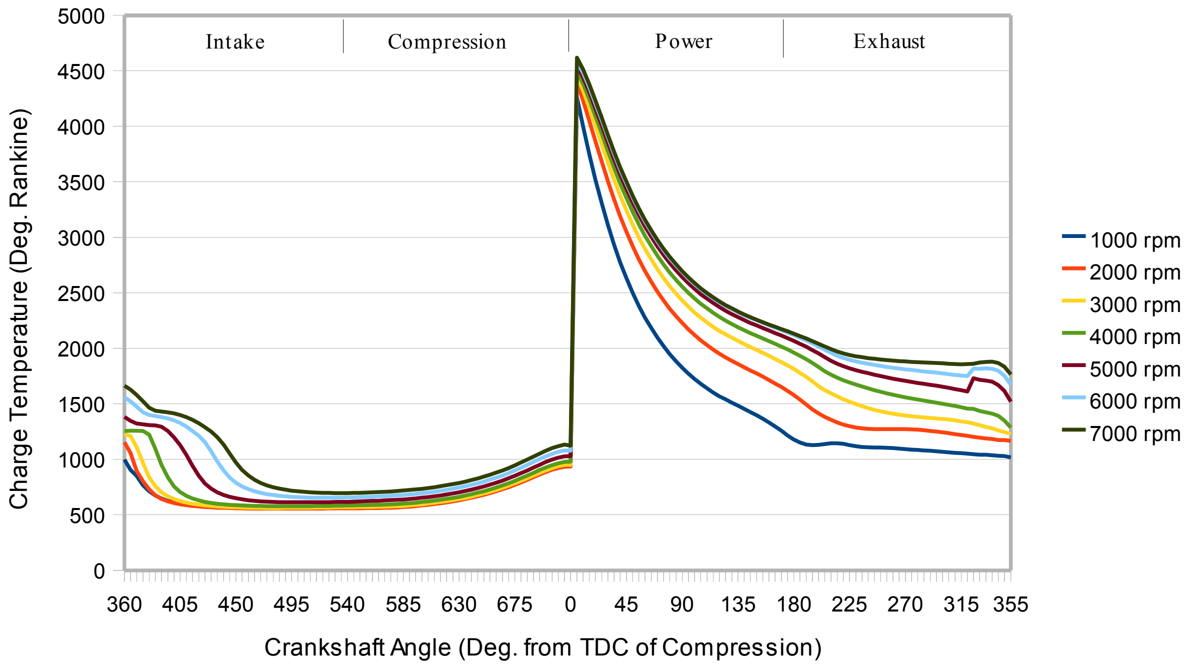

Figure 6: Absolute Temperature of the Charge in the Cylinder

Figure 7 illustrates the pressure of the charge inside of the cylinder over the two-revolution cycle. The variation in pressure from rpm to rpm is a reflection of the dynamically driven variation in density that we saw earlier. This figure shows how strongly the intake and exhaust dynamics affect performance. The maximum pressure inside of the cylinder goes from about 875 psi (58 bar) at 1,000 rpm to 1,133 psi (76 bar) at 4,000 rpm. It then falls back down to 884 psi (59 bar) by 7,000 rpm. Since the piston surface area is constant, these curves also reflect the force on the piston and therefore the torque and power that the engine is generating. Work performed over a single cycle is proportional to the area under each of these curves. Note the difference in area between the 1,000 and 4,000 rpm curves. This emphasizes how getting the intake and exhaust dynamics tailored correctly is critically important to any engine design. Getting them wrong can be highly detrimental. The pressure values also tell us about the forces at work inside of the cylinder. At 4,000 rpm the peak load on the piston and connecting rod is about 10,600 lb. Astonishing.

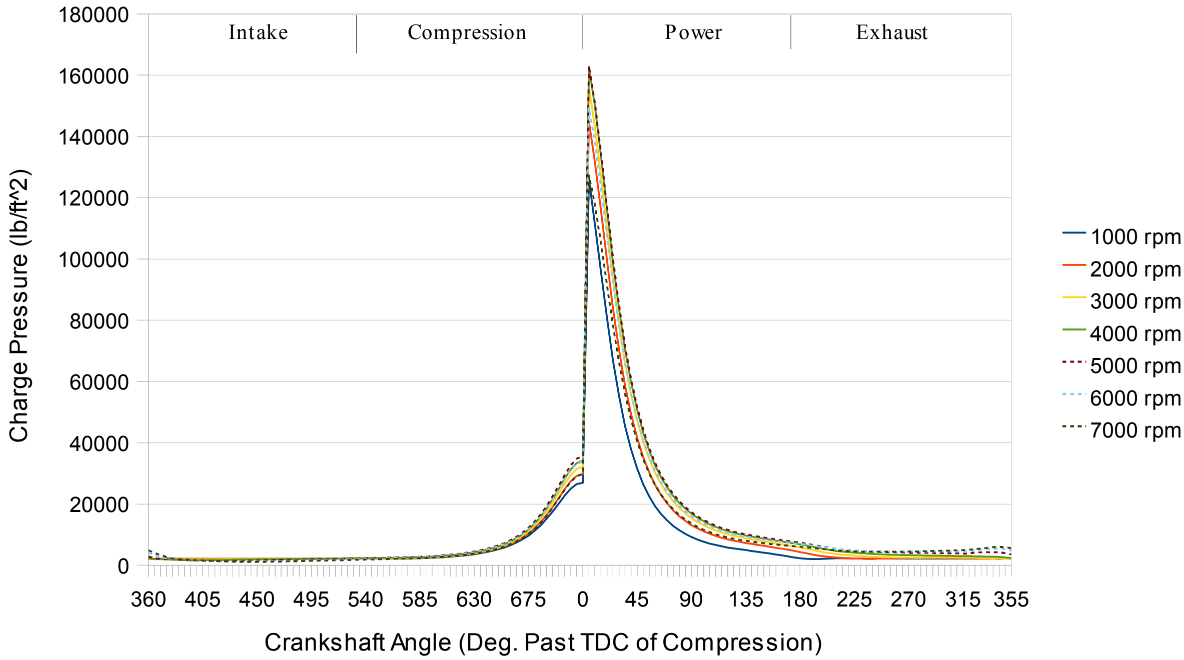

Figure 7: Absolute Pressure of the Charge in the Cylinder

Here is where things really begin to get interesting. Figure 8 shows a three-dimensional representation of the mass flow rate through the entire intake tract—through both the intake valve and the intake runner--at 4,000 rpm. The X-axis represents crankshaft angle. From left to right, the figure spans a single cycle—two crankshaft revolutions. TDC of compression is now at the left edge of the figure, rather than in the center. The Z-axis represents mass flow rate. A *negative* value indicates flow *into* the cylinder. The Y-axis represents the stations along the connected intake valve and intake runner. The near edge of the figure represents the atmospheric end of the runner, and the far edge of the figure represents the cylinder end of the valve. It's easy to see where the intake valve both opens and closes. When the valve opens, flow begins to move inward all along the intake tract. Then as the valve continues to open farther, the rate increases rapidly. As the valve closes, the rate decreases. At valve closure, the air/fuel mixture inside of the intake runner has residual momentum; the mixture at the atmospheric end is still moving toward the cylinder. This causes oscillations to form in the runner just as they do in an organ pipe. In this case, we see a "quarter-wave pipe", with a node at the cyliner end and an antinode at the atmospheric end. The organ pipe oscillations are evident at the near edge, beginning to the right of the point where the valve is closed. The figure is periodic, and so the right-hand edge and the left-hand edge connect. Following around to the left edge of the figure, note how the organ pipe oscillations continue, slowly damping out as time passes.

Figure 9 shows the exhaust flow at 4,000 rpm, represented in the same manner. Here, flow *out* of the cylinder is *positive*. Again, we can see the opening and closing of the exhaust valve. And again, we can see organ pipe oscillations form after the valve is closed. These oscillations are a dominant part of the physics of all piston engine intake and exhaust flows. In this exhaust figure, we can also see one flow feature that isn't present on the intake side. On the atmospheric edge of the figure, there is a notch that occurs as the exhaust flow rises. That notch is preceded by a small hump, and both the hump and notch can be seen rising along the left surface of the figure. This is the passage of a shock wave. As the valve opens, the flow through the gap between the

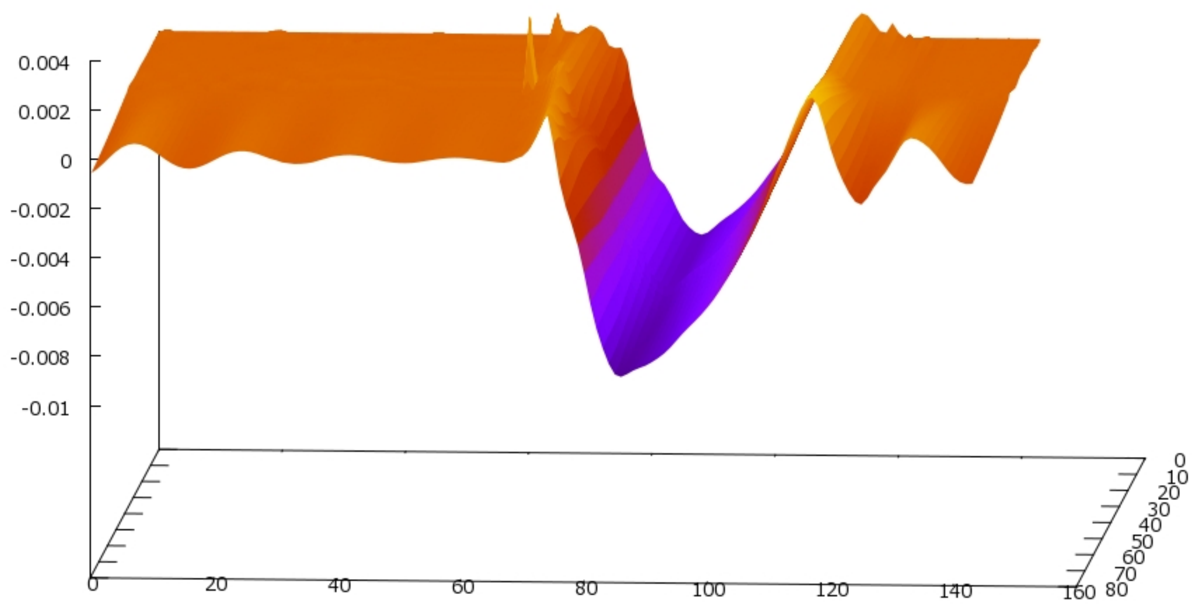

Figure 8: Mass Flow Rate Through the Intake Tract at 4,000 rpm

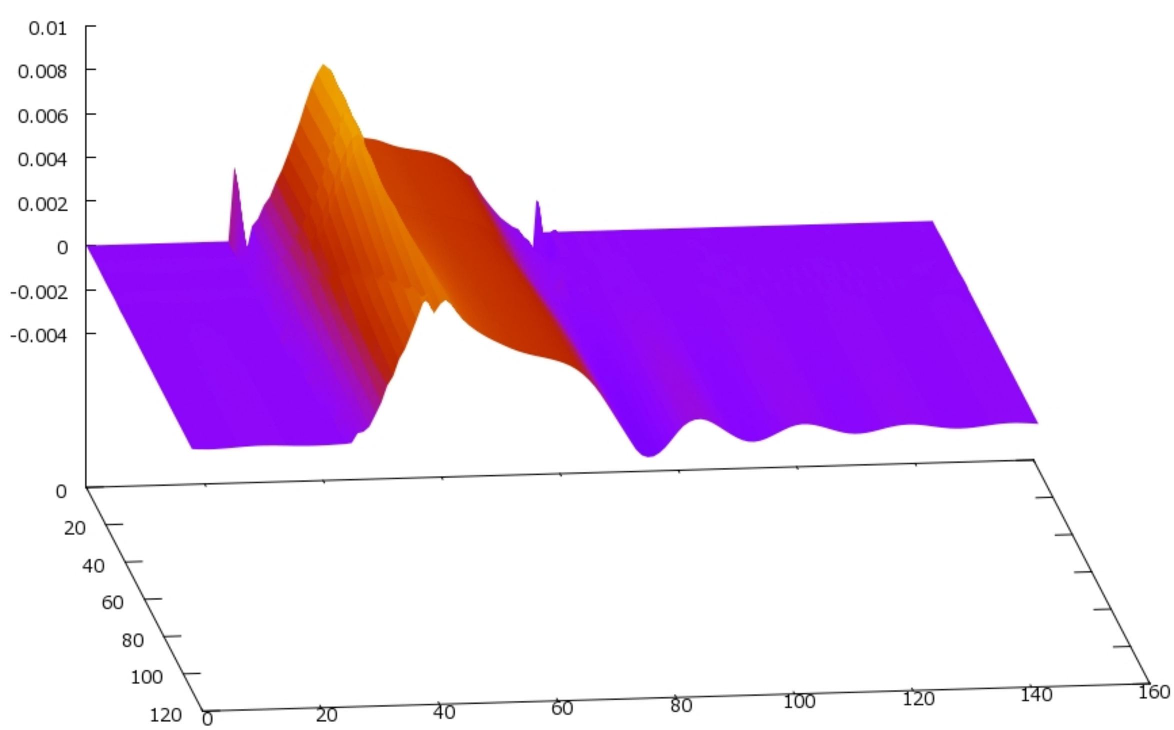

Figure 9: Mass Flow Rate Through the Exhaust Tract at 4,000 rpm

valve and its seat goes supersonic. For a short time, a standing shock wave forms just downstream of the valve. Then as the valve continues opening, the shock wave detaches and flows downstream, eventually exiting the exhaust pipe.

The plots for other rpms are similar to these, and they dispel a commonly held myth that, especially at high speeds, the flow through an engine is nearly continuous. These figures show that it isn't. The flow into and out of the cylinder stops between valve openings. In fact, between valve openings, because of the organ pipe oscillations, the flows actually reverse many times, gases moving *both* into and out of both tracts. Consider the effect these dynamics have on a carbureted engine. With the intake valve open, air flows into and through the carburetor, entraining fuel vapor as it passes. After the valve closes, the mixture near the carburetor is driven back and forth through the carburetor, being *re*carbureted many times because of the organ pipe oscillations. This generates a slug of intake charge near the carburetor that is overly rich with fuel.

It's illuminating to look at the mass flow rate at the atmospheric end of both pipes, and compare those values over the rpm range. Figure 10 illustrates the mass flow rate measured at the intake runner's atmospheric end, over a single cycle. This corresponds to the near edge of the surface plotted in Figure 8. Crankshaft rotation is from left to right. TDC of compression is again at the left end. Mass flow *into* the cylinder is *negative*.

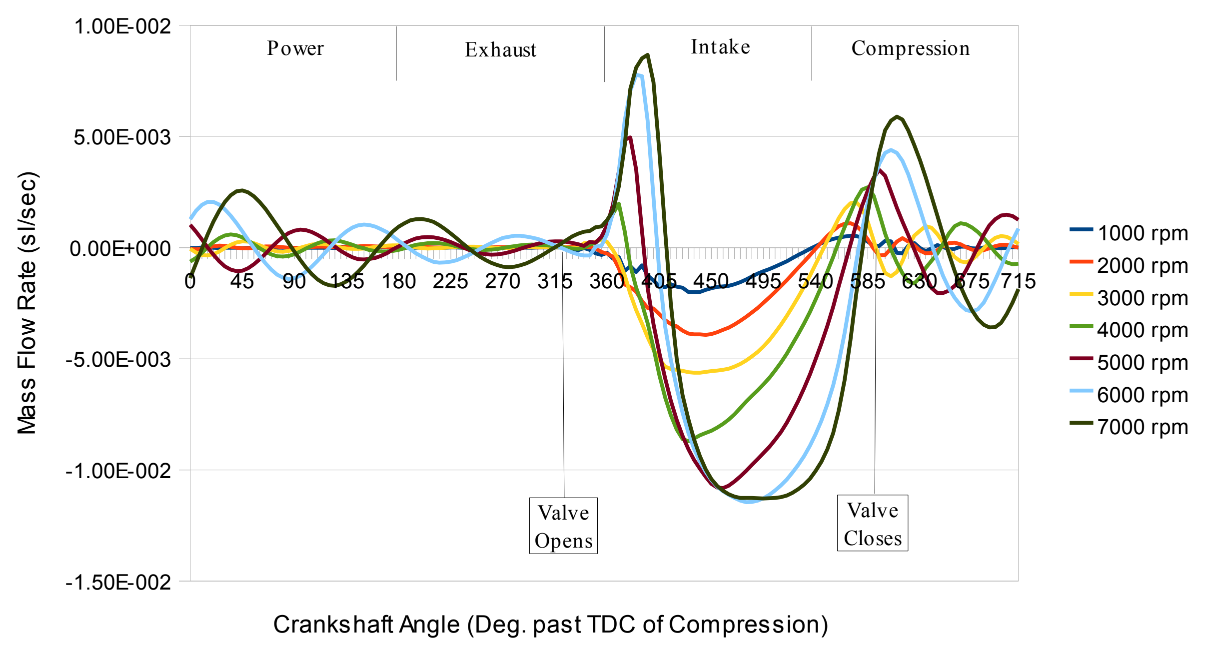

Figure 10: Intake Mass Flow Rates

Start with the 1,000 rpm curve. After the intake valve opens, the mass flow rate begins to grow negative (charge flows into the cylinder) as the piston starts to drop on its intake stroke. As the piston accelerates down (between 360° and 450°), the flow grows more negative, charge flowing more strongly into the cylinder. Then as the piston decelerates (from 450° to 540°), approaching the bottom of its stroke, the flow lessens again. Note the small oscillations in the curve shortly after the valve opens. The volume of the cylinder, connected to the intake pipe, forms a Helmholtz resonator. These small oscillations are Helmholtz oscillations. Note how their wavelength increases as the piston falls and the cylinder volume grows. The flow rate decreases and then stops when the piston reaches the bottom of its stroke at 540°. But the intake valve is still open, and as the piston begins to rise, some of the charge flows back out again, into the intake tract until the valve closes. In a carbureted engine, this regurgitation of the charge forces some part of the mixture within the intake tract back through the carburetor, fueling it a second time and making it overly rich. On the next intake stroke, that same mass of charge will be fueled a *third* time, making it even richer still. This is the reason that engines with high-duration/high-lift cams run rich at low rpm and exhibit a loping idle. After the valve closes, small organ pipe oscillations form, then damp out, the flow in the runner eventually going quiescent (follow around to the left end of the figure). At 2,000 rpm we see similar behavior, but with larger magnitude. The organ pipe oscillations just damp out before the valve reopens on the next cycle.

With increasing rpm, the activity becomes far more energetic. At 3,000 rpm, we can see that the flow rate is still negative when the piston reaches the bottom of its stroke at 540°. At this rpm the intake flow in the intake runner has so much momentum that it continues into the cylinder even after the piston stops and reverses its direction. This causes the pressure in the cylinder to rise well above atmospheric. These are the intake dynamics that we mentioned earlier. Because the intake charge has mass, it doesn't react instantaneously to the movement of the piston. As the engine's rpm rises, the intake flow gets more energetic, and more mass can be driven into the cylinder than at low rpm. Returning back to Figure 5, we can see this effect in the total mass curves for 3,000 and 4,000 rpm. The intake dynamics cause the pressure in the cylinder at intake valve closure to be higher, and therefore the pressure at TDC of the compression stroke is also higher. And that results in more torque being delivered during the power stroke. The rpm at which the most mass is driven into the cylinder always corresponds to the rpm of the torque peak.

Beginning at 4,000 rpm, we can see that there is still some residual pressure left in the cylinder after the exhaust stroke. Just after the intake valve opens, a puff of the residual exhaust charge actually flows out of the cylinder and into the intake runner, just as we saw in Figure 5. Then the flow reverses and flows strongly into the cylinder.

Since this figure represents two revolutions, each curve, happening over a shorter length of time, represents a different timescale. The frequencies of the organ pipe oscillations from 1,000 through 7,000 rpm are actually identical—250 Hz. Why? Because the length of the intake runner dictates these dynamics.

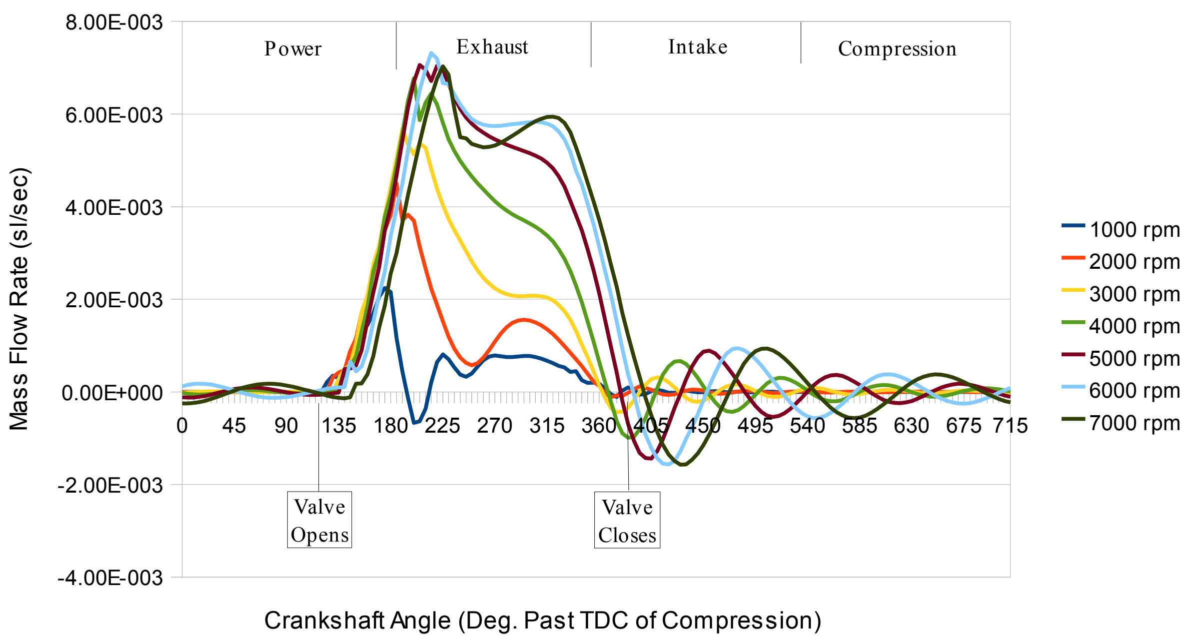

Figure 11: Exhaust Mass Flow Rates

Figure 9 illustrates the exhaust mass flow rate measured at the exhaust header's atmospheric end. Mass flow *out* of the cylinder is *positive*. Let's start with the 1,000 rpm curve. Here, just after the valve opens, we see Helmholtz oscillations again, but much stronger than on the intake side. When the exhaust valve opens at 120°, the pressure in the cylinder is at about 40 psi (2.7 bar). That's well above atmospheric pressure, and so the charge starts rushing out immediately, while the power stroke is still completing and before the exhaust stroke begins. The prompt rush of exhaust gases vents off the cylinder's over-pressure, inducing significant velocity in the exhaust header pipe. Because of the momentum of the gases in the header pipe, the flow through the pipe continues after the pressure in the cylinder has equalized with atmospheric. The pressure in the cylinder continues dropping, and the flow eventually reverses. This is the same kind of oscillation that occurs when a cork is pulled from a wine bottle. Note how the wavelength shrinks as the piston rises over the exhaust stroke, reducing the volume of the cylinder and raising the frequency of the system. When the valve closes, the oscillations revert to the organ-pipe variety, and damp out quickly. We see this same activity at 2,000 rpm. At 3,000 rpm, we don't see any Helmholtz oscillations, because the entire exhaust stroke occurs in less than a single Helmholtz period.

Looking at the 4,000 rpm curve, note the notch near 190°. This is the passage of the shock wave that we identified in Figure 9. Beginning with the 3,000 rpm curve, note how the flow reverses, and some of the expended exhaust gases reenter the cylinder before the exhaust valve closes.

It's clear that piston engines are aerodynamics devices. Their performance and character are molded by the dynamics—*the aerodynamics*—of the flows through their intake and exhaust tracts.

As an exercise, I decided to take this engine model and modify it so that it would become a single-cylinder version of the current Formula 1 (F1) engines. However, going into the exercise, I had no idea how to go about this. I didn't know what valve, cam, intake or exhaust parameters are typically found in F1 engines. Using the simulation, I arrived at the parameters listed in table 2.

Bore	3.46"	88.0 mm
Stroke	2.69"	68.0 mm
Displacement	25.29 in^3	413.6 cc
Compression Ratio	***15.0:1***	
Connecting Rod Length	5.46"	138.7 mm
Blowby Area	0.007 in^2	4.5 mm^2
Number of Intake Valves	***2***	
Intake Valve Diameter	***1.4"***	***35.6 mm***
Intake Valve Lift	***0.708"***	***18.0 mm***
Intake Opens	***340.0° past TDC of Compression***	
Intake Closes	***620.0° past TDC of Compression***	
Intake Cam Profile	Catenoidal	
Number of Exhaust Valves	***2***	
Exhaust Valve Diameter	***1.25"***	***31.8 mm***
Exhaust Valve Lift	***0.504"***	***12.8 mm***
Exhaust Opens	120.0° past TDC of Compression	
Exhaust Closes	***410.0° past TDC of compression***	
Exhaust Cam Profile	Catenoidal	
Intake Runner Diameter	***2.82"***	***71.6 mm***
Intake Runner Length	***5.5"***	***139.7 mm***
Exhaust Header Diameter	***2.24"***	***56.9 mm***
Exhaust Header Length	***12.5"***	***317.5 mm***

Table 2: F1 Single-Cylinder Engine Specifications

Modified parameters are presented in bold italic. All others are the same. Note that the modified engine has four valves, rather than the original engine's two. The resulting power curve is presented in Figure 12. The engine now generates 146 horsepower at 18,000 rpm. Torque reaches a peak of 49 foot-pounds at 14,000 rpm. An engine with six of these cylinders would displace 2.5 liters and generate 876 peak horsepower. That's very close, per unit displacement, to current Formula 1 engines, which generate over 800 horsepower with 2.4 liters and 8 cylinders.

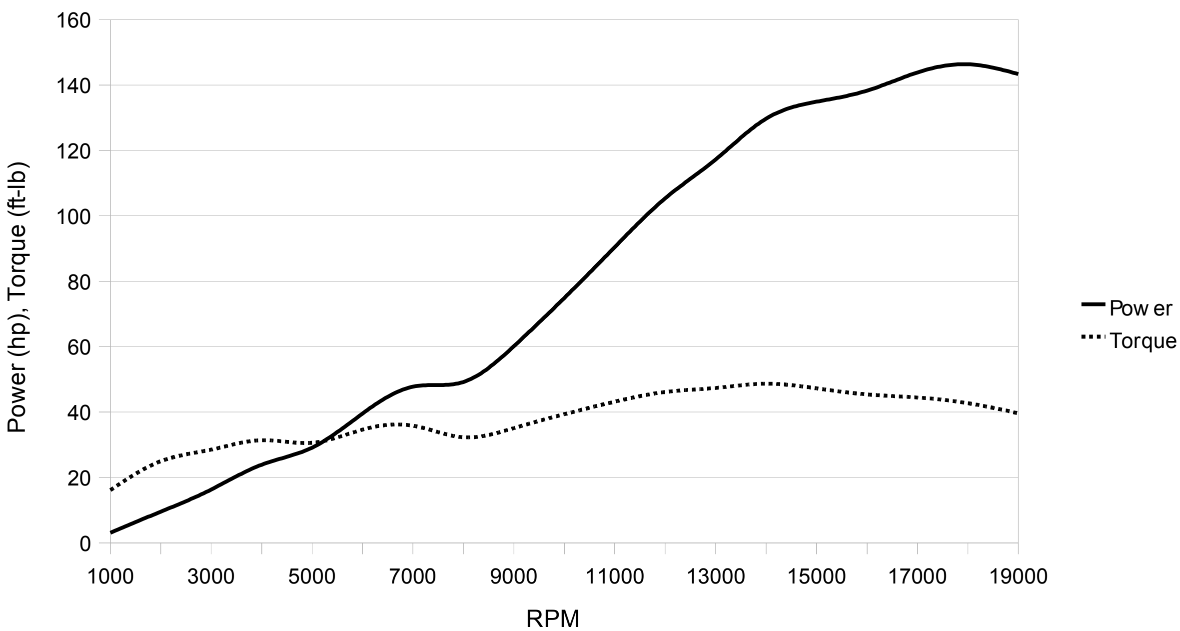

Figure 12: Power and Torque Curves for the "Formula 1", Single-Cylinder Engine

Modifying this engine and arriving at this new set of parameters took about *90 minutes* using the engine simulation. If this same exercise were performed with a real engine and a dynamometer, it would have likely taken *months* of work, as the engine is tested, new parts are drawn up and fabricated, and the engine is tested again. This clearly shows the value of the simulation for engine development.