Computer Simulated Engine Performance

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Abstract

This model is a computer simulation which determines the performance of a four stroke internal combustion (IC) engine. The modeling of this process begins with the simulation of one cylinder of the four stroke IC engine which is assumed to have an ideal pressure-volume (p-V) relationship allowing for computation of peak performance. Once the ideal cylinder is modeled, factors which compensate for less than ideal p-V relationships are injected into the simulation to allow for computation of the performance figures across the entire operating range. The single cylinder model is then expanded to simulate the interaction of multiple cylinders at once and compute their combined effect giving total output numbers for the engine as a whole.

Performance figures computed include torque and horsepower curves for an engine's entire operating range. Additionally, statistics are also available regarding the pressure and torque within a single cylinder as a function of the crank angle at a particular RPM. Finally, the combined efforts of cylinders functioning together and their effect on torque at a particular RPM are presented. This model has been used to simulate the performance of the Audi 1.8 liter I-4, the Subaru 3.0 liter H-6/Boxer, the Ford 347 Stroker, and a theoretical Chevy 350 buildup.

Related Work

1. Engine Dynamometer

An engine dynamometer is a machine which hooks up directly to an internal combustion engine and can determine the performance figures for that engine, including torque and horsepower. This machine is hooked up to a computer with specialized software designed to extract information from the dynamometer. This machine allows users to determine what the performance of an internal combustion engine is over its full RPM range.

2. Virtual Engine Dyno

Virtual Engine Dyno is software developed by Challenger Engine Software, LLC for the purpose of allowing its users to virtually build up an engine on their own computer. This allows users to test out different engine setups before they go out and perform the labor of putting new parts on their engine. The software calculates many different engine specifications and performance figures. Basic versions focus on common engine setups while advanced versions allow users to build an engine from the ground up.

3. Dynamic Analysis of the Internal Combustion Engine

This was a project given to students in MEAM 211, taught by Professor Vijay Kumar, right here at SEAS during the spring of 2005. The goal of the assignment was to have students model a dynamic system of equations, explain what was going in the model and then propose ways of improving the engines they were modeling based on their findings.

The above related works represent good examples of what work has already been done in the fields relating to our project. This project draws benefits from studying the works above and those related to them. At the same time, this model is somewhat of a combination and extension of all of the above works, making it different from all three. Performance figures are provided like an engine dynamometer and the Virtual Engine Dyno software, but the model also offers a more detailed analysis of the performance inside each cylinder like the Dynamic Analysis of the Internal Combustion Engine presents. Furthermore, a user interface is provided on top of the simulation which allows users to easily compare modeled data from various engines in a side-by-side fashion. The related works either focus on just high level views of engine performance across an engine's entire operating range or on the internal workings of the engine at one particular RPM. This simulation provides a picture of engine performance from the moments and pressures within each cylinder all the way out to the torque and horsepower curves measured all the way out at the flywheel.

Technical Approach

We began by trying to break the four stroke internal combustion (IC) engine down into sub-sections on which we would base our simulation. We ended up breaking the engine down into 3 main sections: the single cylinder, the intake and fuel delivery system and the crankshaft.

Our starting point for modeling the single cylinder and the crankshaft was Dr. Kumar's Dynamic Analysis of the Internal Combustion Engine. Dr. Kumar's MEAM 211 assignment provided the derivation of the equations necessary to model the single cylinder and crankshaft at a *constant* RPM.

First, time is abstracted away by basing the entire simulation as a function of the angle of the crankshaft (θ):

N = revolutions per minute (RPM)

$$\theta(t) = \frac{N}{60} 2\pi * t$$

Since the model is for a constant RPM the derivative of the angle of the crankshaft (θ') is:

$$\theta' = \frac{N}{60} 2\pi$$

Next, we needed to have various expressions of the angle and the derivatives of the angle between the connecting rod and the piston. We were able to base the derivation of this angle (Φ) on the angle of the crankshaft (θ), the length of the connecting rod (l), and the radius of the crankshaft (r):

l = length of the connecting rod
r = radius of the crank (= stroke/2)

$$\phi = \sin^{-1}(\frac{r}{l}\sin\theta)$$

$$\phi' = \frac{r\cos\theta}{l\cos\phi}\theta'$$

$$\phi'' = -\frac{-r\sin\theta\theta'^2 + l\sin\phi\phi'^2}{l\cos\phi}$$

Once we have the angle of the crankshaft (θ), the angle between the piston and the connecting rod (Φ) and the derivatives of these angles we can express the position (x), velocity (x') and acceleration (x") of the piston:

$$x = r\cos\theta + l\cos\phi$$

$$x' = -r\sin\theta\theta' - l\sin\phi\phi'$$

$$x'' = -r\cos\theta\theta'^2 - l\cos\phi\phi'^2 - l\sin\phi\phi''$$

Before being able to calculate the contribution a single cylinder makes to the turning moment of the crankshaft, we have to calculate the pressure (P) and volume (V) within the cylinder throughout the 720 degrees of rotation the crankshaft goes through in four strokes. The cylinder volume (V) is equal to the sum of the clearance volume (vc) in the cylinder and the swept volume (V_s) which varies according to the position of the piston (x). In this way, the volume within the cylinder becomes a function of the position of the piston:

D = bore of the cylinder
vc = clearance volume

$$A = \frac{\pi D^2}{4}$$

$$V_s = A(r + l - x)$$

$$V = vc + V_s$$

We can then calculate the pressure within the cylinder from the volume in the cylinder (V), atmospheric conditions, an expansion coefficient (k), and what part of the four stroke cycle the cylinder is in. We base our representation of the four stroke cycle on the Otto cycle. The relationship between volume and pressure under ideal conditions within a single cylinder according to the Otto cycle is shown in Figure 1.

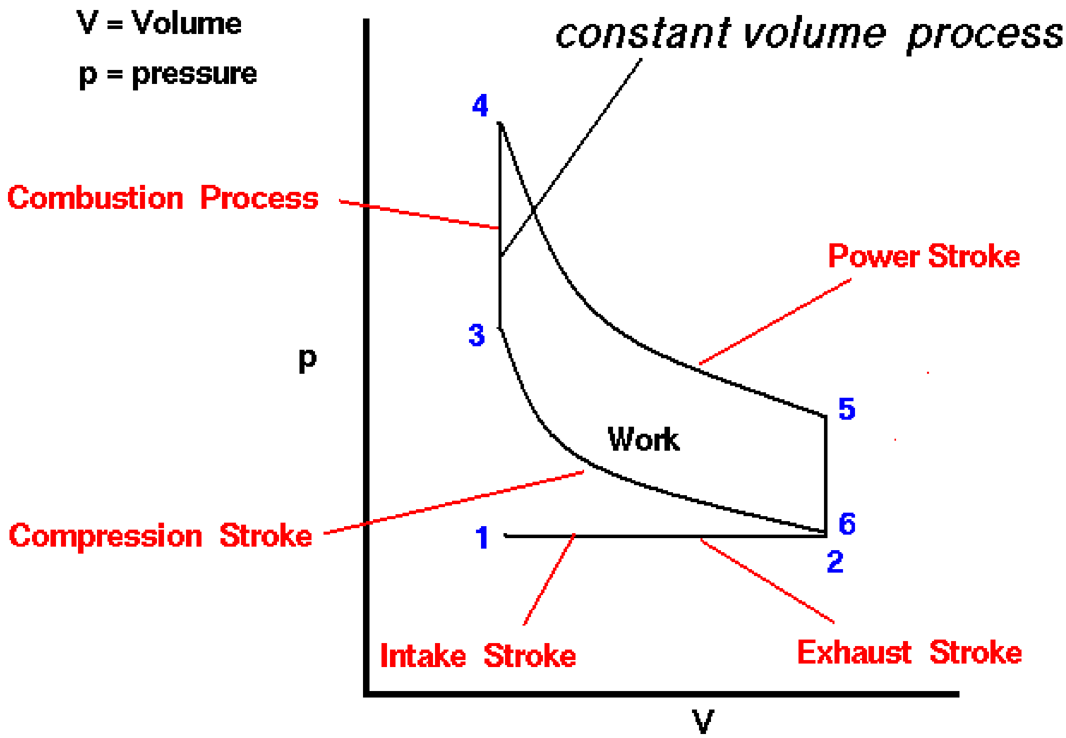

Figure 1: The Ideal Otto Cycle, image courtesy <http://www.grc.nasa.gov/WWW/K-12/airplane/otto.html>

Our understanding of this relationship is what drives our ability to determine the performance of an engine. The pressure created by the compression and combustion of the gas and air within the cylinder is what ultimately pushes down on the piston, creating the turning moment or torque of the crankshaft. Knowing how to calculate the relationship between volume and pressure gives us the ability to calculate how much work each cylinder is doing and then quantify how much power each cylinder is putting out over the course of the four stroke Otto cycle.

Each stroke has its own section of the p-V diagram shown in Figure 1. The four strokes, in order, are the intake stroke (point 1 to point 2), compression stroke (2 to 3), power stroke (4 to 5) and exhaust stroke (6 to 1). Air and gas are sucked into the cylinder through the intake valve at a constant pressure, very close to atmospheric pressure, over the course of the intake stroke. The air and gas is then compressed into the top of the cylinder as the piston moves from bottom center (BC) to top center (TC) over the course of the compression stroke. Combustion happens between points 3 and 4. At point 3, the spark from the spark plug ignites the compressed air and gas in the top of the cylinder. Ideally, this instantaneously raises the pressure to its maximum level at point 4. This pressure forces the piston down through the cylinder and rotates the crankshaft over the course of the power stroke. At point 5, the exhaust valve opens and under ideal conditions all the pressure in the cylinder instantaneously returns to close to atmospheric pressure at point 6. The by-products of combustion escape from the cylinder through the exhaust valve over the course of the rest of the exhaust stroke. We can capture these fluctuations in pressure as a function of volume in the following equations:

k = expansion coefficient
dispvol = displacement volume of the cylinder (area(A) * stroke)

$$P = 0.9 * p_{atm} \qquad \theta < \pi$$

$$P = p_{atm} * \left(\frac{vc + dispvol}{V}\right)^k \qquad \pi \le \theta < 2\pi$$

$$P = p_{\max} * \left(\frac{vc}{V}\right)^k \qquad 2\pi \le \theta < 3\pi$$

$$P = 1.1 * p_{atm} \qquad 3\pi \le \theta < 4\pi$$

Each of the four equations for pressure represents what the pressure in the cylinder is like during each of the four strokes. $\theta < \pi$ represents the intake stroke, $\pi \le \theta < 2\pi$ represents the compression stroke, $2\pi \le \theta < 3\pi$ represents the power stroke and $3\pi \le \theta < 4\pi$ represents the exhaust stroke.

Thus far, however, the p-V relationship had only been calculated under *ideal* conditions. But the conditions within the cylinder walls of a real IC engine are usually *not ideal*. It is only at mid-range RPMs where the engine is producing its peak amount of torque that conditions can be considered close to ideal. This presented us with a real challenge since Dr. Kumar's model only worked under ideal conditions and we wanted to be able to give performance figures across the entire RPM range of an engine. We wanted to try to figure out a way to continue to use Dr. Kumar's model as a baseline for computing the torque generated by an engine even at non-peak conditions. With this in mind, we studied what caused distortions in the p-V relationship and how we could capture these distortions in our simulation.

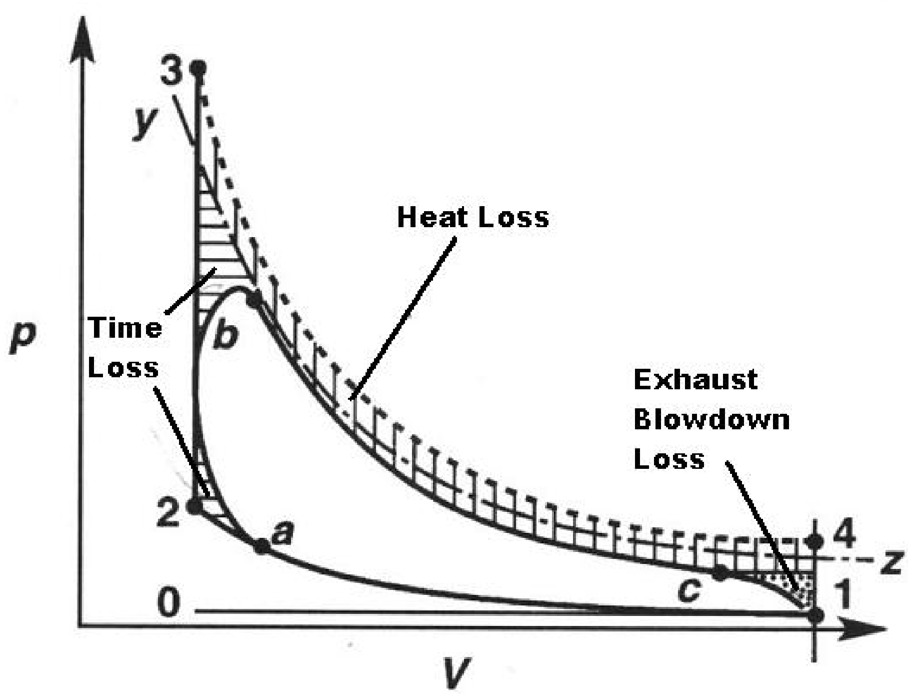

Figure 2: Courtesy Engines: An Introduction p. 7, annotated by Mike Saris

Figure 2 shows a more realistic view of what the p-V relationship looks like within a real cylinder superimposed on top of the ideal view. In this picture we can see 3 main causes for the distortions in the p-V relationship.

The first cause of distortion is time loss. This loss is due to the fact that combustion does not occur instantaneously as is assumed under ideal conditions. Since combustion is not instantaneous, the timing of when the spark plug must ignite the air-fuel mixture is moved up to point a, just before the compression stroke ends. Even after the compression stroke is complete, combustion is still not complete and the pressure in the cylinder is never able to get as high as is assumed under ideal conditions. Figure 2 shows how the p-V relationship is distorted by time loss both at the end of the compression stroke and at the beginning of the power stroke.

The second, and largest, cause of distortion is heat loss. The temperature within the cylinder rises greatly after combustion and some of this heat is lost to the walls of the cylinder and combustion chamber. This heat loss also means that the pressure within the cylinder is lower than under ideal conditions. Figure 2 shows how the entire power stroke is distorted by this heat loss.

The last, and smallest, cause of distortion is exhaust blowdown loss. In real four-stroke IC engines, the exhaust valve opens before the piston reaches bottom center. This means that the pressure in the cylinder begins to come down before the end of the power stroke. Figure 2 shows this distortion at the end of the power stroke.

There are also distortions on the intake and exhaust strokes that can not be seen in Figure 2. Figure 3 magnifies the intake and exhaust portion of the p-V diagrams to show these distortions.

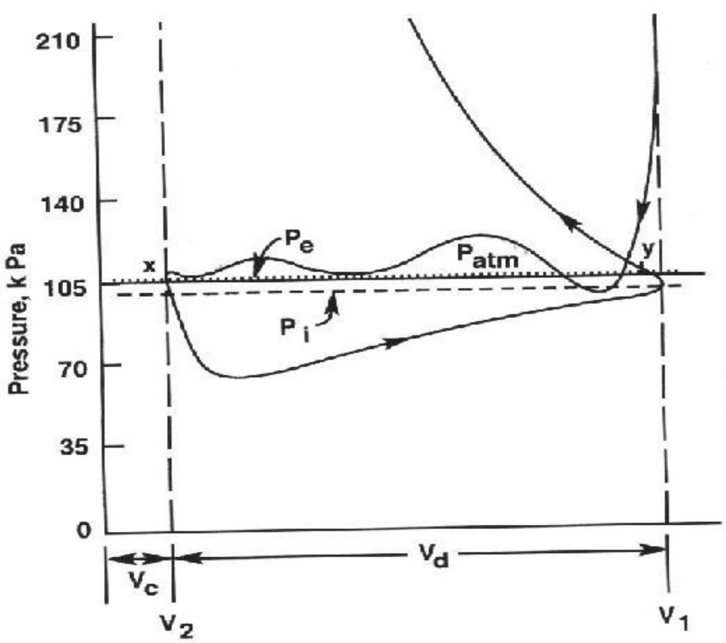

Figure 3: Courtesy <u>Engines: An Introduction</u> p. 8

The area between the curves seen in Figure 3 represents work that is being done on the gases in the cylinder during the intake and exhaust strokes. This work is called pumping loss and is ultimately lost energy that must be accounted for.

To account for all of these different losses (time, heat, exhaust blowdown and pumping) we had to generate distorted p-V diagrams that we could use in our simulation. The guidance of Ani Hsieh, a graduate student in the GRASP lab, was instrumental in helping us figure out how to accomplish this. We initially looked at MATLAB code Ani had for generating potential functions for general 2-D shapes in the plane. We thought that we could use this code to generate a representation of the distorted p-V relationship we could use in MATLAB since we knew the general shape of a distorted p-V diagram. In trying to use the code we had difficulty figuring out how to scale a picture of a distorted p-V diagram such that Ani's code could generate an accurate and usable MATLAB representation of a distorted p-V relationship. After further consultation with Ani, however, we were able to discover that MATLAB provided much simpler functionality which would allow us to capture the distortions.

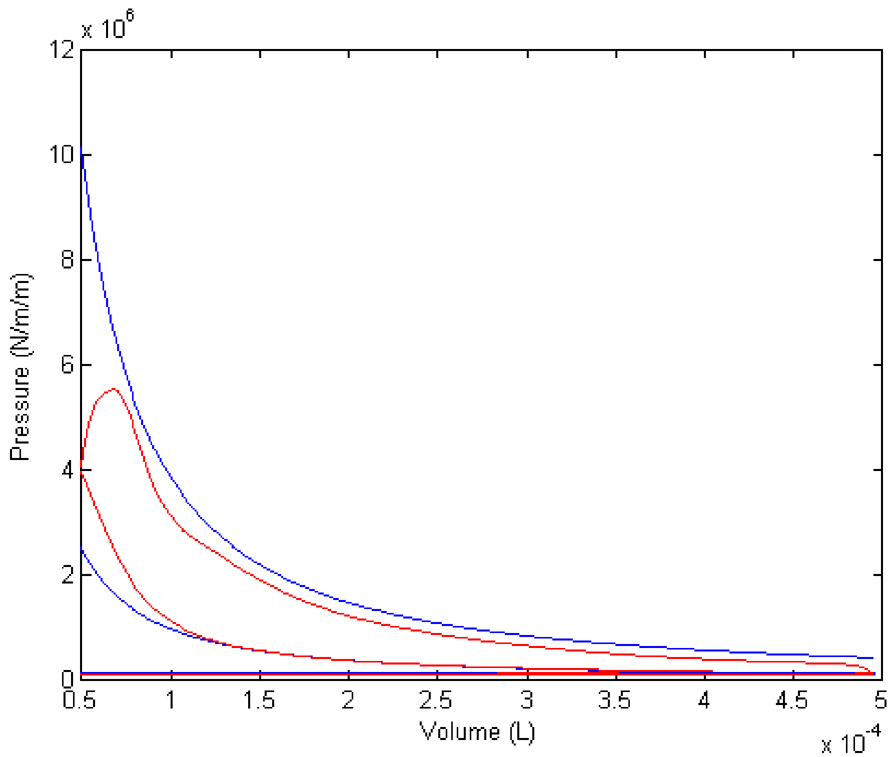

Figure 4: The red curves represent the maximally distorted p-V relationship. The blue curves represent the ideal p-V relationship.

First, we used MATLAB code given to us by Dr. Kumar which captured the equations derived from the Dynamic Analysis of the Internal Combustion Engine to help generate plots for an ideal p-V relationship. This gave us a correctly scaled view of what an ideal p-V relationship looks like. See the Appendix for the code we wrote to generate the ideal p-V diagram for the Audi 1.8 liter engine (idealpV.m). We then used MATLAB's ginput command to manually record points on the graph which lied along the curves in a maximally distorted p-V diagram. These points were recorded in four different sections corresponding to the four strokes each cylinder goes through. The points in each section were then spline fit to give us a continuous curve from which we could determine the cylinder pressure given the current volume of the cylinder and the stroke of the cycle the cylinder is in. Figure 4 shows a MATLAB plot of the maximally distorted p-V diagram as recorded by us superimposed on top of the ideal for an Audi 1.8 liter engine.

With ideal and maximally distorted p-V diagrams as a baseline, we generated 4 more intermediate levels of distortion by the same method used to generate the maximally distorted representation. Figure 5 shows a plot with these distortions in between the ideal and maximally distorted curves.

In the end, this process left us with 6 different representations of the relationship between pressure and volume ranging from ideal to maximally distorted. These representations are used to describe the p-V relationship at the various RPMs over which the engine operates, with most distortion occurring at low and high RPMs and least distortion occurring at mid-range or peak RPMs. Our naming scheme works such that "maximum" distortion is maximum distortion, level 1 distortion is slightly less distorted than maximum, level 2 slightly less than level 1 and so on down to level 4 which is just barely a distortion from the ideal.

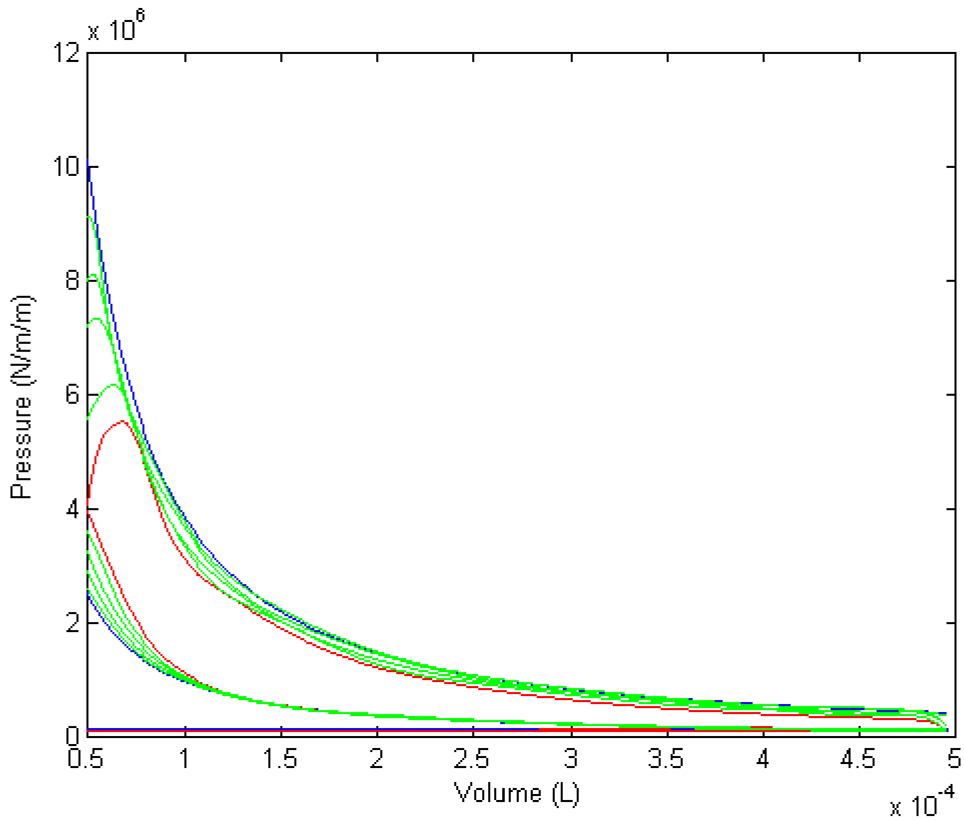

Figure 5: Ideal is still blue and maximally distorted is still red. The 4 levels in between are in green.

This brought us to a most crucial point where we could use Dr. Kumar’s model for a full range of RPMs, not just mid-range or peak RPMs, since we could now represent non-ideal p-V relationships within the cylinder. We used his derivation of an expression from Dynamic Analysis of the Internal Combustion Engine for each cylinder's contribution to the turning moment of the crankshaft (M_t) to finally calculate how much torque each cylinder is contributing to the engine. This contribution is broken down into two parts, the contribution of the pressure in the cylinder (M_p) and the contribution of the inertia of the parts in the cylinder even when there is no combustion (M_i). M_i is the term which accounts for the mass of the piston and connecting rod with m_P summing up their effect (see Dynamic Analysis of the Internal Combustion Engine pp.3-5 for an explanation of the derivation of m_P). M_p is the term which uses the distorted and ideal p-V representations as these representations determine the value of p:

m_{conn} = mass of the connecting rod
m_{piston} = mass of the piston

$$m_P = \frac{1}{2} m_{conn} + m_{piston}$$

$$M_p = (p - p_{atm}) A r \frac{\sin(\theta + \phi)}{\cos \phi}$$

$$M_i = \frac{m_P x'' r \sin(\theta + \phi)}{\cos \phi}$$

$$M_t = M_p + M_i$$

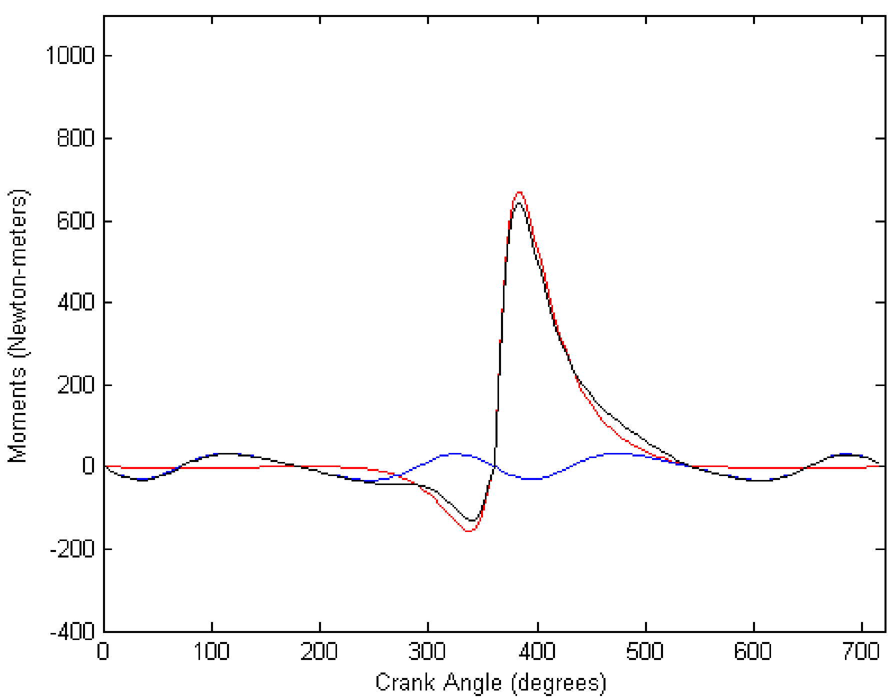

Figure 6: M_t is the black curve. M_i is the blue curve. M_p is the red curve.

Figure 6 shows a graph relating the moments created by a single cylinder to the crank angle for the Audi 1.8 liter engine.

The individual contributions of all of the cylinders in the engine are tied together by the crankshaft. Dr. Kumar's approach has shown us how to link the forces created by each cylinder onto a single crankshaft, and in so doing, translating the linear forces generated by the pistons inside the cylinders into the turning moment of the crankshaft. Each cylinder within the engine is identically simulated based on the equations and p-V representations derived above. When computing the total output of the engine, each cylinder's contribution is offset by a certain number of crank angle degrees such that each cylinder is in the appropriate part of the Otto cycle at any given point in time. The degree offset is determined by the number of cylinders in the engine. This offset is simple to compute taking 720, the number of degrees the crankshaft rotates in four strokes, and dividing it by the number of cylinders (i.e. 180 for 4 cylinders, 120 for 6 cylinders, 90 for 8 cylinders). Since our MATLAB simulation actually goes through the 720 degrees of the four stroke cycle in 200 steps, we actually shift the total moment equations (M_t) for each individual cylinder by 200 divided by the total number of cylinders. For example, in a four cylinder engine, cylinder one has intake beginning at step 1 and exhaust ending at step 200, cylinder two has intake beginning at step 51 and exhaust ending at step 50, cylinder three has intake beginning at step 101 and exhaust ending at step 100 and cylinder four has intake beginning at step 151 and ending at step 150.

Figure 7 shows a graph of what the total turning moment of the engine with contributions from all cylinders looks like as a function of crank angle for the Audi 1.8 liter engine.

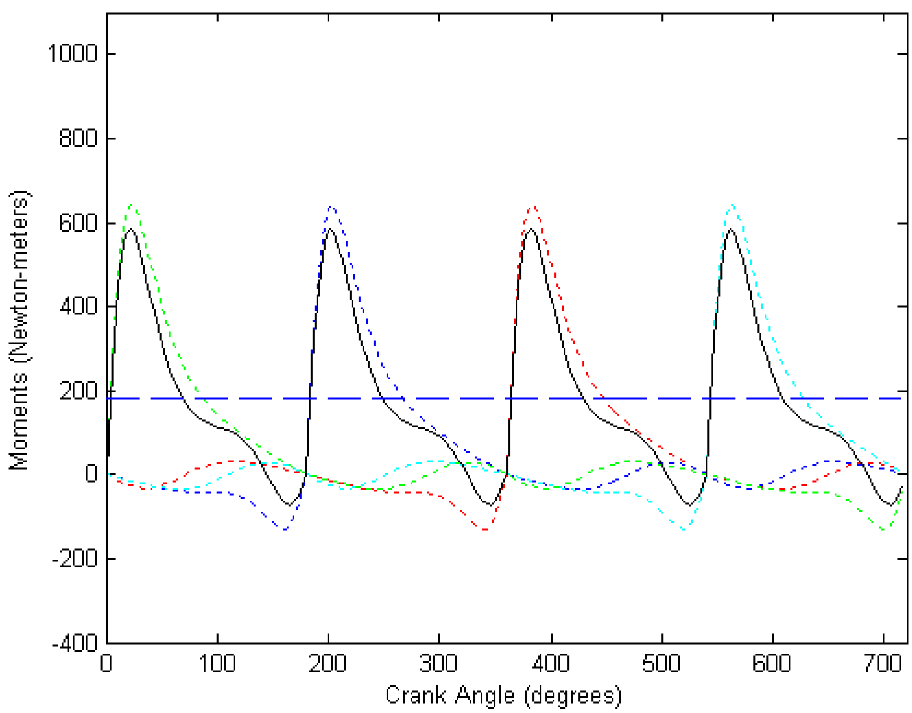

Figure 7: The black curve represents the total turning moment of the crank. Each colored and dashed curve is M_t for one of the four cylinders.

Since there are six different representations of the p-V relationship within the cylinders of the engine, we can calculate six different values for the torque generated by the engine. Each value represents the torque the engine would generate with a given level of distortion in the p-V relationship. Given dynamometer data from tests done on real engines, we can determine how much distortion there is in the p-V relationship within an engine at various RPMs across the engine's operating range.

Figure 8 shows torque and horsepower (HP) for a Ford 347 Stroker. Using the data from a real dynamometer test done for the Official Factory Guide to Building Ford Short Track Power, we were able to create a curve which has both a shape and magnitude very close to that which is seen in real life. Specifically in the case of the 347 Stroker, we found that the real torque at 3000 RPM coincided with what we labeled as level 2 distortion of the p-V relationship. We found level 3 distortion at 3800 RPM, level 4 at 4200 and 6200 RPM and an ideal p-V relationship at the peak of the torque curve at 5400 RPM. Based on these findings we also gave a theoretical view of what the torque and HP curves might look like at lower RPMs than we had real data for, assuming level 1 distortion at 2000 RPM and maximum distortion at 1000 RPM.

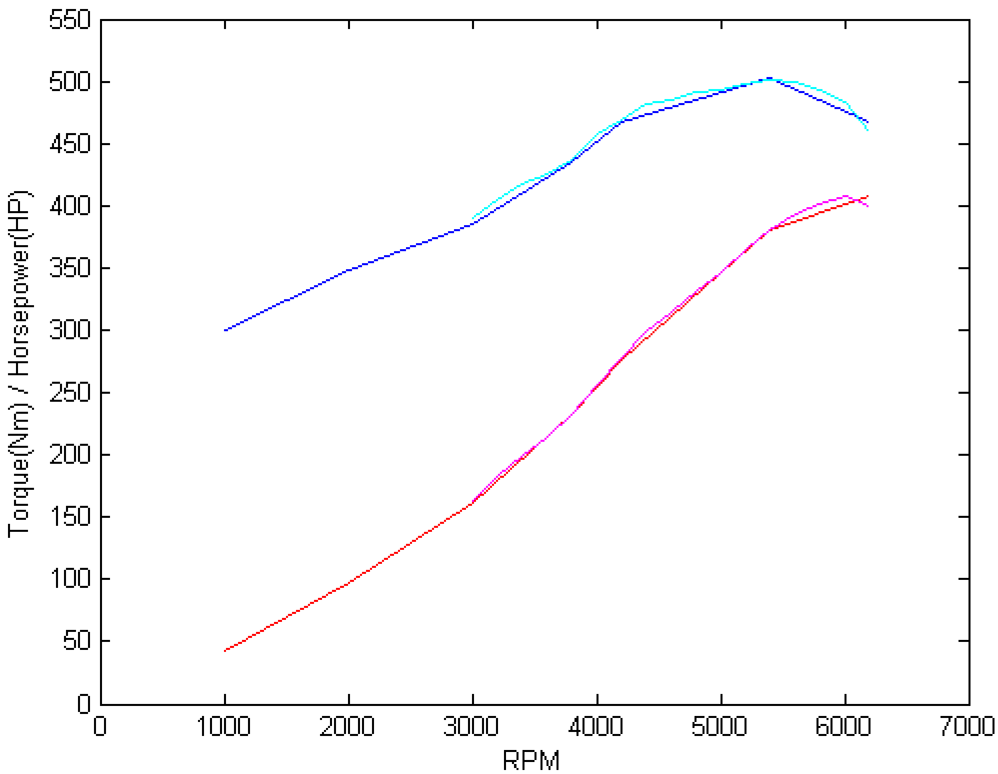

Figure 8: Plots of the HP and torque curves for a Ford 347 Stroker. The light blue curve is the torque curve acquired from performing a dynamometer test on a real 347 Stroker, the light red curve is the HP curve from the same test. The dark blue and dark red curves are torque and HP curves, respectively, generated using our simulation.

We applied this method of fitting our simulation's torque and HP curves differently to each engine we simulated. The engines we simulated include the Audi 1.8 liter engine

Dr. Kumar had originally based his model on, the Subaru 3.0 liter H-6 which came in the 2005 Subaru Outback, a Ford 347 Stroker built up in the Official Guide to Factory Guide to Building Ford Short Track Power and a theoretical Chevy 350 we pieced together out of a Summit Racing Equipment catalog. Note that we did not choose these engines by how we could fit their torque and HP curves but by the fact that these were the only engines for which we could find sufficient specifications.

For the Audi 1.8, we were only able to get complete torque and horsepower curves for the turbocharged version available in the United States. As a result, we went through the exercise of fitting our simulation's curves to this shape despite our knowledge that the naturally aspirated version of the engine would display a much different torque curve. Regardless of whether the shape was true to real life or not, the difference between the magnitudes of the curves was commensurate with what one would expect the differences to be between a naturally aspirated and turbocharged engine.

For the Subaru 3.0 liter H-6 engine, we were only able to get peak torque and HP figures. As a result, we built a torque curve based on what we knew from these peak numbers and then sculpted the rest of the curve by moving through the distortion levels as RPM rose up to and declined from its peak RPM. More concretely, we assumed the engine would exhibit maximum distortion at 2000 RPM, level 1 distortion at 2500 RPM, level 2 at 3000 RPM, level 3 at 3500 RPM, level 4 at 4000 RPM, ideal at 4500-6500 RPM and level 3 again at 7000 RPM. These choices allowed us to see peak torque and HP at RPMs that were very close to the real world peak RPMs for torque and HP.

For our theoretical Chevy 350 build up, we used the same combinations of distortion and RPM that we came up with for the Ford 347 Stroker. Once again, we did this despite the fact that we knew these two engines do not exhibit the same kind of torque and horsepower curves. The theoretical Chevy 350 buildup consists of parts picked directly from the Summit Racing Equipment catalog and represents a combination of parts that would be almost impossible to find real world data on. This engine was simulated, however, just to play with the parameters of our simulation and admittedly because one of the authors has a bias toward GM and Chevy and wanted to see how a theoretical Chevy 350 would stack up against a Ford 347 Stroker when the Ford's torque curve is forced upon it.

Our approach thus far has ignored the intake and fuel delivery system. This is by no means because this is not an important part of determining the performance of an IC engine. In fact, it is one of the most important parts of determining performance. But modeling this part of an IC engine proved to be out of the scope of our background and knowledge. One of our original ideas was to model the combustion within the combustion chamber and the cylinder in as detailed a fashion as our experience would allow and determine the performance of an engine based on this model. We contacted Noam Lior among others at SEAS that have expertise on combustion and their response was either to ignore us or say that they did not feel that we had enough prior experience to make an attempt at modeling combustion. Once we came upon Dr. Kumar's model, however, we had found something that we felt we could understand and add upon as well as a faculty member in Dr. Kumar who was willing to help us along the way.

As such, we capture the contribution of the intake and fuel delivery systems in the way that Dr. Kumar's model from the Dynamic Analysis of the Internal Combustion Engine does. This contribution comes from the value for maximum pressure (p_{max}). This is the pressure in the cylinder right after combustion completes under ideal conditions. The intake and fuel delivery systems have great influence on this value since they mix and introduce into the cylinder the air-fuel mixture that is ignited and combusted. How this process happens and its influence on combustion would have been the focus of a model like the one we were originally interested in pursuing. But without this piece, figuring out the value for p_{max} became an optimization problem. Dr. Kumar had already estimated this pressure to be 100 atm for the Audi 1.8 before we had approached him. For the subsequent simulations we did of a Subaru 3.0 H-6, a Ford 347 Stroker and a theoretical Chevy 350 buildup we took the real world performance data we had and optimized p_{max} so that under ideal p-V conditions our simulation produced peak torque numbers that were in line with the real world data. We settled on values of 107 atm for the Subaru, 96.5 atm for the Ford 347 and made a best guess of 95 atm for the Chevy 350 based on the value for the Ford 347 since no real world data could be available for the theoretical build up. Once we had these values, we were able to go about generating the distorted p-V diagrams upon which we could base our calculations of performance at non-ideal RPMs.

In deriving and calculating all of these values, we came up with a slew of graphs and figures which describe the performance of the four engines we simulated. To make this data more accessible to others, we wrote a Java GUI using the Swing framework. This GUI organizes the graphs and figures first by which engine they pertain to, then by whether they apply to the engine's entire operating range or just a single RPM. Graphs and figures for the engines' entire operating range show complete torque and HP curves generated by our model and compare them to real world data. Graphs and figures which apply to a single RPM provide a picture of what the moments and pressure look like within a single cylinder and then how all the moments from all cylinders in the engine aggregate to give a total torque output. All graphs and figures are displayed in their own windows so that they can be moved around, resized and tiled across the screen so they can be compared with other graphs and figures. In this way, users of the GUI can compare figures from different engines to see how each differs from the others.

Conclusion

When we started this project, we had only a very basic idea of how a four stroke internal combustion engine works. By the end, we had created a model which gave us great insight into how and why engines perform the way they do.

We started with a model that was only suitable for engines operating under ideal conditions. This meant we could only model the engine as it worked at peak RPMs. This led us in to an even more careful study of our original model and how real engines operate in conditions which are not ideal. As a result, we were able to gain an understanding of how to adapt our model so that it could apply across a four stroke IC engine's entire operating range. Once we captured this understanding into our model, we were able to generate lots of interesting performance data on the four engines we had chosen to simulate. To generate this data we had to optimize the model for each engine to fit with the available real world performance data. This led to an exercise which allowed us to figure out theoretical values of p_{max} for each engine as well as how distorted the p-V relationships were at various RPMs across each engine's operating range. The end of the line performance data this allowed us to find included:

- torque and HP curves across the engine's entire operating range
- moments generated within each cylinder at all crank angles
- pressure in each cylinder at all crank angles
- the total turning moment of the crankshaft at low, mid, high and peak RPMs

This data was then organized and presented in the form of a Java GUI to provide others the ability to look at it and compare figures across different engines.

The task of trying to generalize the ideal model to apply to all RPMs at first seemed quite daunting. To be honest, we were not sure if we would be able to do it at all. But like anything else, it seems as if hard work paid off in the end and we were able to put together a way of expanding our original model so that it could give a picture of engine performance across the entire operating range.

A task, however, that turned out harder than expected was finding engine specifications to use in our model. We contacted 15-20 different automobile manufacturers and sources of automotive data from General Motors to Honda to Ward's Auto. Despite this effort we were still only able to get a usable set of specifications for four engines. Even once we had the specifications we still needed to try to find real world performance data on these engines which was another chore in itself. We were only able to get full torque and HP curves for one engine, the Ford 347 Stroker. We could only get turbocharged data on the Audi 1.8, peak data on the Subaru 3.0 H-6 and of course no real life data can be found for the theoretical Chevy 350 buildup.

But even after all that effort was put forth, with a chance to look back, it seems that there is still much more that can be done with this model.

First, all distorted p-V diagrams had to essentially be made by hand using MATLAB's ginput command. This greatly limited the number of engines that we could attempt to model. If this process of generating distorted p-V diagrams could be automated, this model would easily apply to any and all four stroke IC engines. All one would have to do to model any four stroke IC engine is have available the engine's bore, stroke, compression ratio, connecting rod length, connecting rod mass, piston mass, crankshaft mass and a value for p_{max}. These specifications could then be plugged in to a fully parameterized version of our model, providing the user with a full picture of the performance of their engine without having to go anywhere near a dynamometer.

And one of those specifications, p_{max}, could be determined by the kind of model that we initially considered building. As was mentioned previously, one could theoretically attempt a more detailed model of how the intake and fuel delivery system affect the combustion within the cylinder. From this analysis one could calculate p_{max} without having to measure that value in real life or find it from some other source. This would be a huge step since we were not able to find a single source, not auto manufacturer's, not the internet, not anybody, who could give us a value for p_{max} for any engine. If one could compute p_{max} on their own they would have a valuable piece of data that is very hard to come by and is arguably the most crucial piece of data to determining the performance of an engine.

If these two additional tasks could be completed, a whole new world of potential could be unlocked for this model.

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Appendix

idealpV.m

```
k=1.4;
pmax=(14.7*100);         % maximum pressure = 100 atmospheres
patm = 14.7;             % atmospheric pressure
pmax =pmax*6895;         % 1 psi = 6895 N/m/m (parameterize)
patm =patm*6895;
vc= 0.05;                % clearance volume in liters
vc=vc*10^(-3);           % convert clearance volume into m^3
stroke = 86.4*10^-3;     % in m, stroke = 86.4 mm
bore = 81*10^-3;
area=bore^2*pi/4;        % calculate cross sectional area in m^3
dispvol = area*stroke;   % calculate displacement volume

Vol = vc:.000001:vc+dispvol;
intake = 0.9*patm;
exhaust = 1.1*patm;
for iter = 1:length(Vol)
   compression(iter) = patm*((vc+dispvol)/Vol(iter))^k
   power(iter) = pmax*(vc/Vol(iter))^k
end;
figure(1);
plot(Vol,intake,'r', Vol, exhaust, 'b')
figure(2);
plot(Vol, intake, 'r' , Vol, compression,'b', Vol, power, 'g', Vol, exhaust, 'c');
```