Internal Combustion Engine Model

This document describes a Sinaps-based SINDA/FLUINT model of a four-cycle, inline six-cylinder gasoline internal combustion (IC) engine.

The purpose of this model is to demonstrate modeling capabilities and techniques. The model is intended to be representative of a typical design, but is otherwise notional. It is intended to provide a starting point for a user requiring a more detailed analysis of an actual engine design.

The focus of the problem is on the very short time-scale events including pressure waves in the intake and exhaust runners. Details of flows, combustion, and heat transfer within the cylinder itself have been greatly simplified to preserve the focus on the air supply and exhaust systems.

This model was developed as a by-product of an investigation of fast-transient interactions within a turbocharged engine. That variation, including different manifold geometries appropriate to a turbocharged design, is documented separately.

Basic knowledge of SINDA/FLUINT modeling is presumed.

The model is built parametrically using "registers" (user-defined parameters) whose names are indicated within the text using italics such as "*Kair_filter*." Many variations can be made simply by adjusting these values, while others will require changes to the thermal/fluid network and associated user-defined co-executing logic. The model was built using SI units, though some inputs were converted from English (US Customary) units.

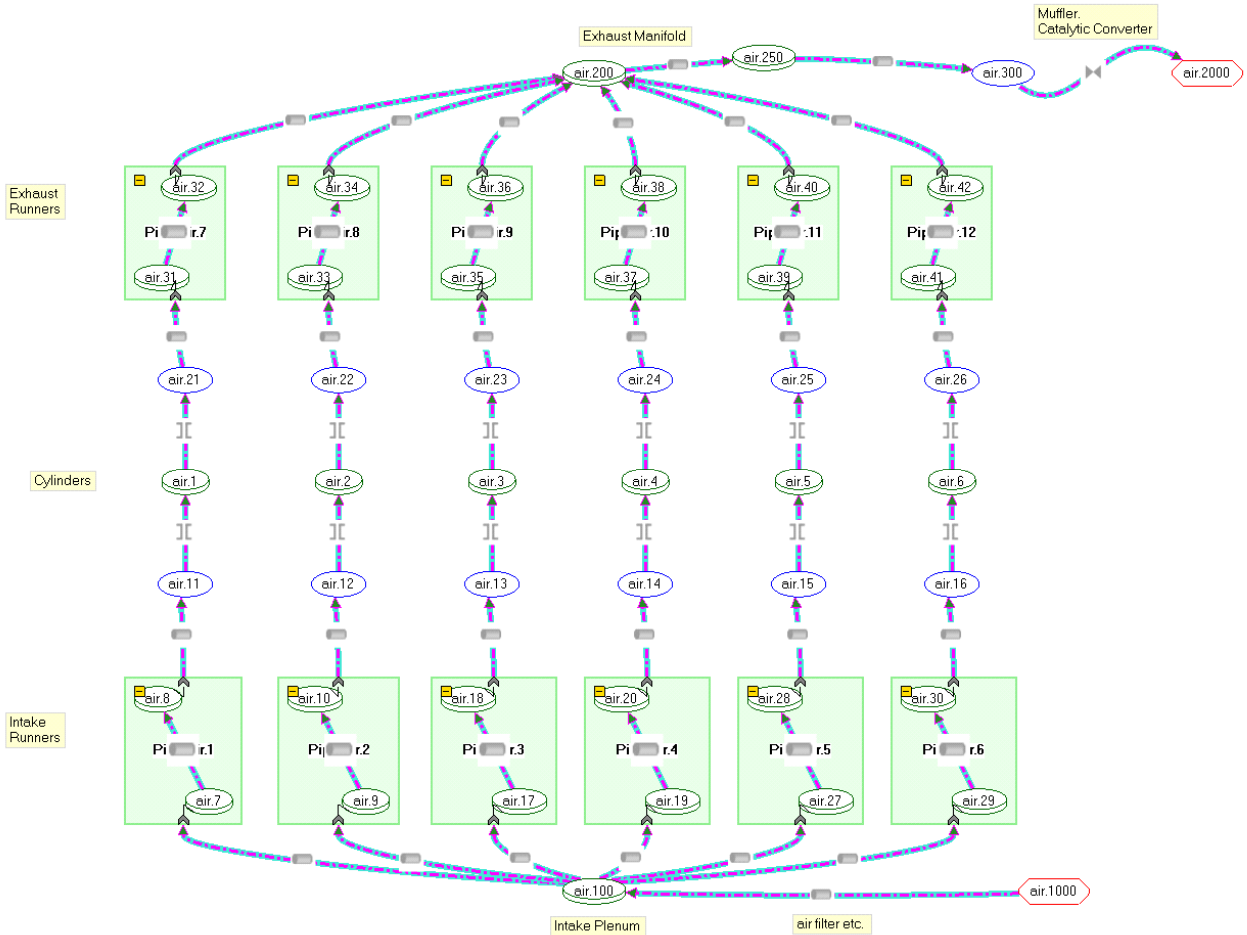
Engine Design Summary

The engine being considered is a 3.7L displacement inline six, with a single over-head camshaft.

The pistons are 93.4mm diameter, with a stroke of 90mm. The compression ratio is 9:1. The firing sequence is 1, 5, 3, 6, 2, 4.

Further details of the design are contained within the following description of the thermal/fluid mathematical representation.

Model Description

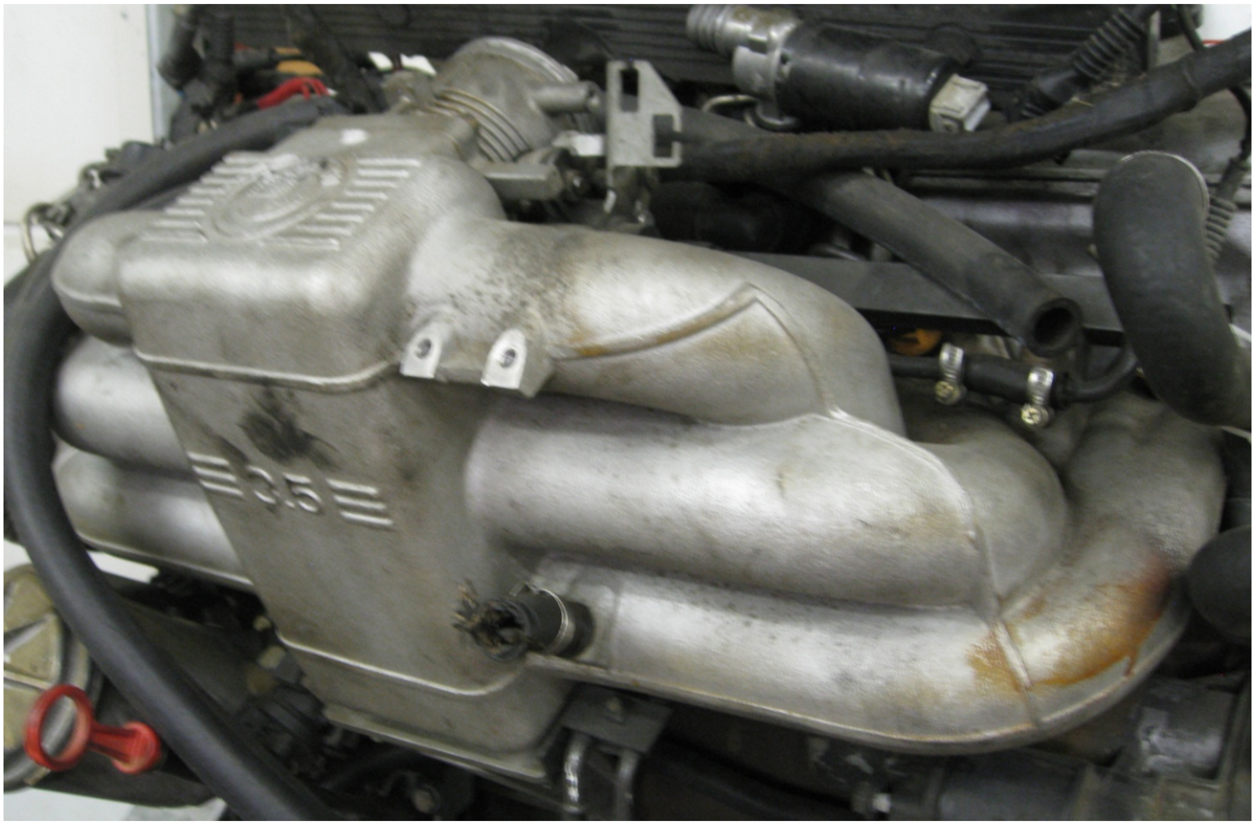


Sinaps Schematic (with cylinder heat transfer on an invisible layer)

Intake and Exhaust System

Ambient is 25°C, sea-level atmosphere. Intake is through a 4" (10.2cm) diameter (*Dintake*), 3' (91.4cm) long duct (*Lintake*). A notional air filter is present and is modeled simply as a K-factor of 20 (*Kair_filter*), which is a bit large only because it is referenced to the low dynamic head of the large-diameter intake. It was chosen to cause a pressure drop representative of typical air filters. The throttle is assumed to be wide open, with the engine RPM prescribed (i.e., an appropriate load is assumed at each operating point such that the details of the rest of the engine, vehicle, road, etc. can be avoided to maintain the focus of the model).

The intake manifold is a box-shaped volume that is 85x110x210mm in dimension. The intake runners are 500mm long (*Lin_runner*) and 45mm diameter (*Din_runner*). In the model diagram above, the intake ducts are located at the bottom of the schematic.



Representative intake manifold

The exhaust runners are 14” (35.6cm) long (*Lex_runner*) and 1.625” (4.13cm) diameter (*Dex_runner*). The exhaust “manifold” is a 1’ (30.5cm) long (*Lexhaust_man*) transition into an exhaust pipe.

Representative Exhaust Manifold, although 6-1 was used in the model rather than the 3-2-1 shown

The exhaust pipe is 10’ (3.05m) long (*Lexhaust_pipe*) by 2.25” (5.72cm) diameter (*Dexhaust_pipe*) with a K-factor of 3 (*Kexhaust_pipe*) to represent bends and other fittings. The notional muffler and catalytic converter are lumped into a K-factor of 4 (*Kmuffler*) as needed to have reasonably representative pressure losses.

The manifolds are modeled as single control volumes (FLUINT "tanks"). The intake manifold is stagnant (zero velocity), whereas the exhaust plumbing maintains the velocity (static pressure).

The runners are coarsely discretized (two-segment simple centered "Pipes"). FLUINT tanks and inertial FLUINT "tubes" are used to represent these runners as needed to capture acoustic waves (pressure delays), as are other significant volumes and inertial flow paths within the system. The time step is limited (via a call to the routine WAVLIM) to capture these acoustic pressure responses.[*]

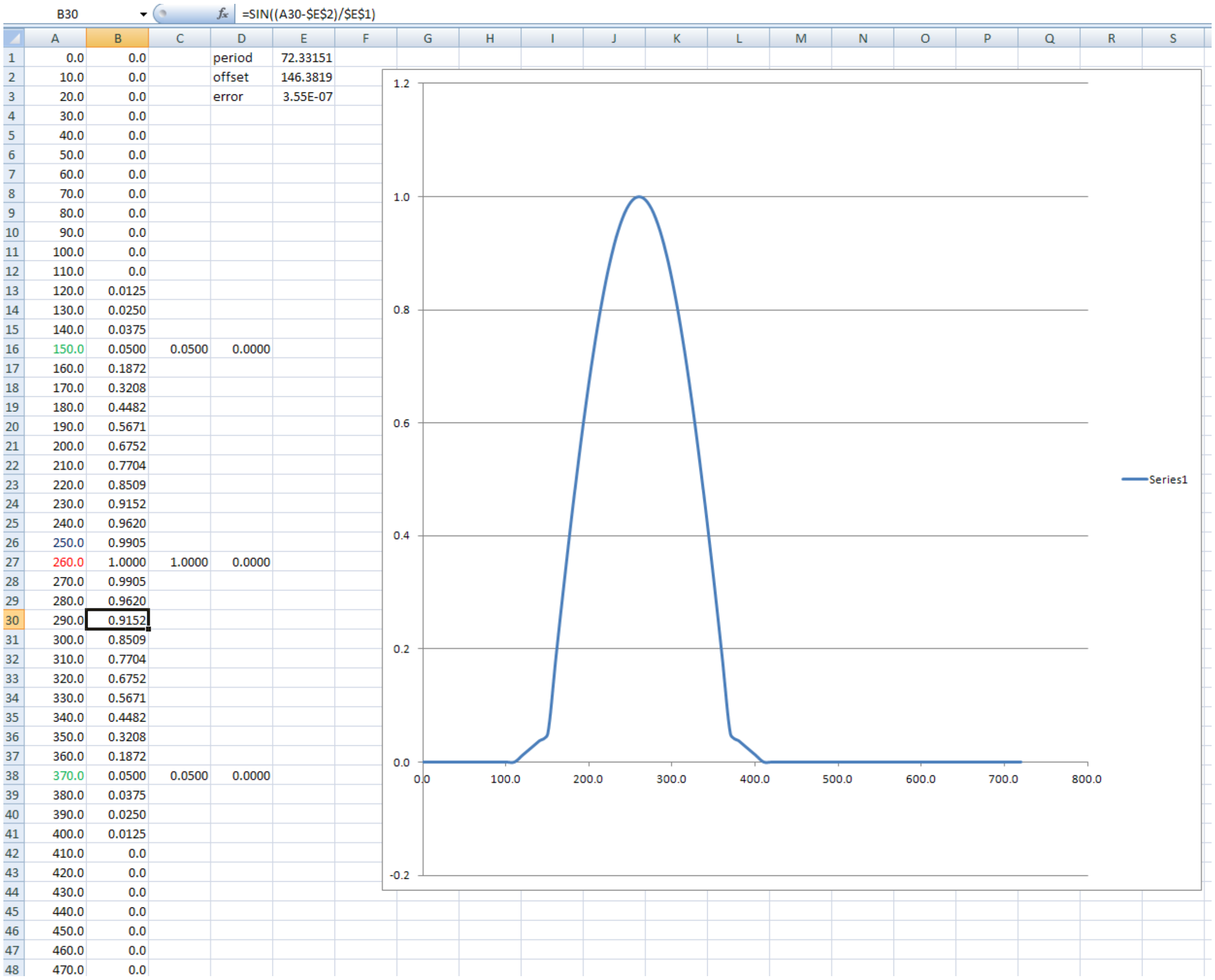
The pressure and velocity profiles within these runners are critical to the predicted efficiency of this engine. In a model used for finalized predictions, higher resolutions than 2 would be required (say 5), as would additional care placed on the specification of cross-sectional changes and definition of secondary flow losses. In this demonstration model, however, a simpler treatment is satisfactory.

All air passages were assumed adiabatic. This is a bit ironic because conjugate heat transfer and structural/thermal predictions are a forte of the CRTech suite of tools. But in this case, heat transfer was deemed a distraction since the focus was on air pressures. Since this model can only operate in a transient (given the way the cylinders were modeled, as described below), adding thermal structures would require either a very good initial condition, or use of a long transient to converge on a cyclically repeatable answer.

Valve and Camshaft

Valve lift profiles are represented as partial sinusoids, starting and ending with linear transitions to and from the closed positions. A Microsoft Excel® sheet (supplied) provides the normalized profiles, which are then cut and pasted into Sinaps Tables.

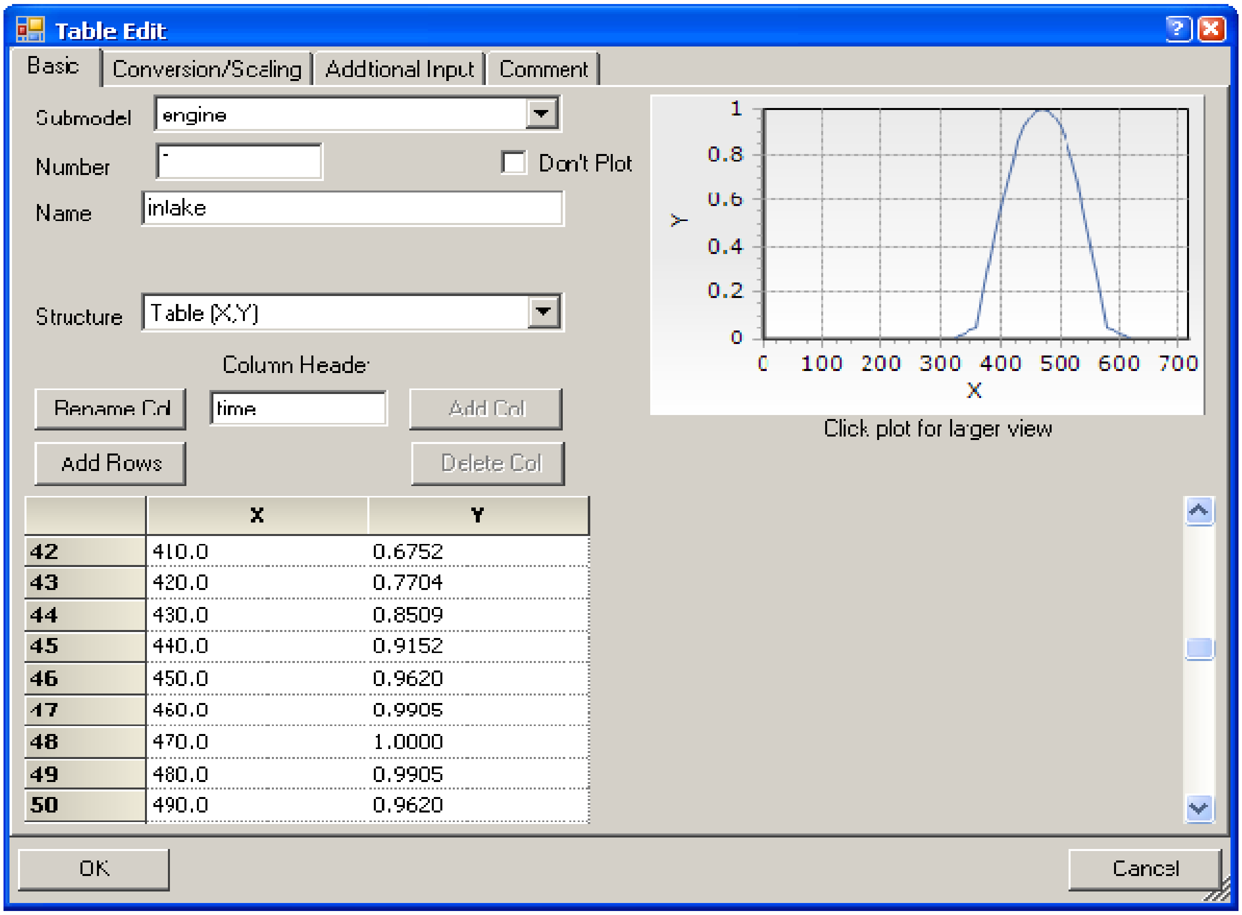
[*] However, another even smaller limit on the crank angle change per time step, as needed to track piston volume changes and valve motions, actually limits the time step most of the time in this model.



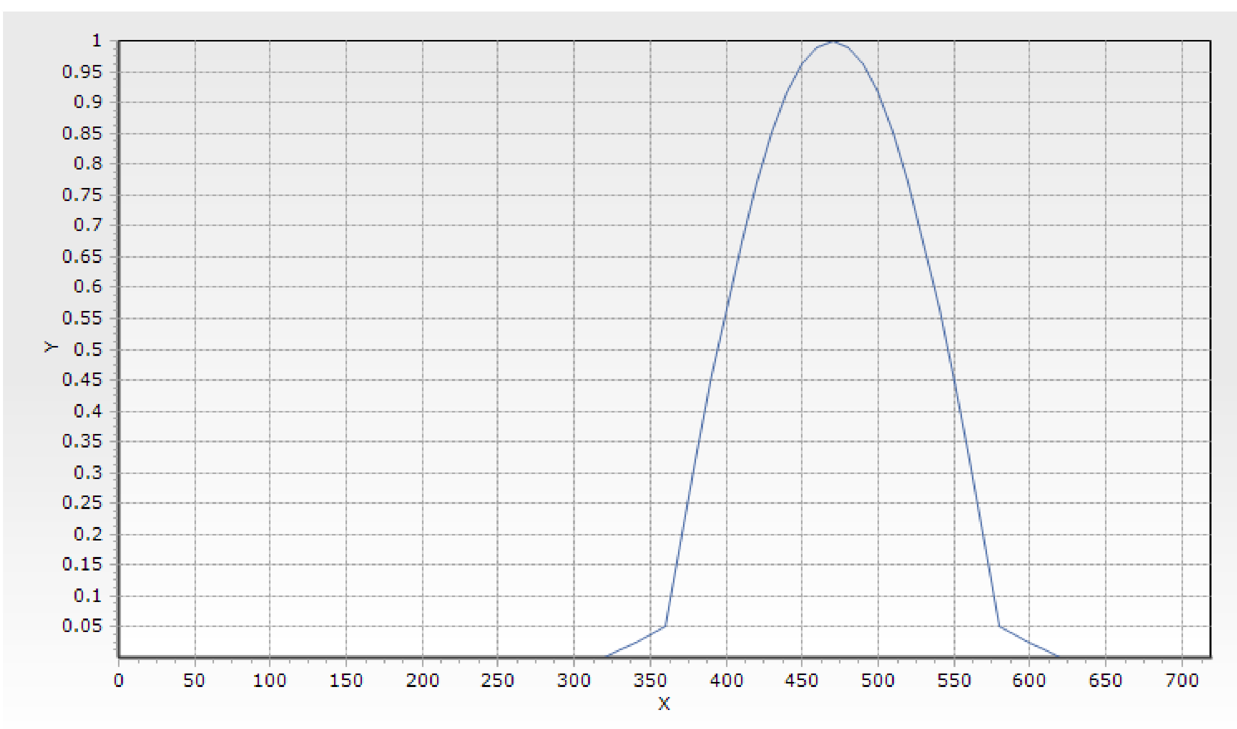
B30 =SIN((A30-E2)/E1)

	A	B	C	D	E
1	0.0	0.0		period	72.33151
2	10.0	0.0		offset	146.3819
3	20.0	0.0		error	3.55E-07
4	30.0	0.0			
5	40.0	0.0			
6	50.0	0.0			
7	60.0	0.0			
8	70.0	0.0			
9	80.0	0.0			
10	90.0	0.0			
11	100.0	0.0			
12	110.0	0.0			
13	120.0	0.0125			
14	130.0	0.0250			
15	140.0	0.0375			
16	150.0	0.0500	0.0500	0.0000	
17	160.0	0.1872			
18	170.0	0.3208			
19	180.0	0.4482			
20	190.0	0.5671			
21	200.0	0.6752			
22	210.0	0.7704			
23	220.0	0.8509			
24	230.0	0.9152			
25	240.0	0.9620			
26	250.0	0.9905			
27	260.0	1.0000	1.0000	0.0000	
28	270.0	0.9905			
29	280.0	0.9620			
30	290.0	0.9152			
31	300.0	0.8509			
32	310.0	0.7704			
33	320.0	0.6752			
34	330.0	0.5671			
35	340.0	0.4482			
36	350.0	0.3208			
37	360.0	0.1872			
38	370.0	0.0500	0.0500	0.0000	
39	380.0	0.0375			
40	390.0	0.0250			
41	400.0	0.0125			
42	410.0	0.0			
43	420.0	0.0			
44	430.0	0.0			
45	440.0	0.0			
46	450.0	0.0			
47	460.0	0.0			
48	470.0	0.0			

Excel-based generation of a normalized exhaust valve profile



Sinaps-based table, cut and paste from Excel (Intake Valve shown)



Pop-up (expanded) view of Intake Valve Normalized Profile. Top Dead Center (TDC) is X=360 degrees

These normalized profiles are interpolated as a function of crank angle (*angle*, which ranges from 0 to 720 degrees), then multiplied by the actual lift (10.9mm for the intake valve, *i_valve_lift*, and 10.8mm for the exhaust valve, *e_valve_lift*).

The valve diameters are 47mm for the intake valve (*i_valve_diam*) and 38mm for the exhaust valve (*e_valve_diam*).

A single profile for each type of valve is applied to all valves using a phase shift of 120 degrees (see registers *phase1* through *phase6*, which also control the shift in crank angle and firing sequence for each cylinder). For example, listed below is the network-based user logic controlling the exhaust valve (modeled as an orifice) in cylinder #3 in FLOGIC 0 (which is executed at the beginning of each time step):

```
call d11cyl(720.0,720+angle-phase3,engine.a2,atest)
aori#this = max(0.0,e_valve_lift*atest)*pi*e_valve_diam
```

D11CYL is cyclic linear interpolation of the lift profiles, stored in temporary variable “atest.” “AORI#THIS” is the aperture size of the current (“#this”) orifice.[†] Nearly identical logic is used for other cylinders, except the “phase3” shift is replaced with “phase2” for cylinder #2, etc. (*Phase2*=480 degrees, *phase3*=240 degrees for this firing sequence. Cylinder #1 fires at zero degrees, essentially acting as the reference point, so *phase1*=0 degrees.)

Lacking more specific data, the valves were modeled as ORIFICE devices, using the default correlations for sharp-edged orifices despite the annular nature of the aperture. If more specific data were available (e.g., test data of flow versus pressure drop for various positions), a TABULAR device would have been more appropriate.

Even though the performance of the valves is critical to the operation of the engine, all of the above treatments are arbitrary from the point of view of the model. In other words, any user can easily modify the lift profiles, or even substitute a function instead of a table. A dual-overhead cam (DOHC) including adjustable valve timings could also be easily modeled (using a separate phase shift angle), as could completely electronic valves. Therefore, the methods used in this model should be viewed as examples only and should not be construed as representing any program limitation.

Cylinder and Combustion Model

The fluid in each cylinder was modeled with a single FLUINT control volume (“tank”), whose volumetric rate or change (VDOT, in units of m^3/s) was adjusted sinusoidally.[‡] For example, for cylinder 3 the VDOT term is:

[†] The program is case-insensitive. Capitalization is used for emphasis or clarity within written descriptions.

[‡] A more realistic motion using a finite rod length (14.35cm) and crank arm size (45mm) is also possible. It would require VDOT as a function of crank angle either by equation or by table interpolation. It would also require a calculation of the initial volume of each piston at time zero. Since the focus of this model is on the air flows outside of the cylinder, this step was not taken.

```
pi*(displace/period)*sin(2*pi*(angle-phase3)/360)
```

The volume (VOL) of a tank cannot be directly prescribed as a function of time, only the current rate of change. The time step was therefore constrained to make sure there was no drift over time. Another option would be to add an error term into the VDOT calculation to keep it on track and allow larger time steps. However, other limits (such as resolution of acoustic waves) would soon be hit, so that method was not used.

Heat transfer to the engine is also very simplistic: a simple constant heat transfer coefficient in parallel with a simple constant effective emissivity,[§] with values chosen to make the total heat transfer roughly 1/3rd of available power. More complex correlations are available and could be used with a little more effort.

The use of a single control volume or "tank" means that perfect mixing is implicitly assumed within each cylinder. Other models were explored in which pressure waves were allowed to travel within the cylinder, but the transit times were very fast even at BDC (bottom dead center), corresponding to just a few degrees of crank angle, and so they did not affect valve response significantly and were neglected in later versions of the model.

The assumption of perfect mixing has more important repercussions for combustion modeling. However, the combustion approach was highly simplified in this case, since it was not the focus of the engine-level model. The various chemical reaction and species-tracking capabilities of FLUINT were not employed. For example, air was preserved as the working fluid throughout the model. Rather, combustion was greatly simplified as a singular event at zero TDC (top dead center): a step change in temperature and pressure. For cylinder#3, this netork-based logic is as follows (again, in FLOGIC 0):

```
atest = mod(angle-phase3,720.0)
if(abs(atest) .le. atol .and. tl#this .lt. 0.75*Tflame)then
       call tanktrho('air', #this, Tflame + (TL100-25),
                   (1.0+FAratio)*DL#this)
endif
```

This logic attempts to detect the correct ignition timing without firing more than once per cycle. It therefore includes some angular tolerance (*atol*) in case the current time step does not cause the crank angle to exactly coincide with the desired angle (which of course it will almost never do exactly).

When ignition is signaled, the TANKTRHO routine is used to raise the air mass within the cylinder instantly to 2200°C (*Tflame*), with a corresponding increase in pressure as needed to conserve mass. In

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§ The base coefficient UB is updated in network logic, linearizing the radiation term $(T_i^4-T_j^4)$ as $(T_i^2+T_j^2)*(T_i^2-T_j^2) = (T_i^2+T_j^2)* (T_i+T_j)*(T_i-T_j)$. Noting that QTIE = UB*AHT*(T_i-T_j) with UEDT=0.0, the final expression (converting temperatures to absolute units, and adding the Stefan-Boltzmann constant) is:

ub#this = Emiss*sbconsi* ((TL#lump-abszro)**2+(T#node-abszro)**2) * ((TL#lump-abszro)+(T#node-abszro))

Of course, the *Emiss* value and the use of AHT, treating the radiation exchange as occurring at the surface, is a gross approximation (of zero optical depth)and is justifiable only because this term has minimal effect on the results.

fact, mass is not actually preserved: it is augmented slightly by the input fuel to air mass ratio of 3% (*FAratio*) at the same time as ignition is assumed.[**] The central assumption is that the working fluid is still air before and after (including any products of combustion). That assumption is largely justified by the small amount of fuel burned.

The logical condition "tl#this .lt. 0.75*flame" helps prevent the ignition from happening more than once per cycle: if the temperature of this tank ("tl#this") is too warm (more than 75% of the flame temperature), it must be because it has already been ignited in this cycle.

The ignition timing could be advanced or retarded as needed with a trivial effort: introducing an angular shift in the "angle-phase3" term in the first line above. Also the combustion singularity could be replaced with a rapid but finite-rate heating.[††] More involved model changes (e.g., the introduction of multiple species, finite reaction rates or degrees of reaction completion) would be required to more realistically model the combustion process, if that had been the focus of the model.

Fluid (air) Properties

Because of the simplified treatment of combustion, a simple perfect-gas set of air properties was used (file "engine_air.inc"). The properties were actually only valid to 2000K, but were manually extrapolated to allow higher temperatures to occur over the very short time required to model ignition. This approximation was also justified because the focus of the model was on cooler temperatures in the intake and exhaust system.

A slightly better approach might be to use a hot/ionized air description, which is available as an output from the NASA CEA chemical equilibrium program.[‡‡] Even better would be to use CEA to generate a "hot products of combustion" variable-molecular weight fluid to represent the post-combustion gases as a single effective species, with one working fluid (air) entering the cylinder, and another (replacing it upon ignition. Such an approach would still have very fast solution times.

As was noted above, multiple species and specific reaction pathways could be modeled too, if that had been the focus of the model. Refer to other example problems at www.crtech.com for more information on modeling reacting flows such as http://www.crtech.com/reforming.html

Registers (Parametric Variables)

This section lists the three groups of registers used to define the model. Some of the registers in the last two groups (on the tabs "results" and "solution control") are described later.

Note that values that have been crossed out depend on "processor variables" whose value is unknown until execution begins (e.g., they only have meaning *during* a SINDA/FLUINT run).

[**] The last argument in TANKTRHO can be left at zero, meaning "no change in density").

[††] However, this would require the code to integrate through that event: an even smaller time step would be required during "combustion.")

[‡‡] Such FPROP blocks are available upon request, or can be generated by the user. See http://www.crtech.com/EQfluids.html for more details.

Global Registers

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	Name	Type	Expression	Comment	Value
1	speed	Float	4000	RPM of engine	4000.0
2	CR	Float	9	compression ratio	9.0
3	Tflame	Float	2200	temperature after combusion, deg C	2200.0
4	period	Float	60/speed	crank period, in seconds	0.015
5	angle	Float	360*timen/period	crank angle	~~37869.875~~
6	Tengine	Float	250	Engine wall temp (C)	250.0
7	FAratio	Float	0.03	F/A ratio (for mass appearing in cylinders)	0.03
8	crank	Float	mod(angle,720.0)-360.0	crank angle, -360 to 360	~~69.875847~~
9		Float			

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	Name	Type	Expression	Comment	Value
1	cyl_diam	Float	93.4e-3	cylinder diameter, m	0.0934
2	displace	Float	3.7/6/1000	displacement per cylinder, m3	0.00061666667
3	Vmin	Float	displace/(CR-1)	min volume per cylinder, m3	7.7083333E-05
4	cyl_stroke	Float	displace/cyl_xsection	stroke, m	0.090005042
5	cyl_xsection	Float	0.25*pi*cyl_diam^2	cross section, m2	0.006851468
6	e_valve_diam	Float	38e-3	exhaust valve diameter, m	0.038
7	i_valve_diam	Float	47e-3	intake valve diameter, m	0.047
8	e_valve_lift	Float	10.8e-3	exhaust valve lift, m	0.0108
9	i_valve_lift	Float	10.9e-3	intake valve lift, m	0.0109
10	HTC_air	Float	2500.0	W/m2-k air to block	2500.0
11	Emiss	Float	1.0	includes effective area multiplier (for volumetric)	1.0
12	phase1	Float	0	cylinder 1 starting position (TDC), degrees	0.0
13	phase5	Float	120+phase1		120.0
14	phase3	Float	120+phase5		240.0
15	phase6	Float	120+phase3		360.0
16	phase2	Float	120+phase6		480.0
17	phase4	Float	120+phase2		600.0
18	Vinit1	Float	Vmin + displace*(0.5+0.5*sin(2*pi*(270-phase1)/360	Initial volume, cylinder 1	7.7083333E-05
19	Vinit2	Float	Vmin + displace*(0.5+0.5*sin(2*pi*(270-phase2)/360		0.00053958333
20	Vinit3	Float	Vmin + displace*(0.5+0.5*sin(2*pi*(270-phase3)/360		0.00053958333
21	Vinit4	Float	Vmin + displace*(0.5+0.5*sin(2*pi*(270-phase4)/360		0.00053958333
22	Vinit5	Float	Vmin + displace*(0.5+0.5*sin(2*pi*(270-phase5)/360		0.00053958333
23	Vinit6	Float	Vmin + displace*(0.5+0.5*sin(2*pi*(270-phase6)/360		7.7083333E-05
24	Tinit	Float	200 + speed/15	Exhaust system temperature, deg C	466.66667
25	Avalve_in	Float	pi*i_valve_diam*i_valve_lift	Flow area near intake valve, m2	0.0016094379
26	Avalve_ex	Float	pi*e_valve_diam*e_valve_lift	Flow area near exhaust valve, m2	0.0012893096
27		Float			

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	Name	Type	Expression	Comment	Value
1	Dintake	Float	4*0.0254	diameter of intake duct, m	0.1016
2	Lintake	Float	3*0.3048	length of intake duct	0.9144
3	Dexhaust_man	Float	4*0.0254	not used here	0.1016
4	Lexhaust_man	Float	1*0.3048	used as transition section length, m	0.3048
5	Dexhaust_pipe	Float	2.25*0.0254	exhaust pipe diameter	0.05715
6	Lexhaust_pipe	Float	10*0.3048	exhaust pipe length	3.048
7	Kexhaust_pipe	Float	3	K-factor of exhaust pipe	3.0
8	Kmuffler	Float	4	K-factor of muffler (and catalytic converter)	4.0
9	Lin_runner	Float	500.e-3	intake runner length, m	0.5
10	Din_runner	Float	45.e-3	intake runner diameter	0.045
11	Lex_runner	Float	14*0.0254	exhaust runner length	0.3556
12	Dex_runner	Float	1.625*0.0254	exhaust runner diameter	0.041275
13	Kair_filter	Float	20	K-factor of air filter (based on large-diam intake!	20.0
14		Float			

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Case Management

Three analytic cases were considered. Each case corresponds to an engine speed: 2000, 4000, and 6000 rpm. The rotational rate was set by a case-specific register override of *speed* and *numcycle*. For example, for the 2000 rpm case:

Registers - Case 2000rpm

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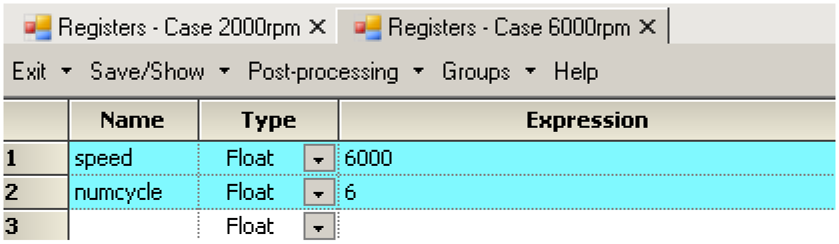
	Name	Type	Expression
1	speed	Float	2000
2	numcycle	Float	3
3		Float	

The register *numcycle* is the number of camshaft rotations (with 2**numcycle* being the number of crankshaft rotations) to integrate starting from the guessed initial conditions. Because the engine is modeled using moving pistons and valves, no single effective/average flow rate and heating can be calculated. This means the model cannot be executed in a steady-state mode, and that in turn means that initial conditions can only be intelligent guesses (usually from prior runs). And *that* means that several cycles are required to "wash out" the effect of these initial conditions, resulting in a repeatable cycle that is independent of the initial temperatures and pressures and flow rates.

With reasonable guesses at the initial conditions, and having neglected heat transfer (except to the cylinder wall), it takes 3 camshaft rotations (*numcycle*) to get repeatable response at 2000 rpm. It takes more rotations at higher RPM: enough absolute time needs to elapse to flush out the effects of the initially guessed temperature in the manifolds. Therefore, *numcycle*=6 at 6000 rpm:



Registers - Case 2000rpm | Registers - Case 6000rpm

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	Name	Type	Expression
1	speed	Float	6000
2	numcycle	Float	6
3		Float	

If volumes increase, more cycles may be required to reach a quasi-steady waveform for key variables. Similarly, if the initial guesses are farther off the mark, more cycles will be required. The register *Tinit* contains a reasonable guess at the exhaust system temperature of “200+speed/15” (in degrees C) based on prior runs. The sole purpose of this register is to keep the run times small by limiting the *numcycle* necessary.

The need to run several cycles in order to achieve convergence about a final cycle also requires some additional logic or controls: it is best if default plots only showed the last cycle and not initial cycles.[§§] Therefore, output should be disabled for all but the last cycle, as controlled in this model via the register *just_last*.

A register named *crank*, containing the crank angle for each cam cycle (from -360 to 360 degrees) is available for use as the X axis instead of time.

Notes on Run-time messages

When execution starts, a few warnings are produced by the code.

One type of warning is as follows:

```
  *** WARNING *** FK WAS 0 FOR TUBE WITH UNEQUAL AFI,AFJ.  ONE-TIME (CONSTANT) SUDDEN EXPANSION
FK CALCULATED.  MODEL: AIR, TUBE:        28
```

In many of the ducts (such as the valve-ends of the runners, and the exhaust 6-1 transition section), the flow areas are not constant from inlet to outlet. When this happens, secondary losses should be specified by the user to “explain” to the program the *manner* of the transition: smooth, sudden, etc. If the user has not supplied any K-factor for irrecoverable losses, the problem calculates them itself assuming sudden expansion or contraction in the positive flow direction. This default system is allowed in this model, despite being a likely over-estimate of the losses, because the flow area changes are not significant and more realistic losses could only be estimated with a more specific definition of the geometry of each passage.

Another type of warning appears as well:

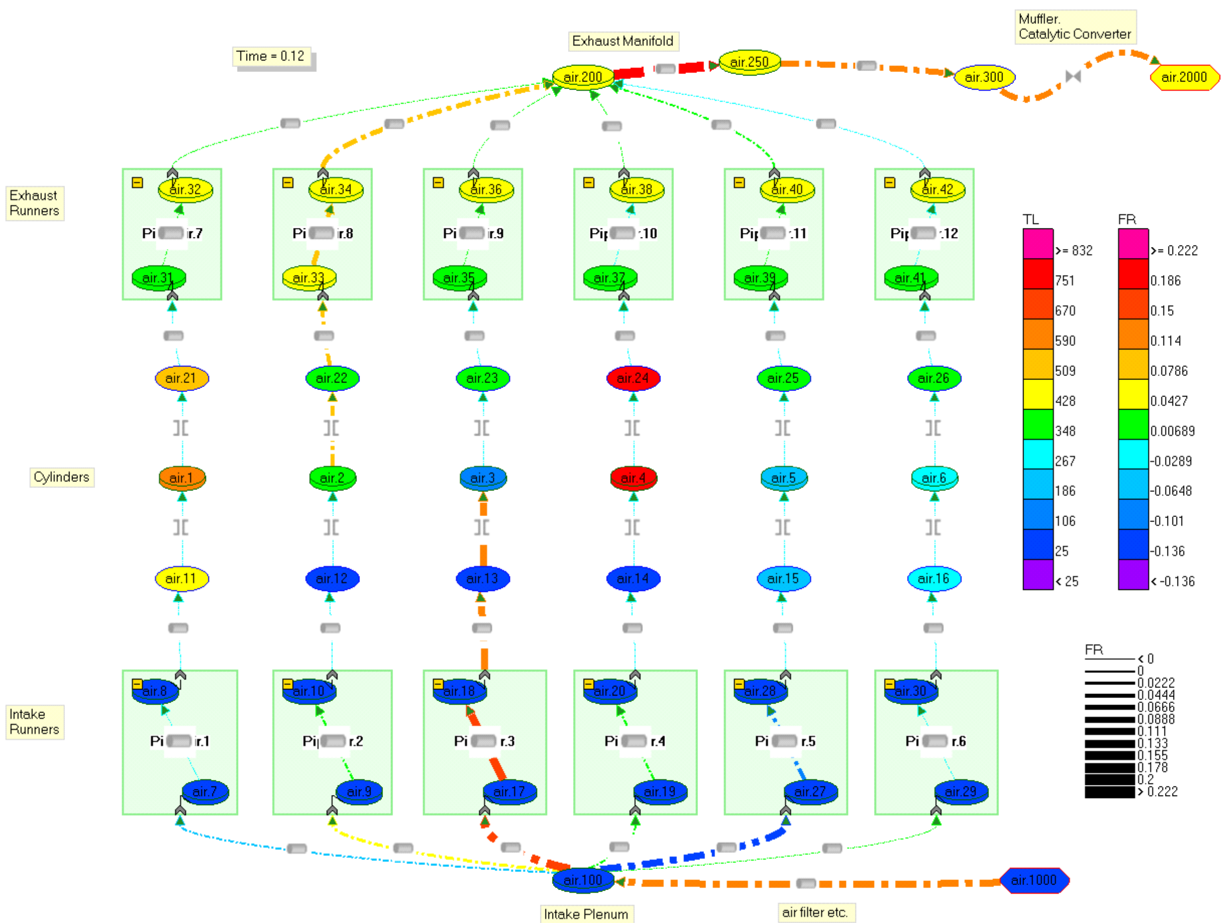
```
  *** WARNING *** UPF RUNAWAY DETECTED. INCREASE RESOLUTION AND CHECK PRESSURE DROP IN SIMILAR
PATHS. UPF SET TO 1.0 FOR  MODEL: AIR, TUBE:        28
```

[§§] There are two instances where this isn’t true, and where all data needs to be plotted by setting *just_last*=0. The first includes debugging purposes: making sure the model is responding well in case a problem occurs before the last cycle has been started. The second instance is verifying that enough cycles have been performed to get a valid profile in the event that key parameters are modified: that *numcycle* is sufficiently large.

By default, the program uses an average of upstream and downstream densities in each path when calculating frictional pressure drops: UPF=0.5 by default. For severe pressure drops, this can be unstable and either more spatial resolution (more lumps) is required, or a larger UPF should be used to discount the higher velocities at the exit of each flow path. While increased runner resolution is probably a good idea, frictional losses are actually not very high in the runners of this engine, so just letting the program override a numerically dangerous (but physically inconsequential) UPF with 1.0 is allowed.

Results

Below is a postprocessed depiction of the network corresponding to the end of the 4000 rpm run, as cylinder 1 is back to top dead center (TDC) and ready to fire again.

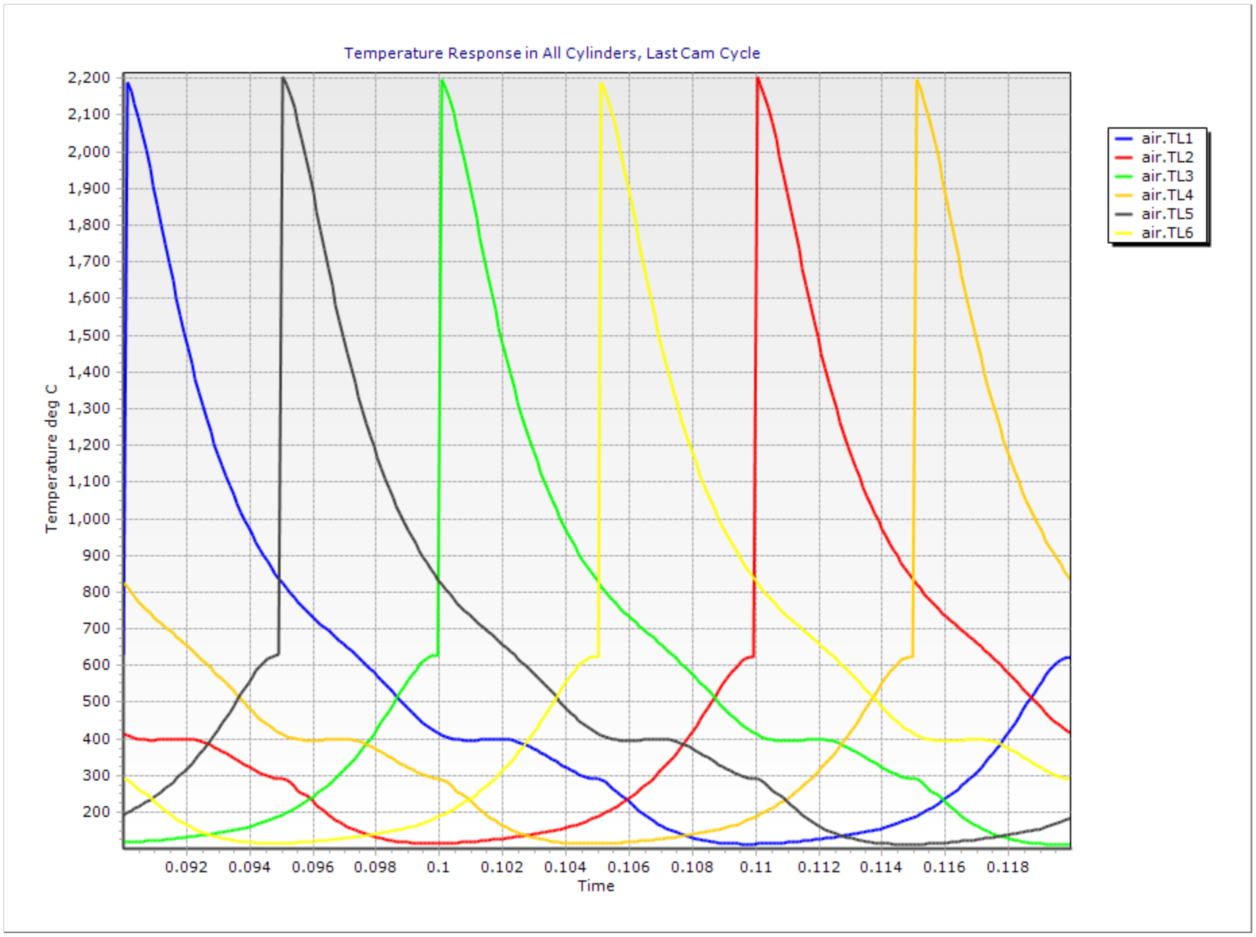


In the above diagram, the lumps are colored by temperature. The flow paths are colored by mass flow rate, allowing negative (reverse flow) values to be represented in blue shades. The paths have simultaneously been thickened by the absolute value of mass flow rate.

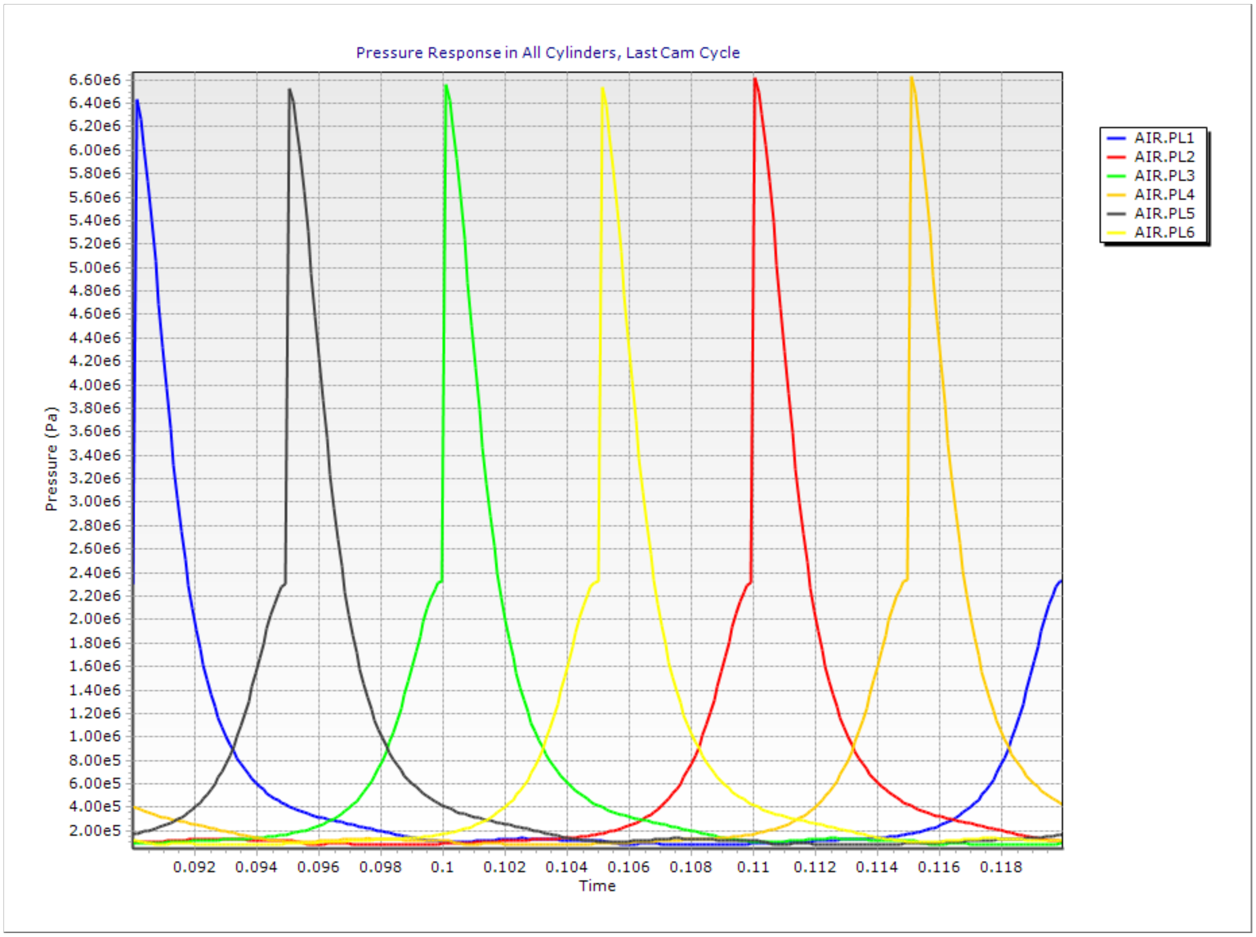
Using this information, it can be seen that cylinder 1 is warm from compression, cylinder 3 has recently discharged and exhausted (though the valves are currently closed). The exhaust valve for cylinder 2 is still open, and the intake valves for cylinders 3 and 5 are both open … except that the flow is *backwards* in the intake valve of cylinder 5 (see later discussion on volumetric efficiency and valve timing).

Cylinders

The temperature response for all cylinders is shown below for the last camshaft rotation at 4000 rpm. Temperatures are about 600°C before ignition, at which time they jump (thanks to the TANKTRHO invocation) to the *Tflame* value of 2200°C. The subsequent drop is due to expansion during the power stroke. The temperatures continue to drop during the exhaust stroke, with the cooler air from the intake manifold lowering the temperature to 100°C before the next compression stroke begins. These temperature profiles are representative only, given that the heat transfer to the cylinder was highly approximated: an average film coefficient was selected strictly as needed to achieve about 1/3rd of the thermal energy flowing into the engine wall. More complex correlations can be added by an end user.



The corresponding pressure profiles in each cylinder are presented below at 4000 rpm. Peak pressures (which are exaggerated given the instantaneous ignition assumption used) reach 64 bar.



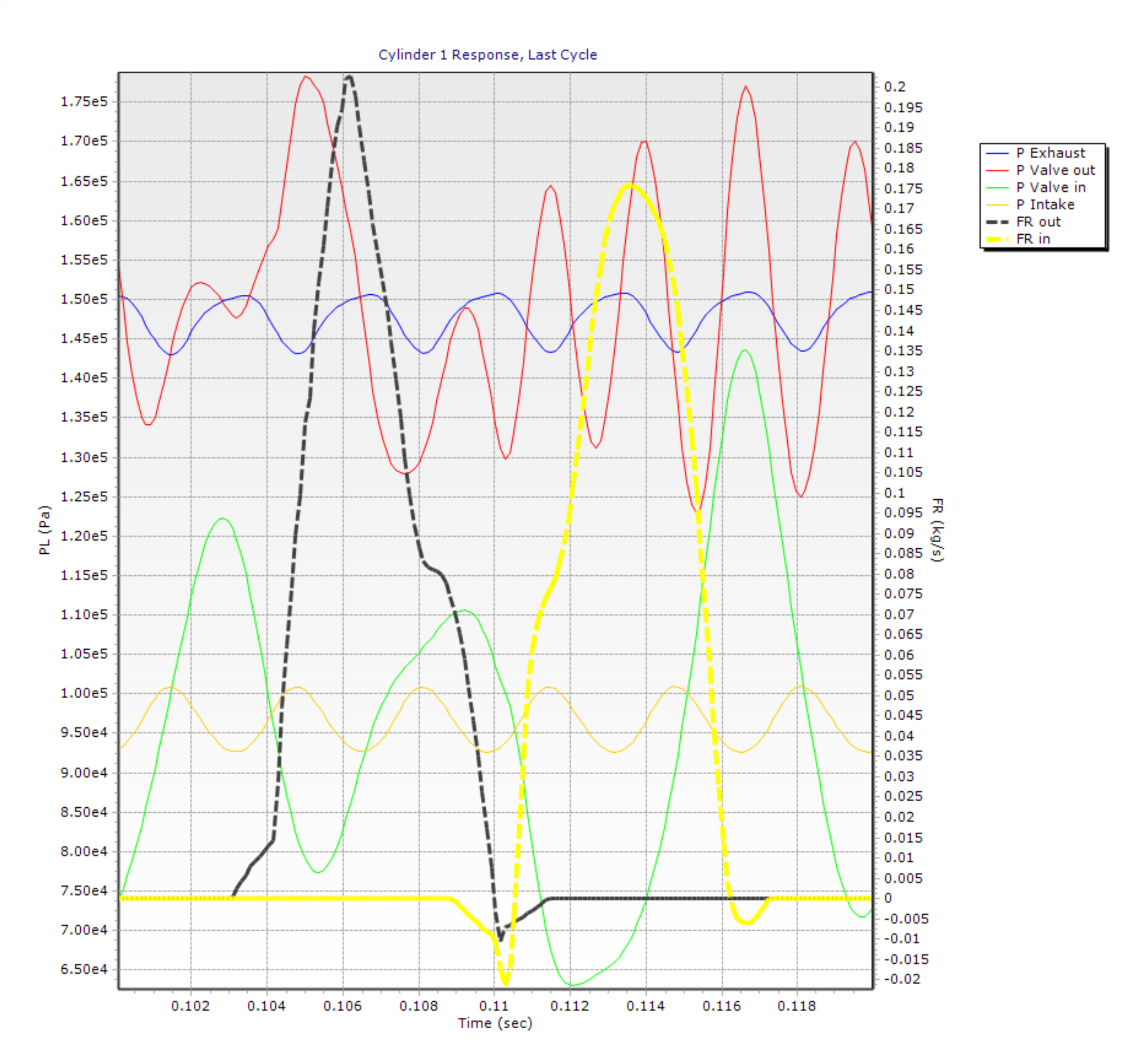
Runners and Valves

The following plot shows pressure and mass flow rate responses for cylinder 1 at the last camshaft rotation for the 6000 rpm case.

The thin orange line (“P Intake”) is the pressure in the intake manifold, which can be seen (on the left axis) to be oscillating just below atmospheric pressure. The thin blue line (“P Exhaust”) is the exhaust manifold static pressure, which is oscillating at almost 1.5 atmospheres.

The pressures near the valves have much greater amplitude and much less regular shapes, as would be expected: this is the effect of the pressure lags in the runners and the valves opening and closing. The thin green line (“P Valve in”) is the pressure on the intake runner side of the intake valve, and the thin red line (“P Valve out”) is the pressure on the exhaust runner side of the exhaust valve.

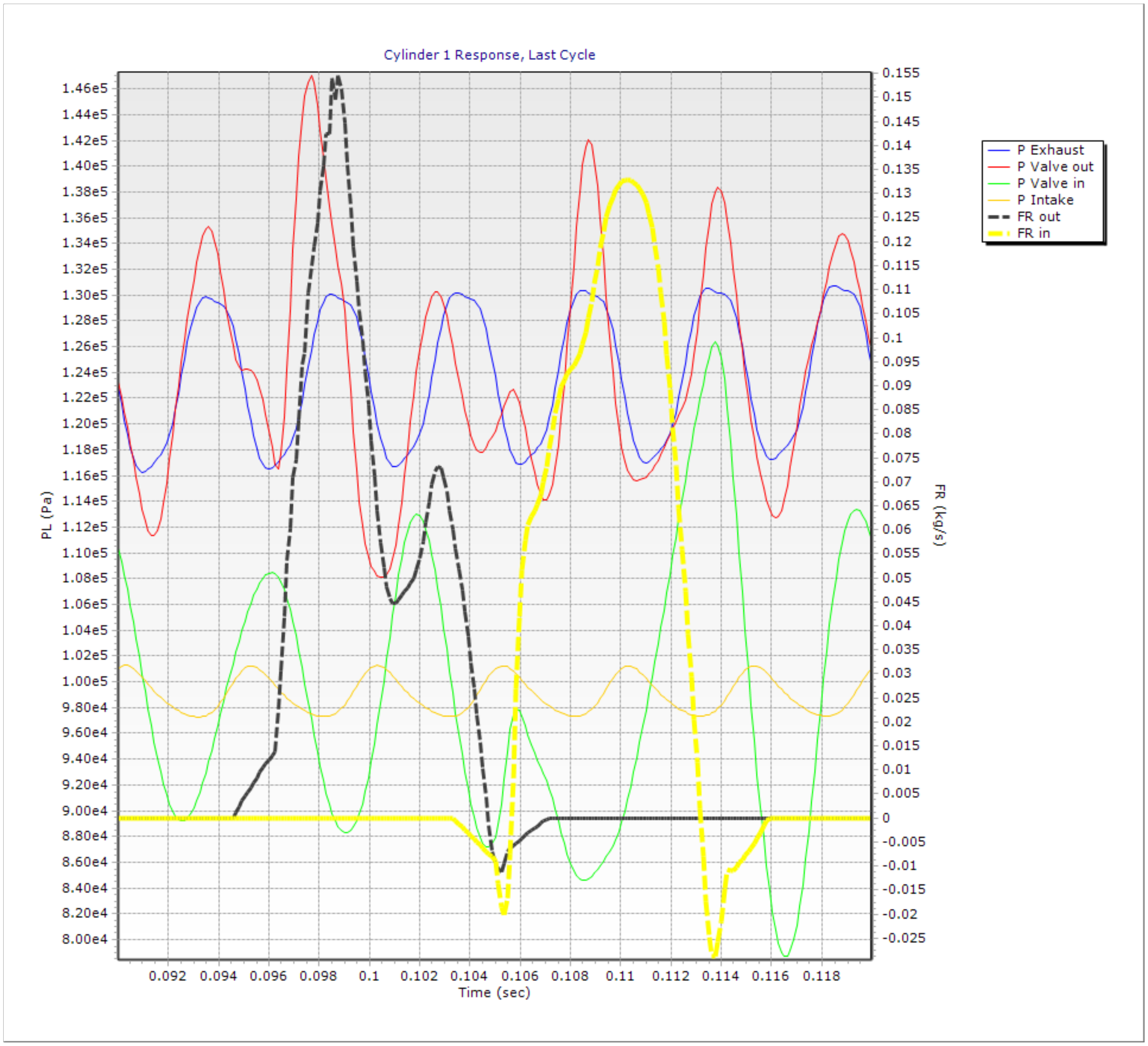
The mass flow rate out of the exhaust valve is plotted as a thick dashed black line (“FR out”), and the mass flow rate into the piston through the intake valve is a thick dashed yellow line (“FR in”). Some degree of reversed flow is evident before each valve closes, and it is also evident as the intake valve begins to open.



6000 RPM, Flows and Pressures

This response is well tuned on the intake side: the increase in pressure just as the valve opens and again as it closes keeps flow reversals minimal. However, the decrease in pressure at maximum intake valve lift is suboptimal. The major contributor to less-than-optimal volumetric efficiency (VE) in the intake is the fact that the intake valve stays open so long.

For the exhaust system, however, the resonant frequency of the runner appears to be too fast to assist much, though a dip in pressure does occur shortly after peak flow has occurred, so a second peak in mass flow rate is starting to form as the valve is closing.

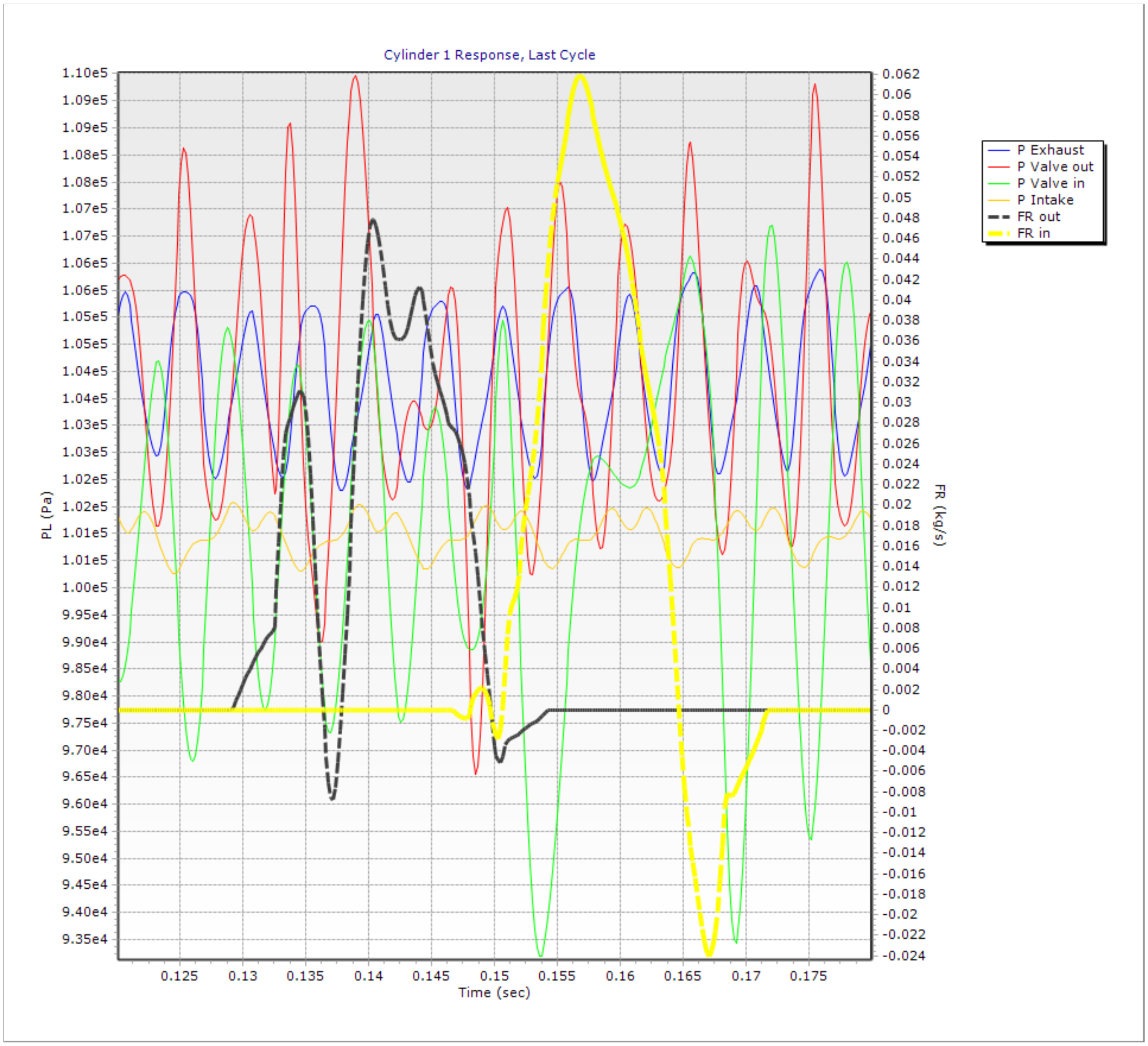


4000 RPM, Flows and Pressures

The above plot shows the same responses at 4000 rpm. Notice the time scale has been enlarged 50% over the prior plot since it a single valve cycle takes longer to complete at the slower engine speed.

The exhaust manifold is now resonating with the runner, and a strong second peak in exhaust flow rate forms as a result. The intake responds continues to help stop backflow in the intake despite the valve staying open so long after BDC, achieving (as will be shown later) slightly higher intake VE than did the 6000 rpm case.

The plot below shows the response at 2000 rpm. Clearly, the ducting has not been tuned for such low speeds, as disruption in the exhaust flow and significant reversed flow in the intake valve are now both evident. (This plot is difficult to read since the difference in pressures between intake and exhaust is now less significant, so the waveforms too often overlap.)



2000 RPM, Flows and Pressures

By the Numbers

In addition to plots, various performance metrics were tracked or calculated during the execution such that they could be reported numerically at the end of the run.

Within each intake valve, for example, the net volume ingested by the cylinder was summed in FLOGIC 2 at the end of each time step:

```
vol_in1 = vol_in1 + fr#this/dl1000*dtimuf
```

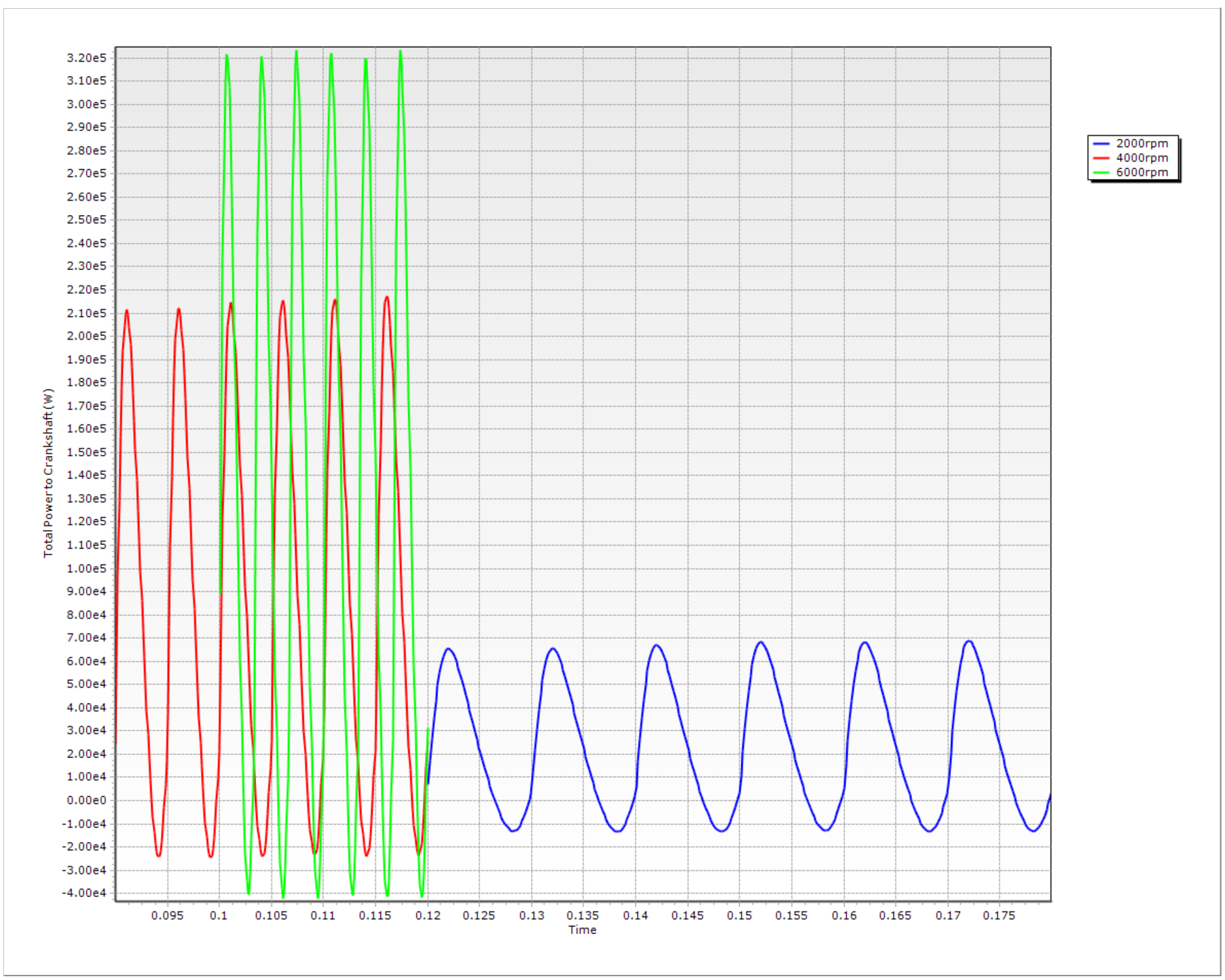
“FR#THIS” means the mass flow rate (kg/s) of this valve (positive into the cylinder for intake valves), and “DTIMUF” is the time step last taken, such that the product of these two is the mass (in kg) that flowed into the cylinder during that last time step. The total for each cam rotation is then compared with half the displacement to calculate the volumetric efficiency (VE). If the flow rate is negative for any time step, mass is subtracted and the VE was therefore degraded. (Reversed flow was the main cause of suboptimal VE in this design.)

Actually, the above estimate is the volume based on the current/local mass flow rate, but it is divided by the atmospheric density ("dl1000", the density of a "plenum" or boundary condition, rather than the local upstream lump's density ("dl#up"), in order to report VE using customary methods.

The power into the crankshaft was calculated for each cylinder, and then summed into a total value:

```
power1 = (pl#this-pl1000)*vdot#this
work1 = work1 + power1*dtimuf
```

In the above logic, "pl#this-pl1000" is the pressure difference across the piston (in Pa) and "vdot#this" is the growth rate of the cylinder (m^3/s), yielding *power1* in units of Watts. This power is summed with that of other cylinders to yield *powerT*, the total power. This power is shown below for each of the three engine speeds (plotted against absolute time rather than crank angle so that frequencies are visible).



Since that is the instantaneous power, the total work is summed (*work1*), again using the last time step DTIMUF. This integrated work term yields the time-averaged power (*powerA*) when divided by the camshaft's period.

Similarly, the total energy that flows from the cylinder fluid into the wall is tracked over each camshaft rotation, such that the fraction of thermal energy leaving the cylinder (and posing a load on the coolant system) can be calculated.

These calculations yield the following results:

	2000 rpm	**4000 rpm**	**6000 rpm**
Volumetric Efficiency	61.1%	85.9%	81.9%
Thermal Efficiency	28.6%	39.6%	41.3%
Coolant Load	56.2%	31.7%	24.5%
Power to Crank	22.6kW	84.4kW	127kW

Efficiencies peak for the design point of 4000 rpm, and are still reasonably high at 6000 rpm, but they drop quickly as the speed drops to 2000 rpm. As a result, $1/4^{th}$ as much power is produced at 2000 rpm than was produced at twice the engine speed. But power scales more linearly with speed between 4000 and 6000 rpm.

Using a constant engine temperature (*Tengine*=250°C) and constant heat transfer coefficient … constant not just over time but even as engine speed changes …. means that the above trends cannot be trusted. However, they illustrate the type of information that *could* be gleaned from a more realistic, engine-specific model.