Blax/Beech Consulting Engineers

Date: September 7, 2002

To: Richard K. Herz

From: Fred Flintstone

Subject: Engine Simulation Project, Phase 1

This memo reports the results of Phase 1 of the design of a four-stroke, internal combustion engine to be used in a new ultra-light, hybrid gas-electric automobile. The goal of Phase 1 is a simulation and initial design of a one-cylinder version of the engine. This goal has been achieved, and this memo briefly describes the simulation and presents initial results and initial design studies.

The initial engine design is a 1 cylinder engine with a 10:1 compression ratio and a displacement of 0.40 liters. Conditions for Case 1 are listed in detail in Appendix A of this memo.

Under the assumptions listed below, the simulation of Case 1 predicts a power output of 3.33 kW (4.47 horsepower) at 2000 rpm, with 32.5 % of fuel energy going to the crankshaft. When the simulation is improved in Phase 2 of the project, we expect the design of a single- or multiple-cylinder engine can be optimized for use in the new hybrid automobile.

The primary assumptions made in Phase 1 were that (a) frictional losses are neglected, (b) valves are either full open or full closed and valve overlap is not allowed, (c) constant fuel burn rate during the fuel combustion period, (d) heat loss to the coolant is modeled with a constant, mean cylinder wall temperature, and (e) the cylinder gas has the physical properties of air and obeys the ideal gas law during all strokes.

The simulation was programmed in Matlab. In addition to the main script file, there are seven function files and an input parameter file. The Matlab files that contain the simulation have been sent to you today via email.

Several simulation cases were run in order to validate the simulation. Figure 1 below was run with the conditions of Case 1 with the exceptions that there is no heat transfer to the cylinder wall (adiabatic operation), no fuel, all valves closed at all times. That is, the engine acts as a closed, adiabatic gas compressor and expander.

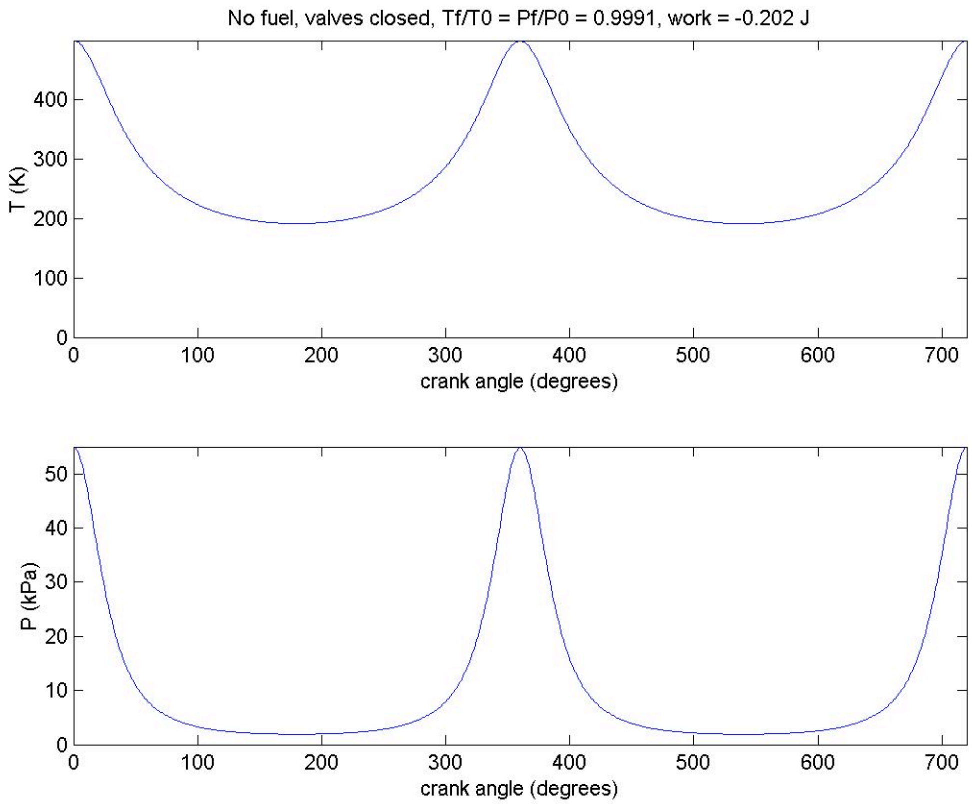

Figure 1

The results show that the simulation does a good job at returning the final temperature (Tf) and pressure (Pf) to the initial conditions (T0, P0), and the energy balance is close to returning the desired zero net work result.

Figure 2 below shows another validation run. The conditions of Case 1 hold with the exception that no fuel is fed to the engine. That is, valves open and close, and there is heat loss to the cylinder wall, but there is no power generated by fuel combustion. Per two-revolution cycle, work done by the engine on the gas is 44.8 J, net energy lost to coolant is 9.8 J, and net energy lost to exhaust is 32.5 J. The fact that the energy balance equals 2.5 J rather than the desired zero shows the influence of errors inherent in the numerical approximations currently used in the simulation.

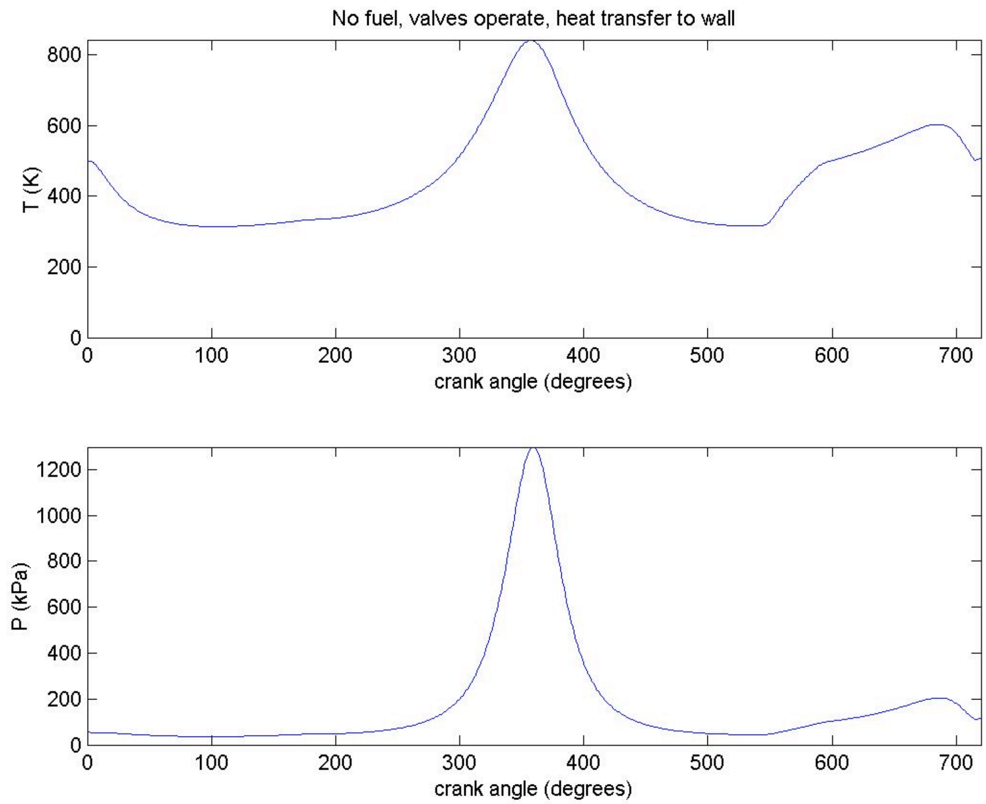

Figure 2

Figure 3 shows the fully operational Case 1. The peak temperature (T) is 3359 K, and the peak pressure (P) is 3071 kPa. Per two-revolution cycle, the gas does 200 J net work at the crankshaft, net energy lost to coolant is 294 J, and net energy lost to exhaust is 84 J.

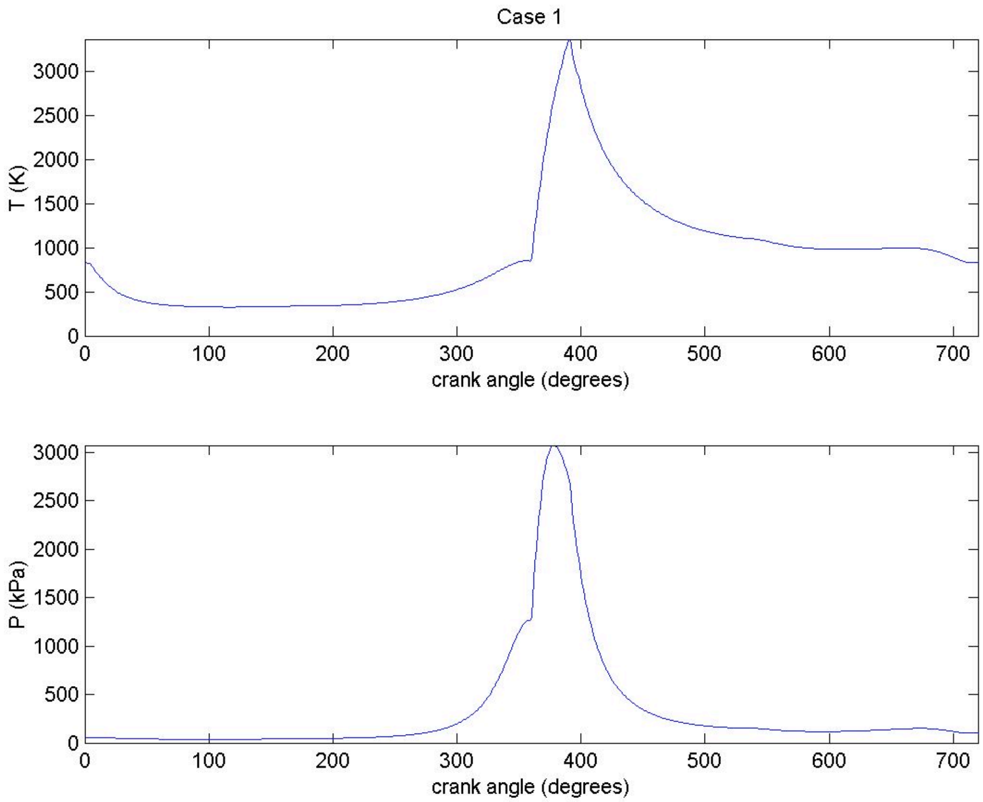

Figure 3

P-V and work rate vs. crank angle plots for Case 1 are shown in Appendix B of this memo.

Even though this Phase 1 simulation program involves many simplifying assumptions, the results shown in Figure 3 compare favorably with results for a similar engine produced by the commercial simulator EngineSim Otto by Androma. With improvements planned for Phase 2, our proprietary simulator will surpass commercially available simulators in features and accuracy.

The Phase 1 simulator is being used for initial design studies. First, the compression ratio was varied by changing the piston-cylinder head clearance at top dead center and keeping the displacement and other conditions constant.

Compression ratio	Power (kW)	Power (HP)	Efficiency
9:1	3.27	4.38	31.8 %
10:1 (Case 1)	3.33	4.47	32.5 %
11:1	3.37	4.51	32.8 %

These results show the advantage of increased compression ratio. Offsetting this advantage is the requirement for higher octane fuel at higher compression ratio.

In the second design study, the engine speed was varied.

Engine speed (rpm)	Power (kW)	Power (HP)	Efficiency
1500	2.60	3.49	30 %
2000 (Case 1)	3.33	4.47	32 %
2500	3.38	4.54	31 %

These results show the expected increase in power with engine speed. The engine efficiency appears to remain constant within the accuracy of the simulation.

In Phase 2, the simulation will be improved by (a) introducing a description of frictional losses, (b) describing the valve cam shape and allowing valve overlap, (c), introducing a more accurate approximation of the fuel burn rate, and (d) describing the effect of fuel and combustion products on gas properties.

The programs will be improved in Phase 2 by adding a graphical user interface (GUI) using Matlab's GUI tools.

Work on Phase 2 will begin as soon as I return from my well-deserved vacation to Jellystone National Park.

APPENDIX A - CASE 1 CONDITIONS

```
bore = 80 * 1e-3; % (m), Bore
gap = 8 * 1e-3; % (m), Clearance between piston and cylinder top at TDC
conrod = 160 * 1e-3; % (m), Connecting rod length
crankrad = 40 * 1e-3; % (m), Crank radius
RPM = 2000; % Engine speed (revolutions per minute)

Pint = 55 * 1e3; % Intake manifold pressure (Pa)
Tint = 300; % Intake manifold temperature (K)
Pexh = 100 * 1e3; % Exhaust manifold pressure (Pa)
Texh = 830; % Exhaust manifold temperature (K)

diamIV = 36 * 1e-3; % (m), diameter of intake valve, or equiv diam of 1 valve
in 2 intake valve engine
diamEV = 30 * 1e-3; % (m), diameter of exhaust valve, or equiv diam of 1
valve in 2 exh valve engine
angIVopen = 5; % (deg), angle at which intake valve opensangIVclose = 175; %
(deg), angle at which intake valve closesangEVopen = 545; % (deg), angle at
which exhaust valve opensangEVclose = 715; % (deg), angle at which exhaust
valve closes
CdIV = 0.13; % (dimensionless), discharge coefficient of intake valve
CdEV = 0.14; % (dimensionless), discharge coefficient of exhaust valve

fuelHV = 44e6; % (J/kg), Energy content of fuel (heat of combustion)
AFR = 14.5; % A/F of input air-fuel mixture = stoichiometric = 14.5
angBurnOn = 360; % (deg), Angle at which fuel combustion starts
angBurnOff = 390; % (deg), Angle at which fuel combustion ends
```

```
% heat transfer to wall assumed proportional to gas pressure
Hwall = 75; % (J/(m2 s K)), Wall-gas heat transfer coeffic at 100 kPa
Twall = 500; % (K), Average T of cylinder wall

R = 8.3144; % ideal gas constant, J/(mol K), also (Pa m3)/(mol K)
Cp = (7/2)*R; % heat capacity press, ideal diatomic gas, J/(mol K)
Cv = (5/2)*R; % heat capac const vol, ideal diatomic gas, J/(mol K)
k = Cp/Cv; % heat capacity ratio
MW = 0.029; % mean molecular weight of air, kg/mol

% Initial Conditions (IC) at TDC, start of intake stroke.
T0 = Texh; % (K) initial T, at end of exhaust stroke
P0 = Pint; % (Pa) initial P, cyl at end of exh stroke
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APPENDIX B - ADDITIONAL CASE 1 RESULTS

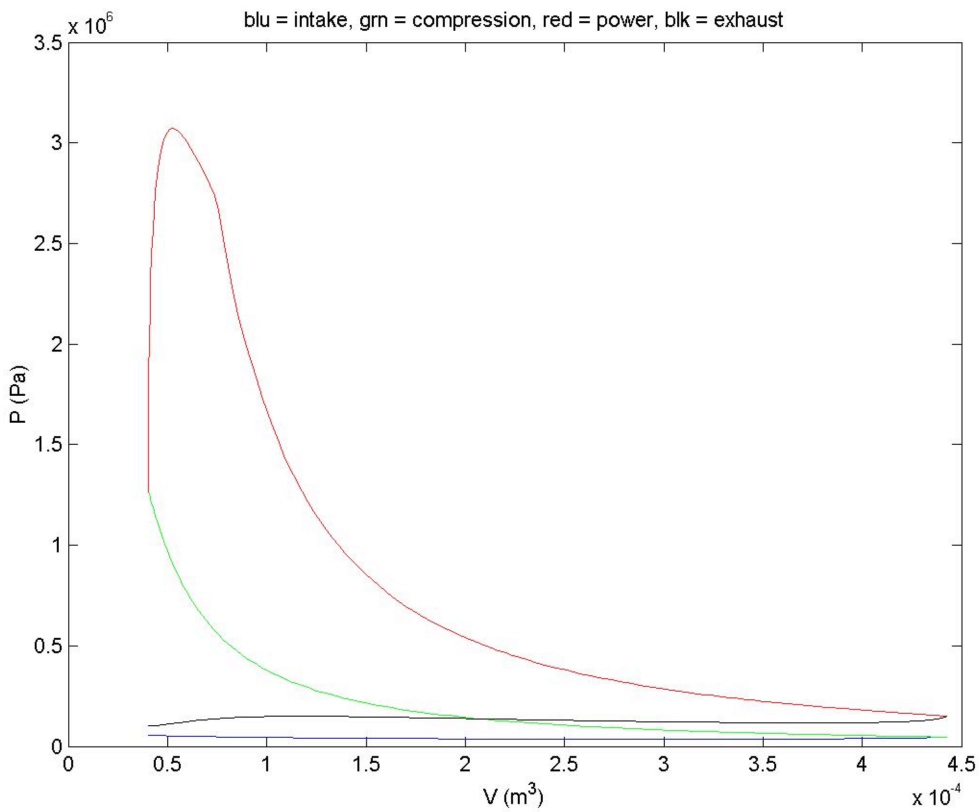

Figure B-1

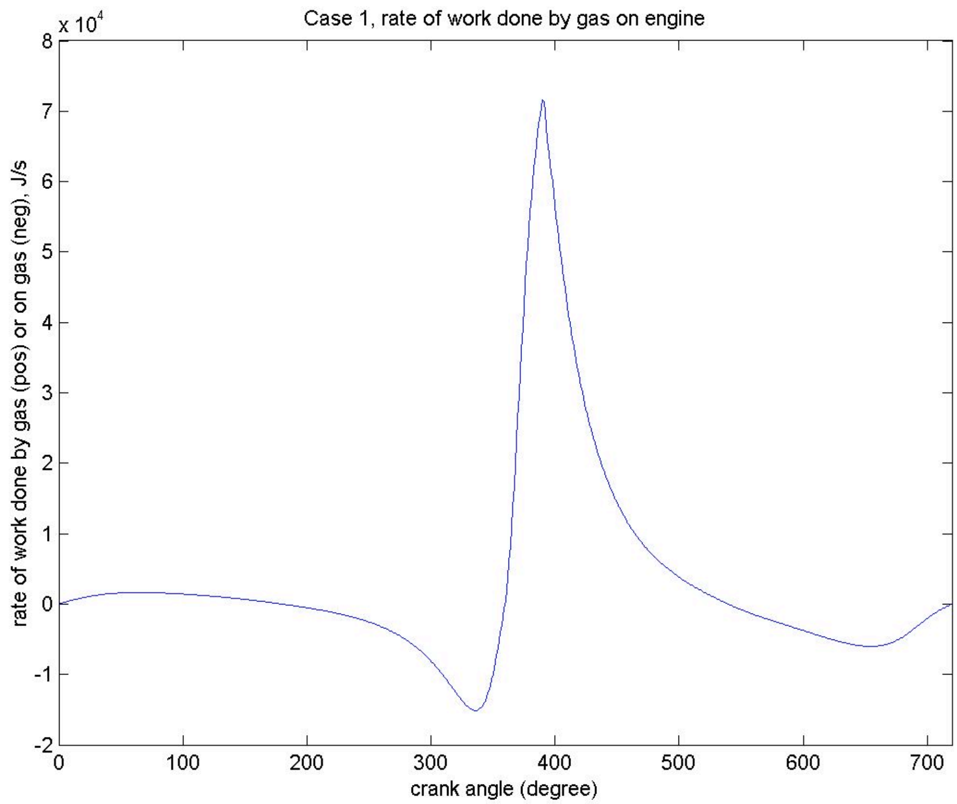
Case 1, rate of work done by gas on engine
x 10^4
rate of work done by gas (pos) or on gas (neg), J/s
crank angle (degree)
8
7
6
5
4
3
2
1
0
-1
-2
0
100
200
300
400
500
600
700

Figure B-2