Investigating the Effects of Knock on Engine Thermodynamics

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Abstract

The effects of knock on engine thermodynamics and efficiencies were investigated by separating cycles that were calculated to knock from cycles that weren't using an auto-ignition delay. The knock cycles on average had peak temperatures and pressures due to the violent effects of auto-ignition as expected. The narrower heat release rates and high coefficient of variations during combustion proved these assumptions to be true. The knock cycles had higher peak temperatures and pressures throughout. However, the loss of energy due to vibration and the high variance associated with lead to slightly lower efficiencies and higher specific fuel consumptions—both of which are unfavorable. These losses also led to a slightly lower total heat release. The average total heat released was also higher for the non-knocking cycles by 5.32 J at 120^{o} after TDC. The knock cycles efficiencies were found to be 0.12, 0.19, and 0.12 percent less efficient than the mean efficiency in terms of net, gross, and net fuel conversion efficiencies respectively while offering 0.19% more mechanical efficiency. The cycles that didn't knock were found to be 0.05, 0.08, and 0.05 percent more efficient then the mean efficiency in terms of net, gross, and net fuel conversion efficiencies respectively while offering 0.08% less mechanical efficiency.

Introduction

Internal combustion engines have long been the standard for powering motor vehicles. Their impressive power to weight ratio is almost unrivaled, making them an attractive power source. However, as CO_2 emissions continue to rise along with global temperatures, the combustion engine's emissions and dependence on fossil fuels has made it a target for controversy. As the world shifts towards electric vehicles, automobile manufacturers will soon be confronted with an important decision. Either adhere to growing public and political pressures and begin phasing out the internal combustion engine in favor of a fully electric fleet, or, begin improving on the efficiencies of the internal combustion engine while limiting emissions.

While at the surface the electric vehicle seems to offer enormous environmental benefits compared to its combustion engine counter parts, those notions are ill conceived. While the emissions directly from an electric vehicle are essentially zero, there are issues that lie deeper. In investigating the environmental impacts of electric vehicles Dessus et al. (1994) says "At world level, the impact of the electric vehicle when considering the greenhouse effect nevertheless remains less detrimental than that of internal combustion vehicles. It is only in Denmark, a country where 90% of electricity is produced from coal, that other fuels (LPG and gas oil EURO standard 96) emit greenhouse gas emissions to those of the electric vehicle." This is because electric vehicles get their energy from the nation's power grid which is heavily reliant on fossil fuels and has an average efficiency of approximately 33.16% (Lawler, 2018). The United States is projected to still produce 67% of its electricity from coal and natural gas in the year 2040 (Sesay, 2018). In addition, the amount of CO_2 emitted from producing the large batteries needed for these vehicles is equivalent to that of driving around an average ICE engine for 2-8 years (Lawler, 2018).

As the so called “benefits” of electric vehicles continue to dominate main-stream media, it is up to engineers to provide methods of improving internal combustion engines and leave no speculation as to which of the two is better. Numerous methods are currently being researched in order to come up with such a break-through-- everything from advanced combustion modes to alternative engine architectures. Engine performance, like all other systems in nature, has tradeoffs. Operating “lean” (Air:Fuel ration greater than stoichiometric) improves overall engine efficiency. However, operating under such conditions leads to the production of high temperature NO_x which is harmful to the environment cannot be easily be cleaned up. The cost of cleaning up such emissions is economically unfavorable to manufacturers and as such, this method cannot be used as of now. Another way to increase engine efficiency is to operate at higher compression ratios. By doing so, more work can be extracted from the engine cylinder and less heat is rejected to the exhaust, thereby increasing engine efficiency (Lawler, 2018). Diesel engines operate at relatively high compression ratios (15-20) but form soot due to the heterogenous nature of the air and fuel mixture. SI engines operated at higher compression ratios run into the problem of “knocking” which occurs when the air and fuel mixture auto ignites before being consumed by the flame. This sharp rise in pressure causes the engine to ring and essentially steals energy from the cylinder, thereby hurting efficiencies (Lawler, 2018). As defined by Zavala and Folkerts, knock is “Noise which is transmitted through the engine structure when essentially spontaneous ignition of a portion of the end-gas occurs (Zavlava and Folkerts, 2011).”

It is not enough to just know why knock is detrimental qualitatively. Analysis must be done in order to determine its quantitative effects. In doing so, judgements can be made as to whether it is allowable to operate at such conditions for any period of time. When the effects are known, they can be better accounted because engineers can determine where the efficiencies are being lost and plan accordingly. If the benefits of operating at higher compression ratios and advanced combustion phasing outweigh the momentary drawbacks of knock, perhaps engines can be operated closer to the edge of knock more confidently. In addition to thermodynamic properties of the engine, the material properties must also be taken into account as the cyclic stresses brought on by knock can be extremely detrimental to the engine material. However, that is outside the scope of the paper and only the thermodynamic properties shall be evaluated.

Methodology

In order to complete the analysis, a four-stroke, one-cylinder CFR engine will be operated at 60 kPa with a compression ratio of 10 and a spark timing of 11 deg BTDC. The pressure inside the engine will be recorded at 0.2-degree increments of the crank angle over the 720-degree cycle. The data will then be loaded into MATLAB to be analyzed. Because filtering may manipulate data in a way that is not representative of the actual cycle, it will not be used. Heat transfer losses will be calculated by integrating the apparent heat release rate from -120 to 120 ATDC and adding it to the heat transferred to the cylinder wall. The heat transfer coefficient, h, will be calculated using known relations with gas velocity, bulk temperature, pressure, and bore. In order to calculate the heat transfer losses to the wall, the cylinder wall temperature will be assumed to be a constant value of 400 K. Once both these are calculated for the 200 cycles, they will be added to obtain an "apparent gross heat release." This will be assumed to take into account blow-by losses as well. Next, each cycle will be normalized by its max heat release value to obtain a mass-fraction burned curve. From this curve, the CA50 for each individual cycle can be calculated. Finally, the efficiencies of the engine will be determined using the measured torque at the dyno and the work calculated by numerically integrating the pressures over the volume trace. The MATLAB code can be found the appendix.

After completing the analysis all 200 individual cycles, a calculated ignition delay will be used to determine which of the cycles knock. Once identified, the cycles that are calculated to knock will be isolated from those that aren't and a separate analysis will be done on those cycles similar to that of the original one. Once this process is done, the data will be reviewed in order to determine the effects of knock on this particular set of data and whether or not is holds up against theory and exiting literature.

Results

In order to isolate the effects of knock, the knock cycles must be isolated from the rest of the cycles that didn't. This was done using the equation:

$$1 = \int_{IVC}^{Ign_{CA}} \frac{1}{\tau} dCA$$

(Lawler 2018) (1)

Where τ is calculating using the equation:

$$\tau = 1.3 * 10^{-4} * p^{-1.05} * \phi^{-0.77} * x_{O_2}{}^{-1.41} * \exp\left(\frac{33{,}700}{R * T}\right)$$

(Lawler 2018) (2)

Where:

p=Cylinder pressure in atm

ϕ=Equivalence Ratio

x_{O2}=Percent oxygen in mixture

R=Universal gas constant in cal/mol-K

T=Isentropic Unburned Temperature

By taking an “indefinite integral” of 1/τ and seeing if the integral crosses one, the cycles that were knocking could be isolated. The results of the integral can be seen in figure 1 on the below:

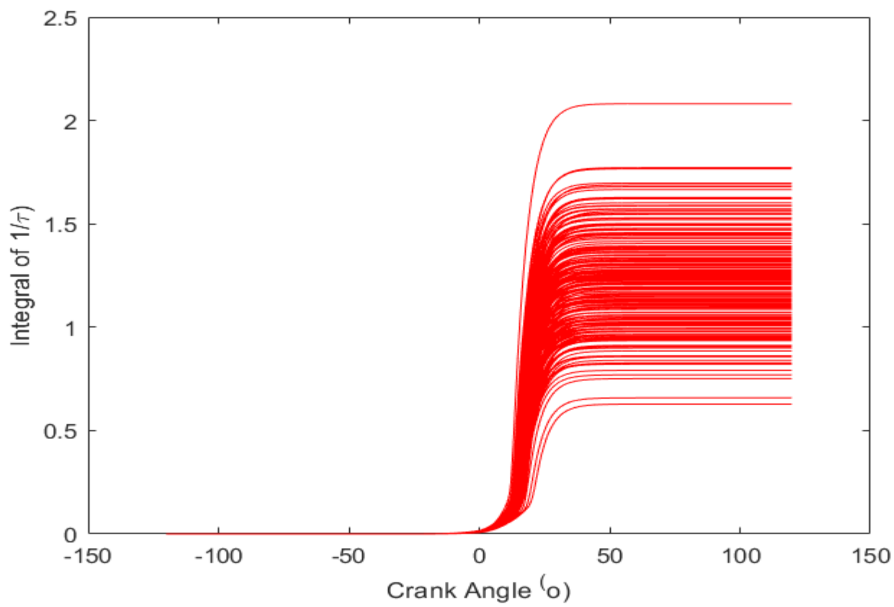

Figure 1. Auto-ignition criteria used to determine if knock was present. As seen in the graph, not all cycles auto-ignite.

The data from figure 1 was used to isolate knock cycles. Not only does this parameter predict the presence of knock, it can be used as a gauge to determine to what degree. Cycles can be further classified as high, moderate, low, and no-knock. For the data obtained in this lab, only 200 cycles were recorded and of those, only 37 did not knock. There was not enough data to separate the data further and still obtain meaningful values.

Because knock occurs from auto ignition which is characterized by rapid rises in temperature and pressure, it is expected to have higher pressure values throughout the cycle in comparison to the no-knock cases which can be seen in figure 2.

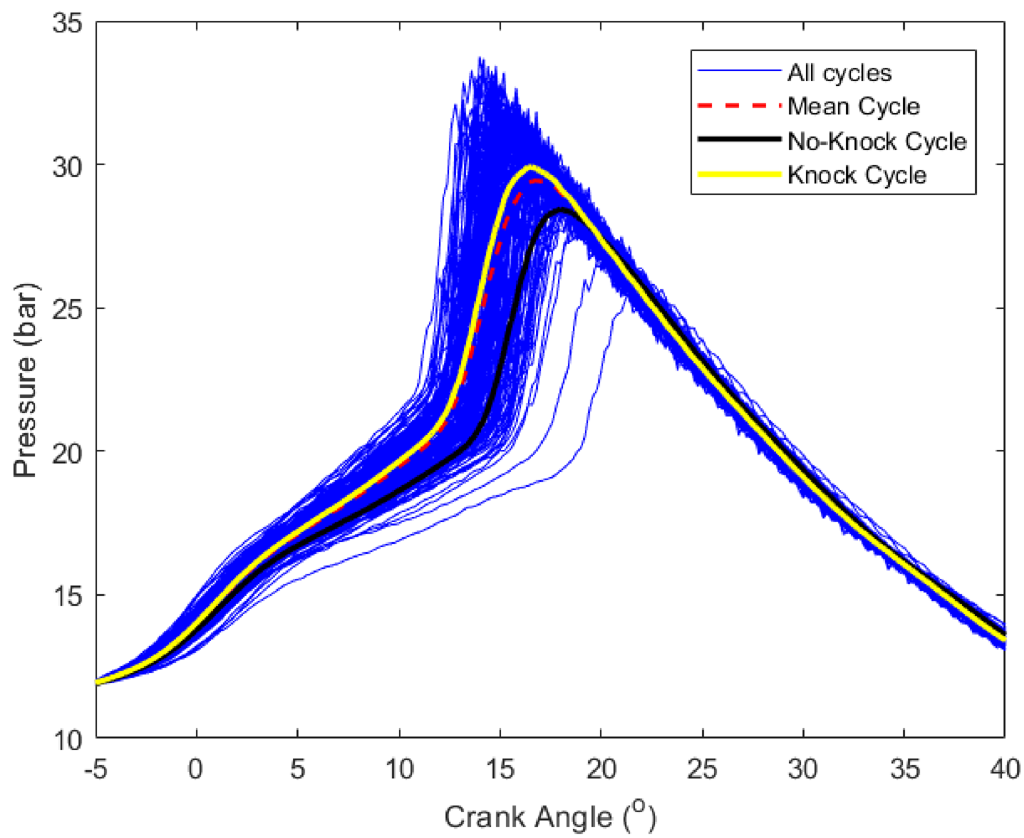

Figure 2. Graph of pressures for the average of the knock and no-knock cases compared to the average of all 200 cycles

The graph agrees with theory and intuition. The knock cycle is at a higher pressure throughout combustion. The peak difference between them occurs at 17 ATDC which is slightly after the average CA50 of the knock cycles. The difference in their peak pressures is 1.47 bar. Because there were more than 5 times as many cycles that knocked as opposed to those that didn't (163 and 37 respectively), the mean pressure is much closer to the knock pressures as it is weighted heavier. However, it is likely that the mean cycle would lean towards the knock pressures regardless as the higher spikes and larger deviations as a result of auto-ignition will skew the average towards it anyway.

The same conclusions can be drawn from a log-log plot of the Pressure vs Volume graph shown in figure 3a:

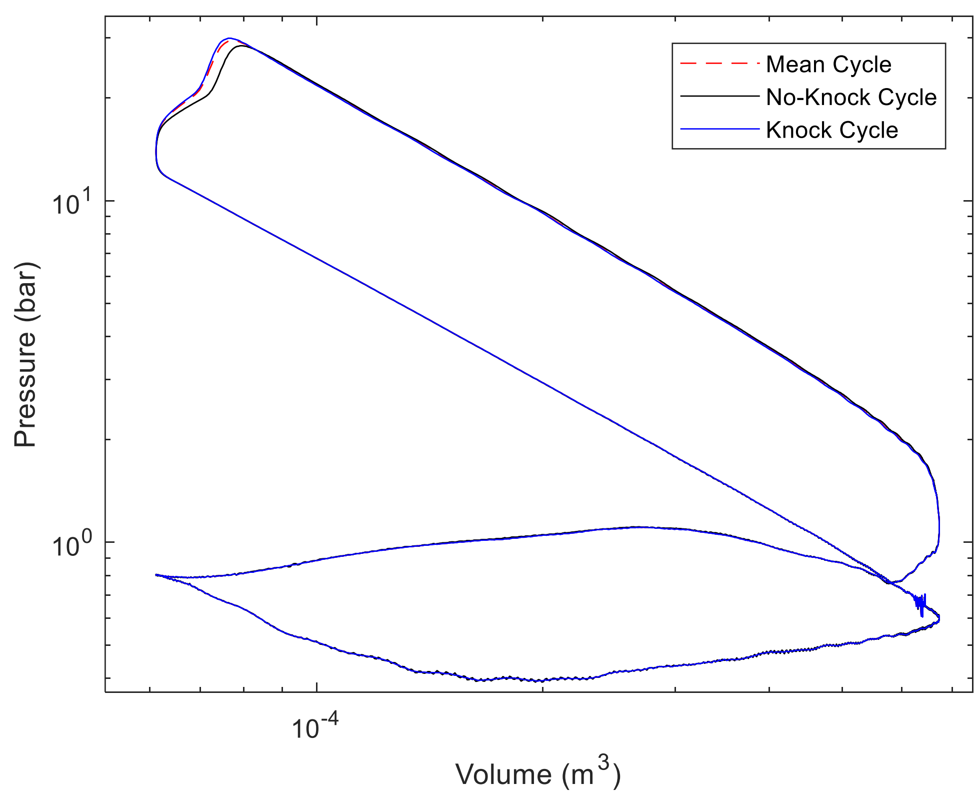

Figure 3a. Pressure vs Volume on a logarithmic scale

Again, it can be seen that the knock cycle has slightly greater pressures throughout combustion. Outside of combustion, the pressure traces are merely identical. This is because knock occurs around combustion. Once it subsides, the rest of the cycle operates under the same parameters (e.g. compression ratio and spark timing). An interesting phenomena can be seen around IVC when the pressure starts to ring just as it does during knock. Taking a closer look in figure 3b:

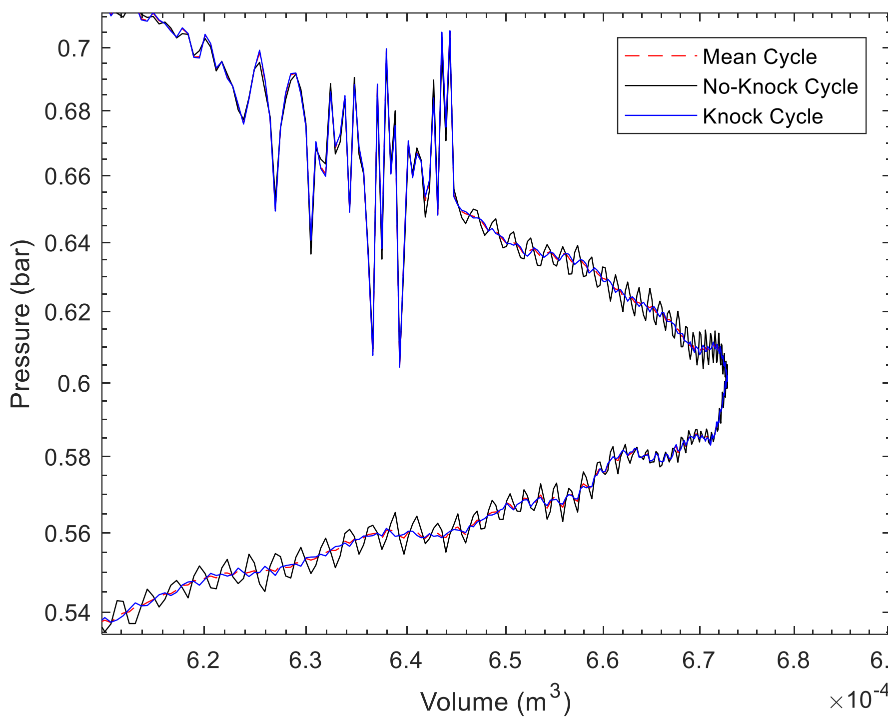

Figure 3b. Zoomed-in view of Figure 3a

Figure 3b shows that after the large ringing in the pressures have subsided around 6.4E-4 m^3, while the knock pressures have stabilized, the no-knock cases continue to ring. This could be because the frequency of this ringing is around 180^o out of phase with the knock frequency, cancelling each other out. The ensembled average along with the individual cycles can be seen for both the no-knock and knock cases in figures 4a and 4b respectively:

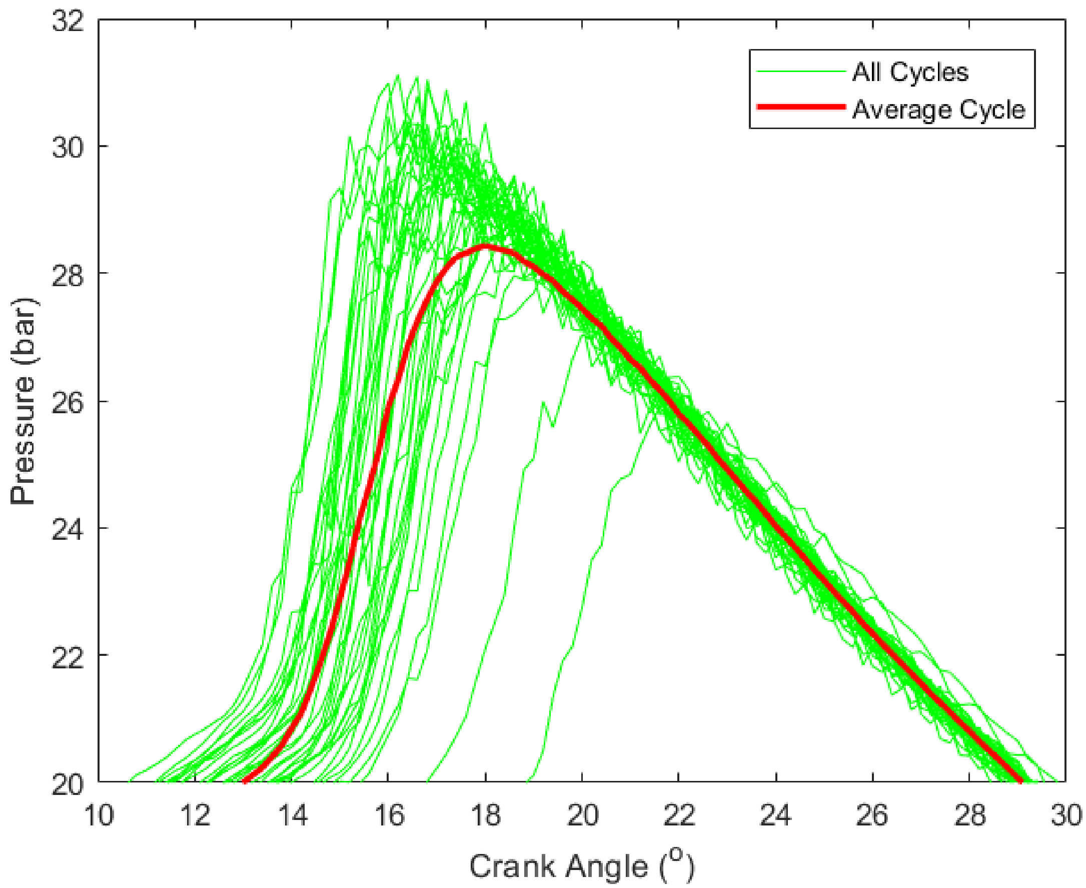

Figure 4a: Pressure vs Crank Angle for the cycles that didn't knock

Figure 4a shows that the criteria used for determining knock were relatively accurate. The pressures, although noisy, don't ring for the most part. Because there were so few cycles that fit the criteria, we see the average skewed downwards by two major outliers. The peak pressure of these cycles is slightly above peak average of the knock cycle differing by 4.05%.

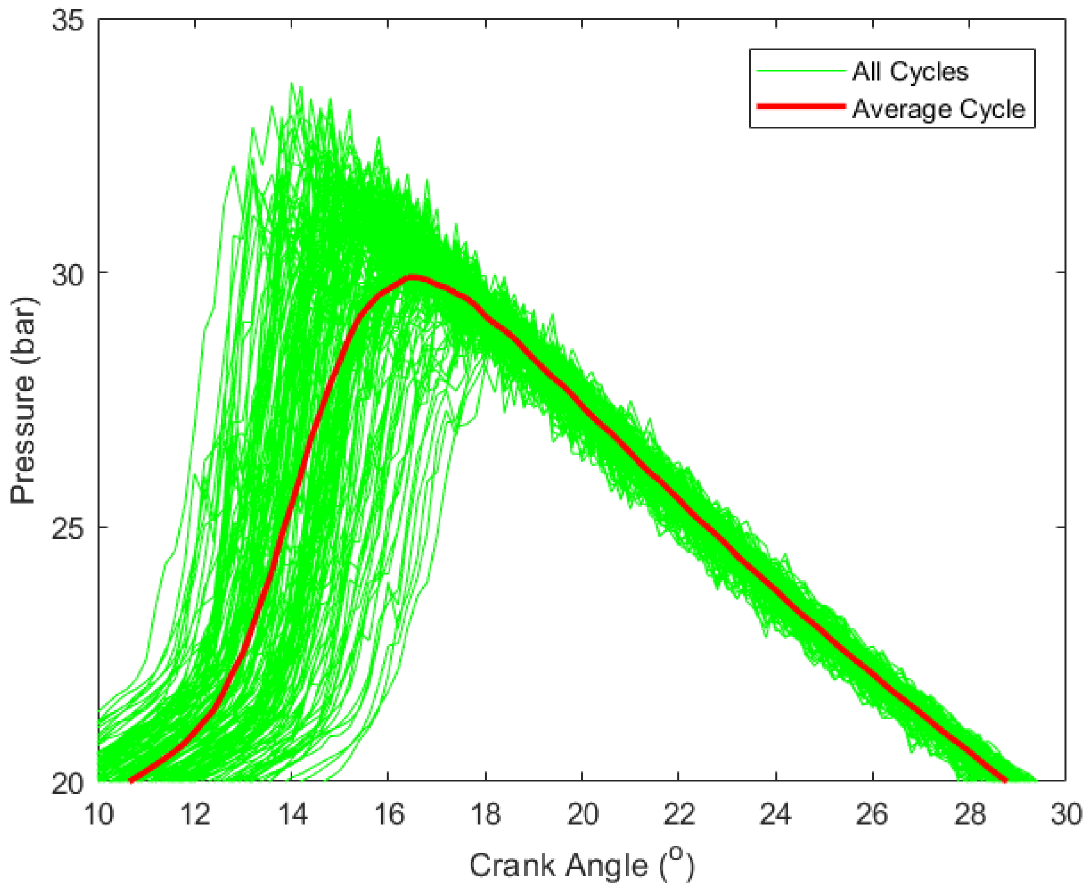

Figure 4b. Pressure vs Crank Angle for knocking cycles

Figure 4b shows the ensembled average pressure trace of the knock cycles in relation to the individual cycles. We can see the onset of ringing occur at the peak pressure and continue throughout expansion. Because the sample size was relatively large compared to the no-knock cases, the average value better represents the cycle.

The simulated motored pressure used to determine the gas velocity was found by using the relation:

$$P_{simulated\ motoring}(i) = P_{IVC} * \left(\frac{V_{IVC}}{V(i)}\right)^{k}$$

(Lawler 2018) (3)

Where:

P_{IVC}=Pressure at IVC

V_{IVC}=Volume at IVC

V_i=Volume inside cylinder

k=Polytropic exponent

A for loop was used substituting values of k ranging from zero to five in order to find the k value that most closely matched the actual pressure trace from IVC to 15 BTDC as these values should be the same because they are not influenced by combustion. The k value was found to be 1.25. The result can be seen in figure 5:

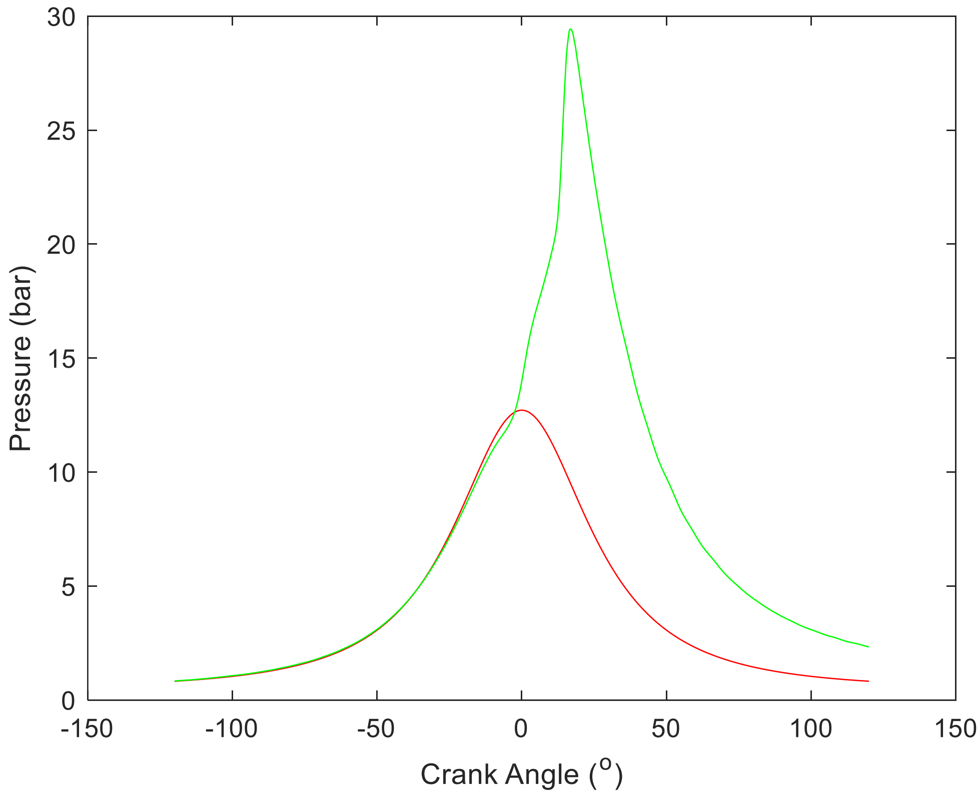

Figure 5. Pressure vs Crank Angle for simulated and ensembled fired case

Figure 5 shows how much pressure is produced by combustion. The peak values differ by 16.72 bars. Since pressures for the knock and no knock are relatively the same before combustion, so the simulated pressure will be the same with the same k value. Therefore, there is no need to recalculate it.

After obtaining the simulated motored pressure the gas velocities can be calculated using the equation:

$$w = C1 * Sp + C2 * Vd * T_{ref} * \frac{(p_{cyl} - p_{sim,motoring})}{(p_{ref} * V_{ref})}$$ (Lawler 2018) (4)

Where:

C1=2.28

C2=0.00324

S_p=Mean piston speed (m/s)

V_d=Volume displaced by engine (m^3)

T_{ref}=Intake temperature of air

P_{cyl}=Pressure inside the cylinder

$P_{sim,motoring}$=Simulated motored pressure

P_{ref}=Pressure at BDC

V_{ref}=Volume at BDC

This is an essential property that was used to determine the knock condition. As stated earlier, knocking can be characterized as a rapid expansion of the gasses in the cylinder chamber. This results in high gas velocities. The difference between the different conditions can be seen in figure 6:

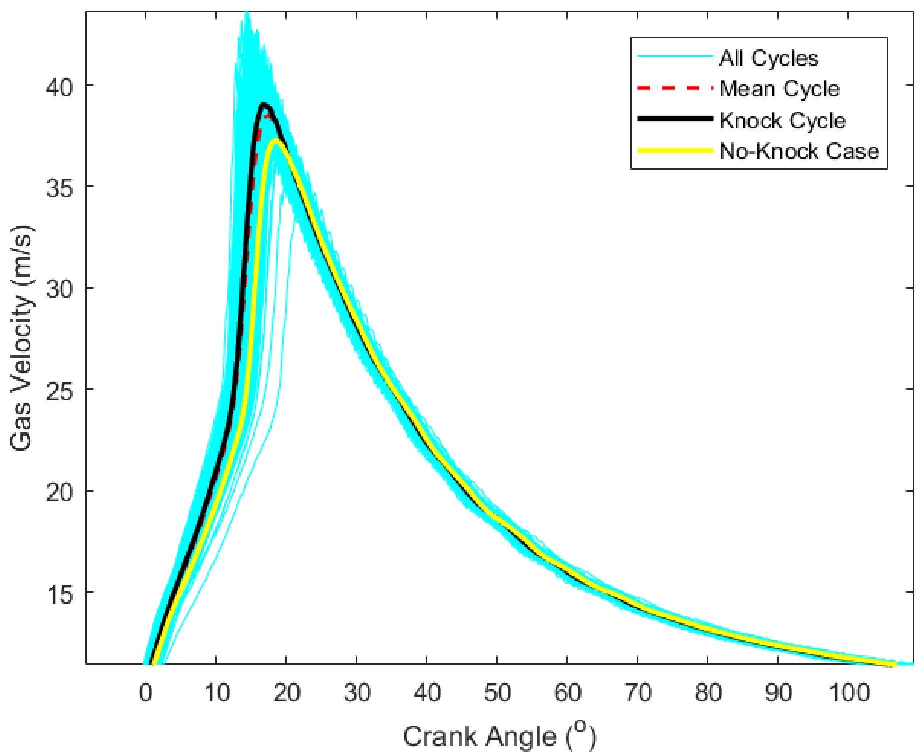

Figure 6. Gas velocities for all conditions

Figure 6 shows the gas velocities in the cylinder chamber during combustion and expansion. Because the velocities is a function of the difference in pressure between the fired and simulated motored pressures, the graph closely resembles the shape of the pressure vs crank angle plot in figure 2 because all conditions use the same motored pressure as reference.

After gas velocities are calculated, the bulk temperatures can be calculated and analyzed to see thermal effects of knock. A graph of all conditions can be seen in figure 7:

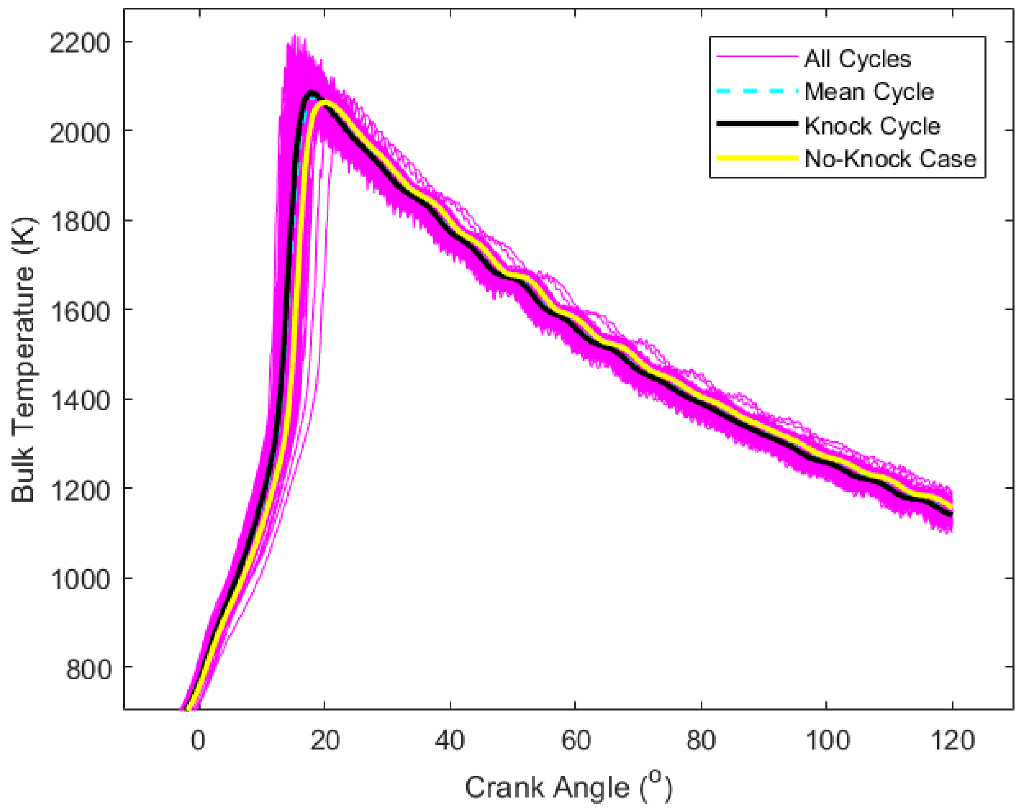

Figure 7. Bulk Temperature vs Crank Angle

Figure 7 shows that up until the peak pressure, the knock cases are hotter. As they continue to rise, the no-knock start dropping in temperature. At about 22° ATDC we see the knock temperatures drop below the no-knock cases and continue to stay below. This could indicate that the knocking causes the cylinder to lose energy to the ambient as it vibrates, decreasing bulk temperatures inside. Figures 8a and 8b shows the bulk temperatures for the knock and no-knock cycles separately:

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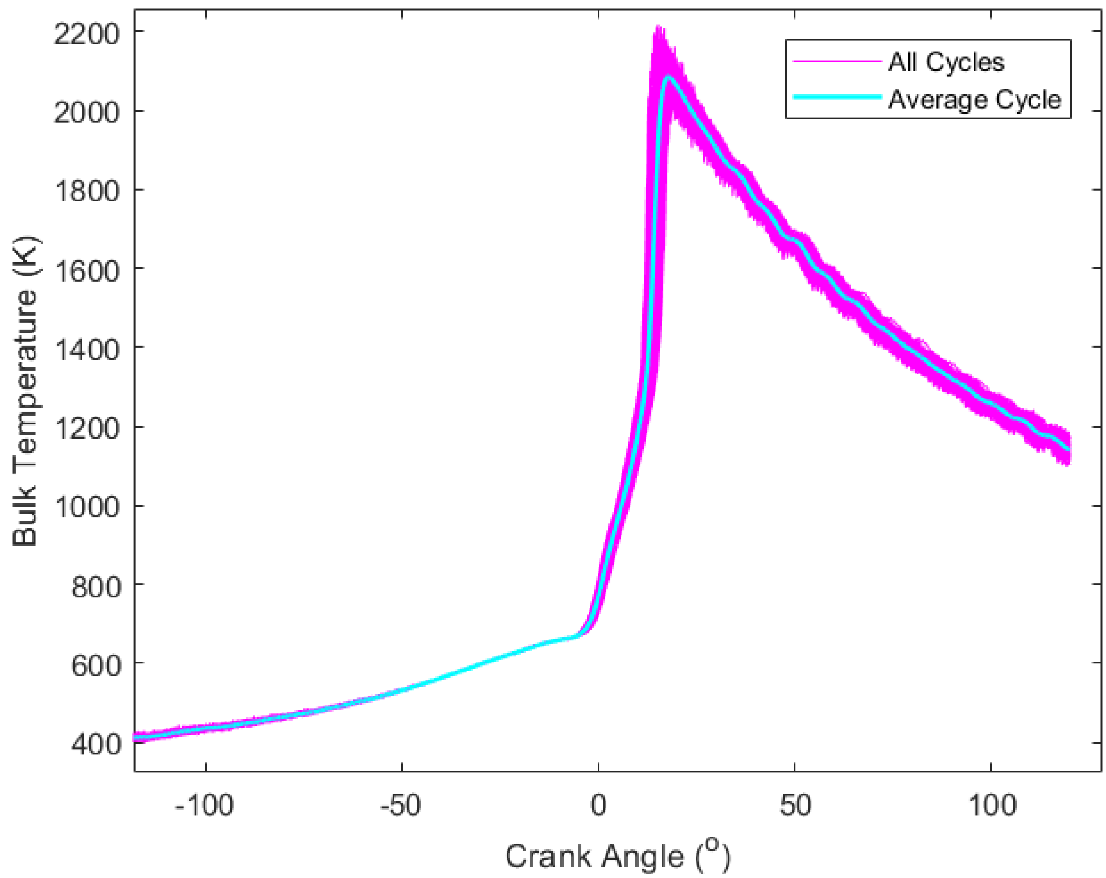

Figure 8a. Bulk Temperature vs Crank Angle for knock cycles

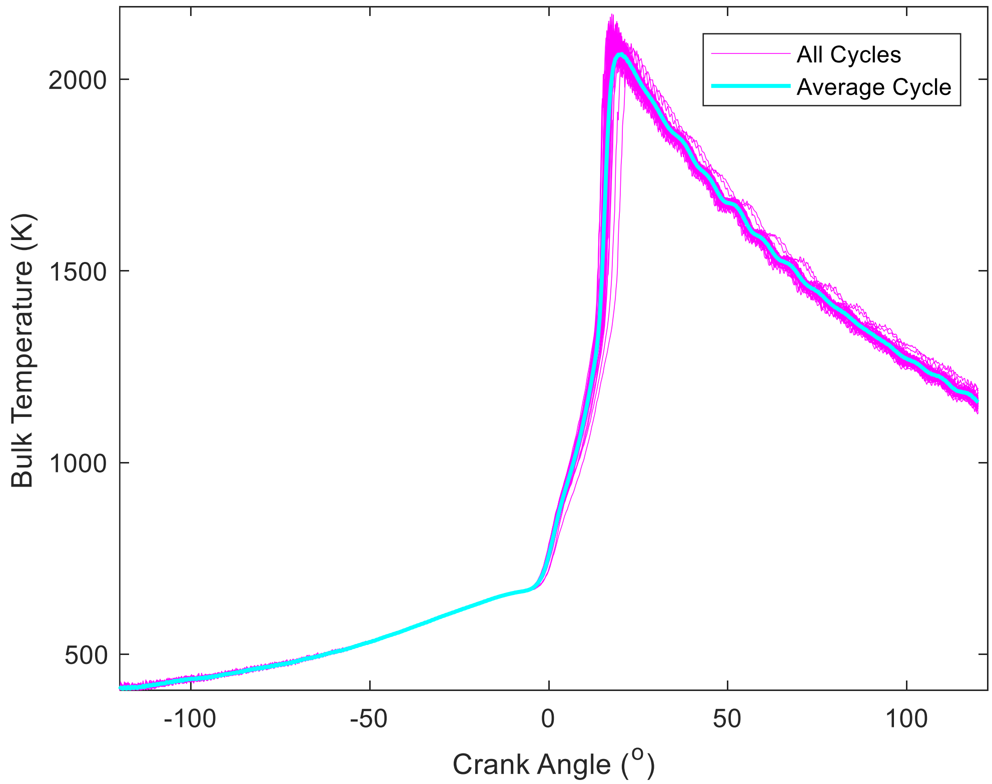

Figure 8b. Bulk Temperature vs Crank Angle

Figures 8a and 8b shows that the no-knock cases reject more heat to the exhaust as the temperatures are higher during the power stroke and compression strokes. The temperature at -120° for the no-knock case is about 50°C hotter than the knock case. The reason for that can be further investigated by analyzing heat transfer losses and heat release rates.

In order to find the heat transfer losses, h must first be calculated. In order to do so equation 4 is used:

$$h = 3.26 * B^{-0.2} * p^{0.8} * T_{bulk}^{-0.55} * w^{0.8}$$ (Lawler 2018) (4)

Where:

B=Bore of piston (m^3)

p=Pressure in cylinder (kJ)

T_{bulk}=Bulk Temperature in cylinder (K)

w= Mean gas velocity (m/s)

The h values calculated can be seen in figure 9:

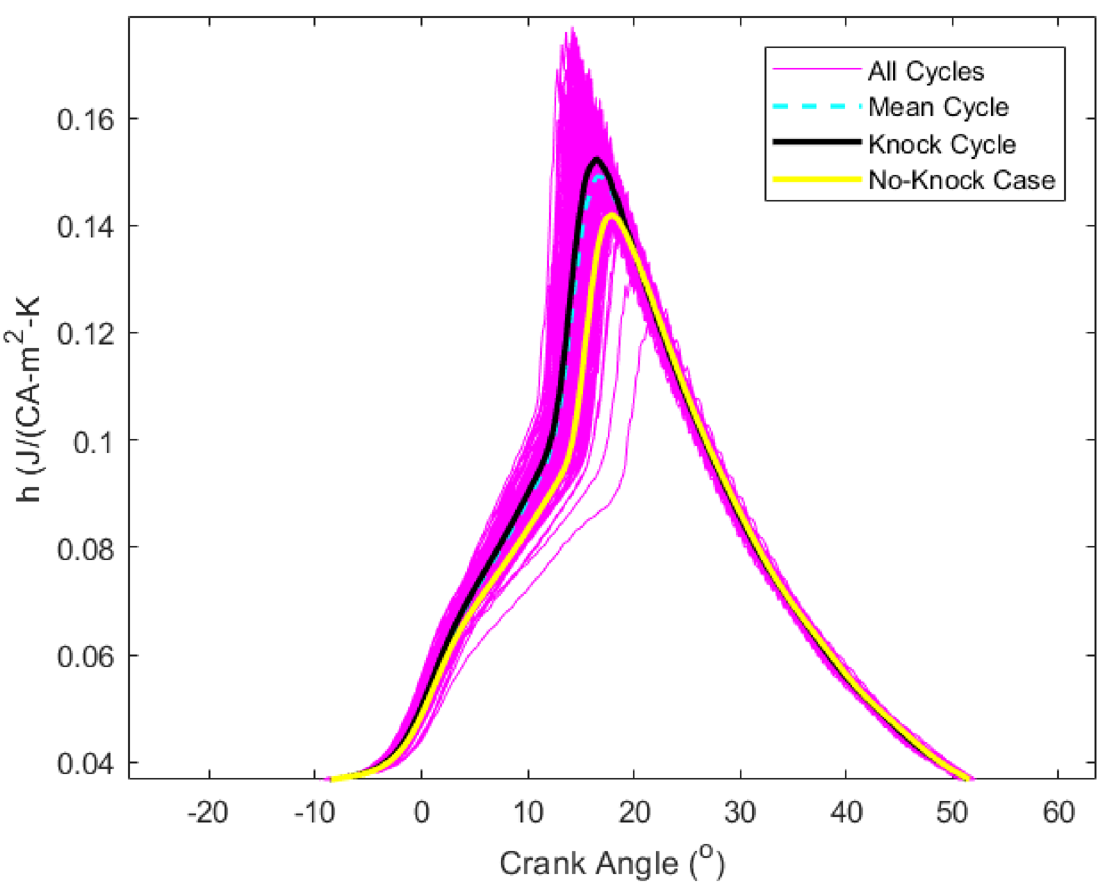

Figure 9: Heat transfer coefficient h in terms of Crank Angle vs Crank Angle

Figure 9 shows the different heat transfer coefficients which again resembles the pressure traces. However, we see again that the no-knock cases are slightly above the knock cases, indicating that there is potentially less heat transfer losses during expansion.

Figures 10a and 10b will look at the knock and no-knock cases respectively to see if any differences can be observed.

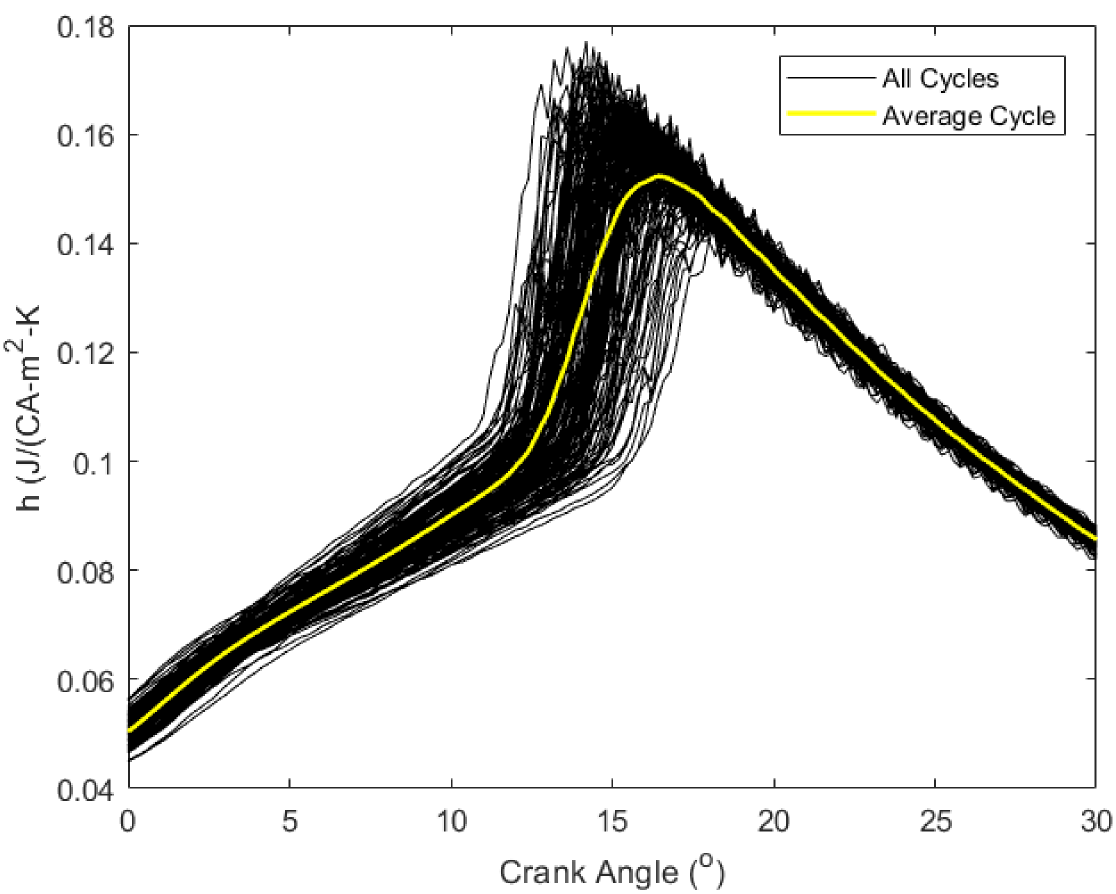

Figure 10a. h vs CA for knock cases

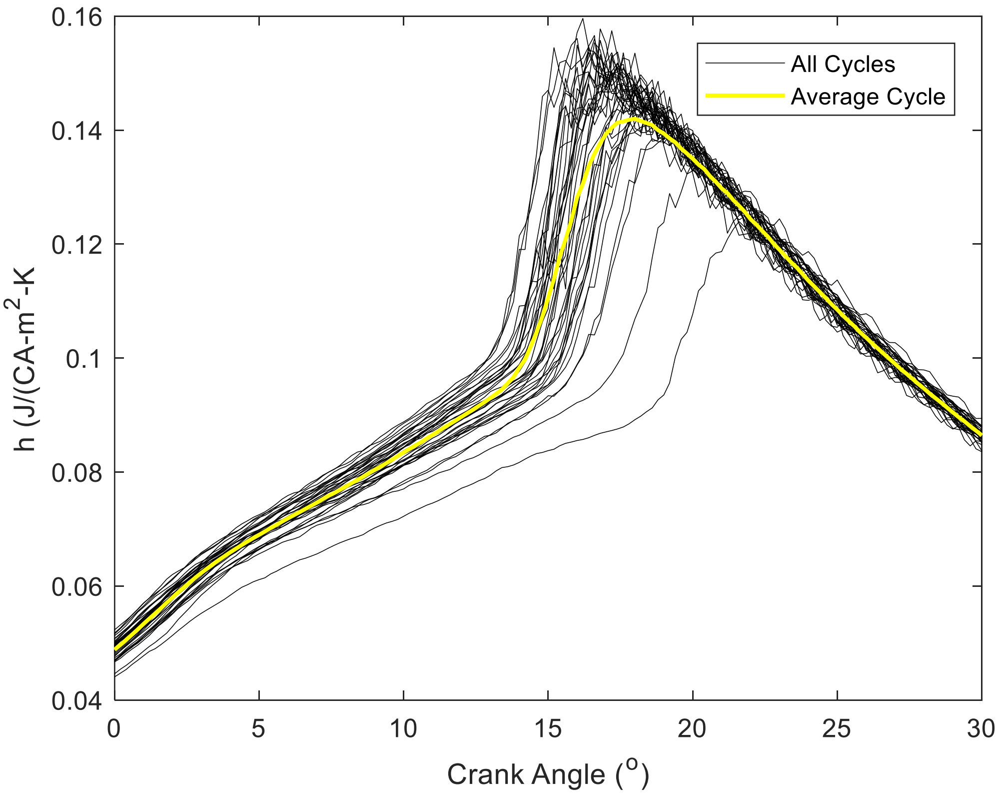

Figure 10b. h vs CA for no-knock cases

Figures 10a and 10b shows that the peak heat transfer coefficient is lower for the no-knock cases. However, after the peak value subsides, the value for h is greater for the no-knock cases. Albeit a low difference, after multiplying by the difference in temperature between the bulk and the wall and integrating over 240 degrees, that small difference will begin to add up and results in losses. The heat loss rate to the walls can be seen in Figure 11 which was calculated using equation:

$$Heat\ Loss = h * Area * (T_{bulk} - T_{wall}) \quad \text{(Lawler 2018) (5)}$$

Where:

h=Heat transfer coefficient (W/m^2-K)

Area=Surface area of heat transfer (m^2)

T_{bulk}=Bulk Temperature (K)

T_{wall}=Temperature of wall (K)

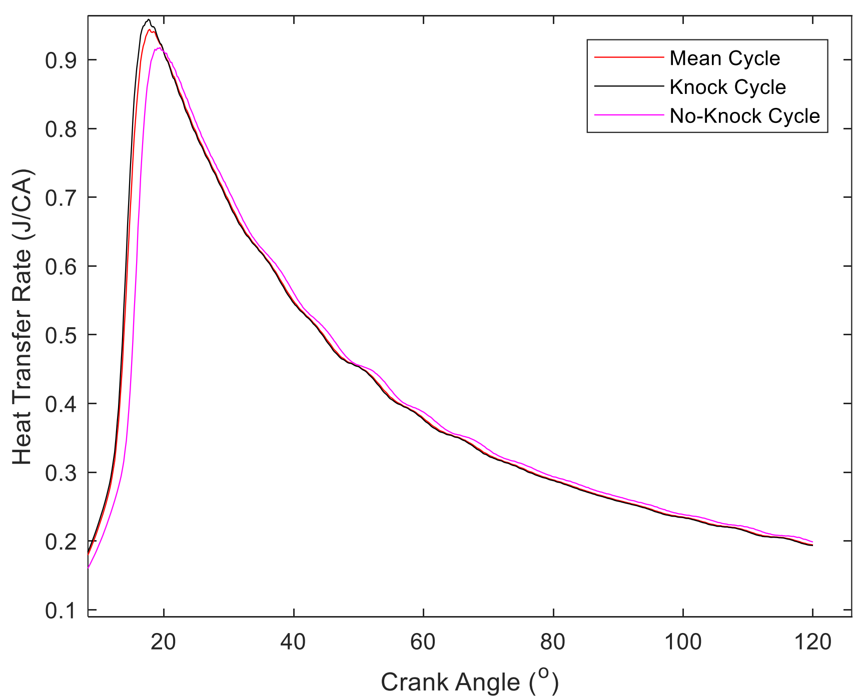

Figure 11. Heat Transfer Rate vs Crank Angle

Figure 11 shows the heat transfer loss rate to the cylinder wall. Because the bulk temperature is higher for the no knock cycle during expansion and the h value is higher, it makes sense that the rate is also higher during expansion. However, Unlike in the previous graphs, upon compression, the heat transfer rates are lower than that of the knock cycles. The width of the graph is slightly greater which is to be expected due to the violent nature of auto ignition which lends itself to rapid losses at peak pressures.

Once the rate is calculated, the total heat loss to the walls can be found by integrating over a given crank angle domain. For this analysis, the range was chosen to be -120 to 120 degrees. The graph can be seen in figure 12:

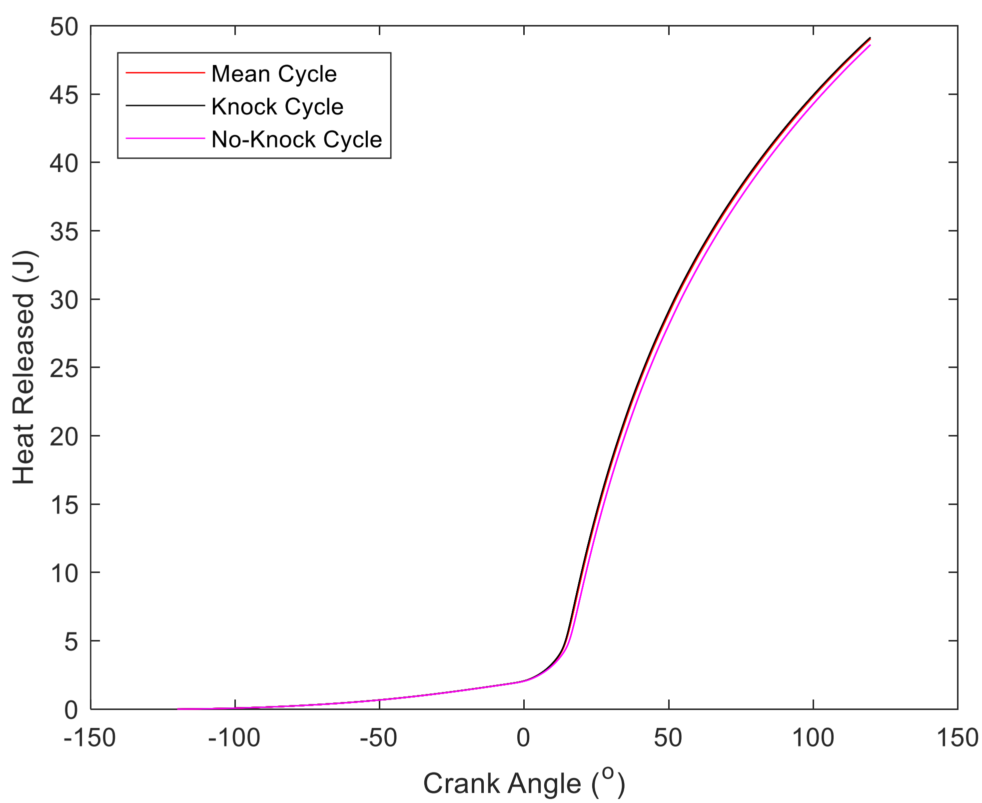

Figure 12. Heat Transferred Loss to Wall vs Crank Angle

Figure 12 shows that the heat transfer loss for the knock case is greater although slightly. The slightly greater rates during expansion were not great enough such that the knock cases rejected less heat. The final value for the difference between them is about 0.54 J.

After finding the heat transfer losses to the wall, the heat release rate was calculated for each condition. The release rates were calculated using equation:

$$\left[\frac{J}{CA}\right] = \frac{\gamma * pdV}{\gamma - 1} + \frac{Vdp}{\gamma - 1}$$

(Lawler 2018) (6)

Where:

Y=Ratio of specific heats

p=Pressure inside cylinder

dV=Derivative of V at point i

dp=Derivative of pressure at point i

Calculating the release rate shows the thermal width of the cycle and can be used to predict heat rejected to the exhaust. The graph can be seen in figure 13 on the next page:

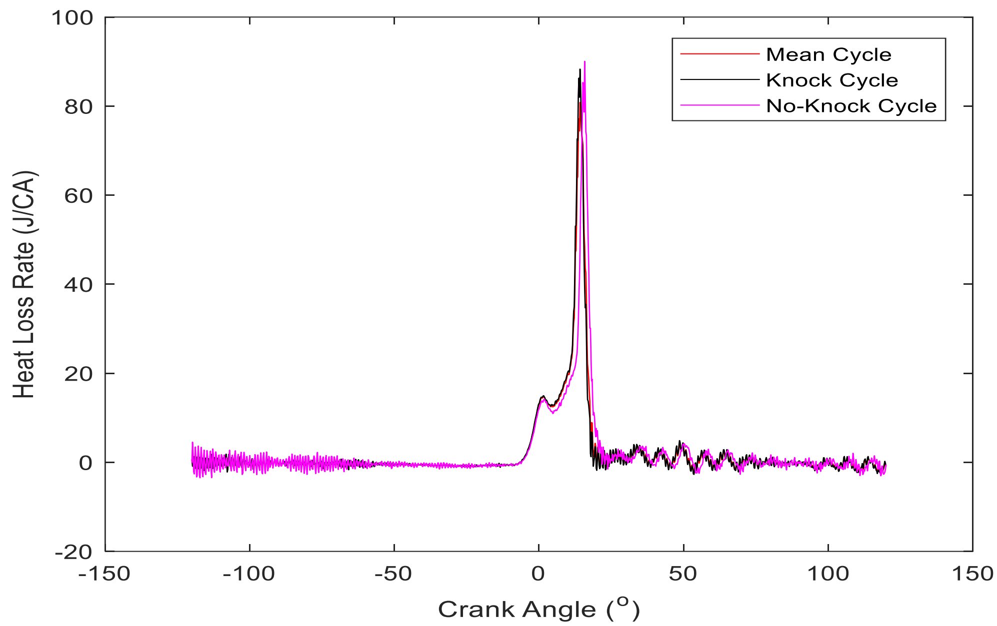

Figure 13. Heat Release Rate vs Crank Angle

Figure 13 shows the heat release rates of the knock and no-knock cycles along with the mean. We see that around IVC there is a large ringing in the no-knock cycle rates. This could be because there aren't enough cycles to get a meaningful average and the fluctuations cause the rates to fluctuate because of how sensitive it is to the derivative terms. Derivative makes noisy signals more noise (Lawler 2018). The results from Zavlava and Folkerts (2011) shows high ringing in the no-knock cases as well just as in this graph. However, the heat release rates in this data shows that the no-knock cycles have a higher peak rate than the knock-cycles. The reason for this could be the high variance in the knock rates as it would have a higher propensity to have more static in its values due to ringing and the derivative term in a noisy and ringing signal. This variance can be sees in figures 14a and 14b on the next two pages:

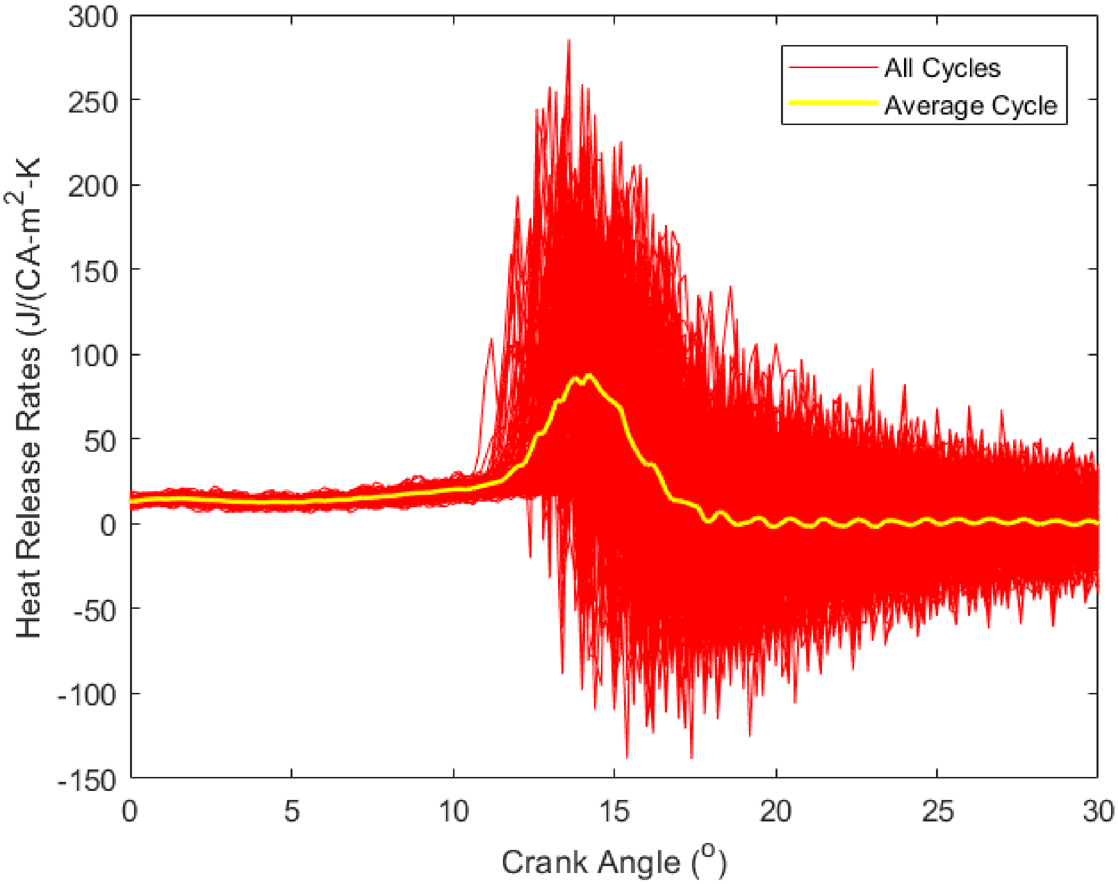

Figure 14a: Heat Release Rates vs Crank Angle for knock cases

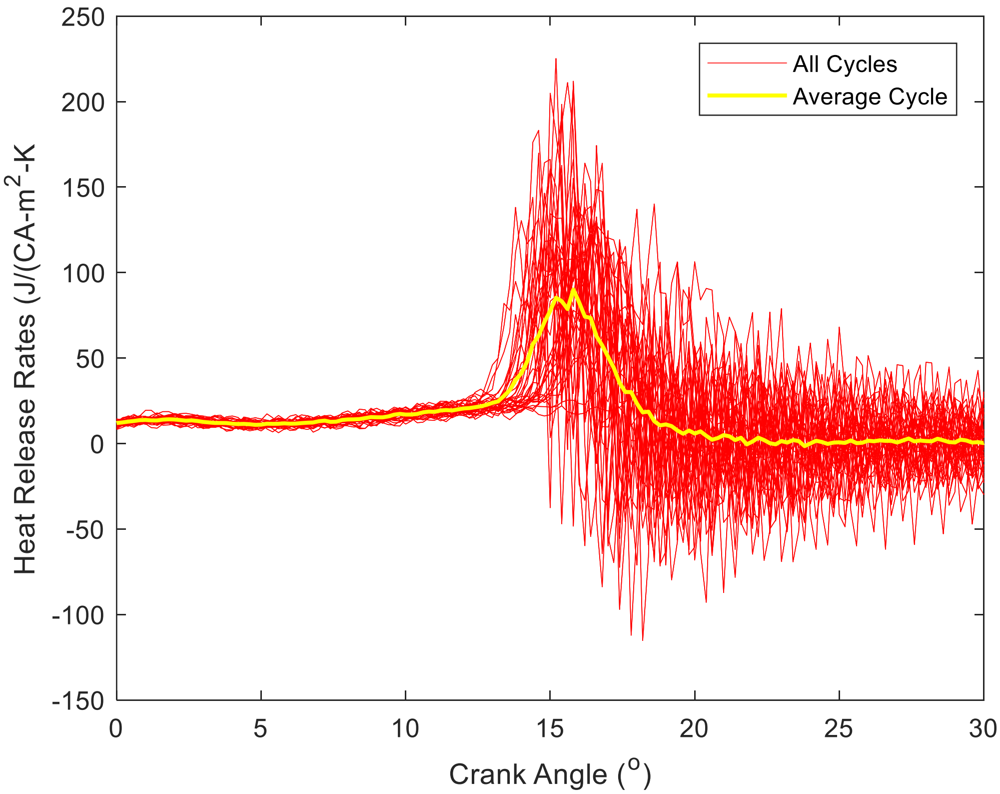

Figure 14b: Heat Release Rates vs Crank Angle for no-knock cases

Figures 14a and 14b show that the reason for the slightly higher release rates for the knock cycle could possibly be due to the high variance in the knock cases. The high static later on in the combustion leads to more negative values. At their peaks, the highest heat release rate for the knock case is 60.47 J/CA higher than for the no-knock cases which is a significant difference and is more than half the average peak values. A cycle with more no-knock cases would be better suited to analyze the difference between the release rates.

In order to show the high variance, the coefficient of variation for the release rate from 0 to 30 degrees ATDC was plotted against crank angle shown in figures 15 on the next page:

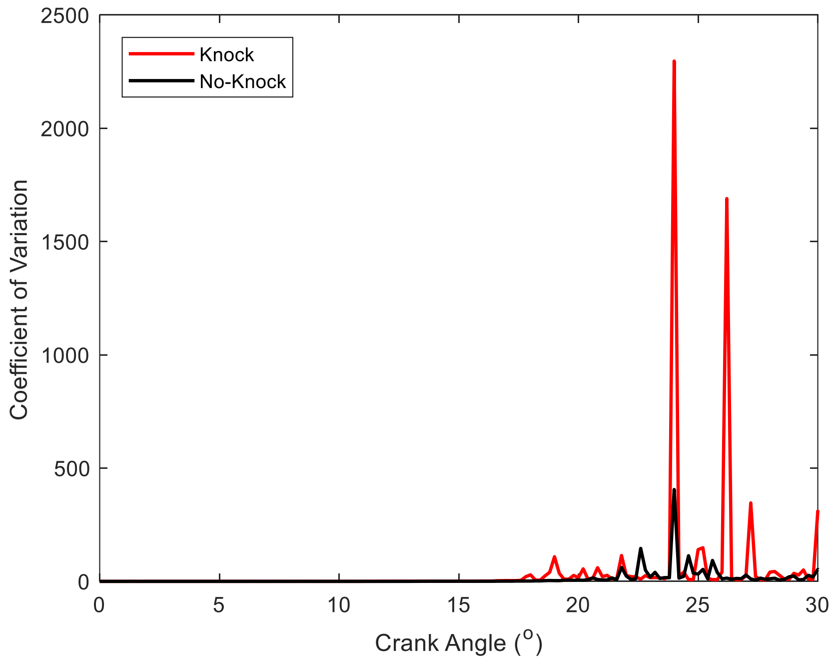

Figure 15. Coefficient of Variation vs Crank Angle for both knock and no-knock cases

The graph shows the magnitude of the difference in variability during combustion between knock and no-knock cycles. The COV for the knock cycle is 55.05 times greater than the no-knock cycle at 24° ATDC. The no knock-cycle has high variations at higher points but those are at very low pressures which contribute negligible amounts to the total heat release rate and total heat loss. This can be seen in figure 16 on the next page:

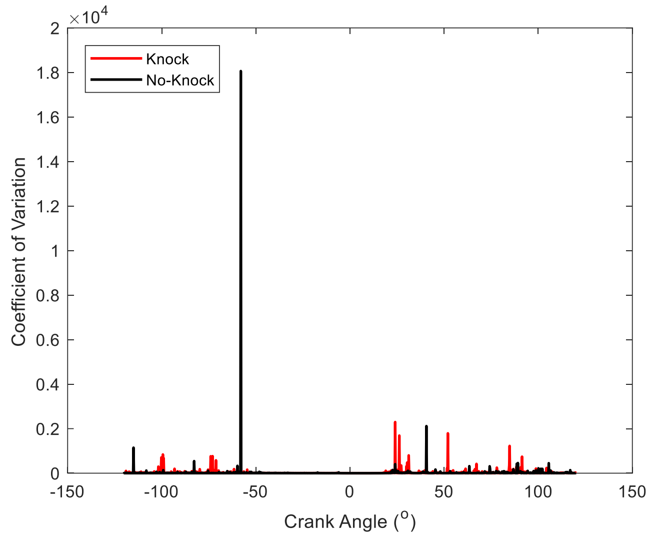

Figure 16. Coefficient of Variation vs Crank Angle for both knock and no-knock cases

The highest variance occurs at around 58° BTDC. There are no known significant events at that crank angle degree so the cause for this large variance could not be determined. Perhaps the mean is very small here so a moderate variance produces a very high variance.

The Mass Fraction Burned curve is an important parameter as it can be used to determine the CA50 and burn times of the air fuel mixture in the cylinder. In order to obtain it, the heat release rates must be integrated over the desired crank angle range to obtain the total heat release. Next, adding in heat transfer losses calculated earlier, the net apparent heat release can be calculated. After normalizing by the max values, the MFB is obtained. The MFB for the different conditions can be seen in figure 17 on the next page:

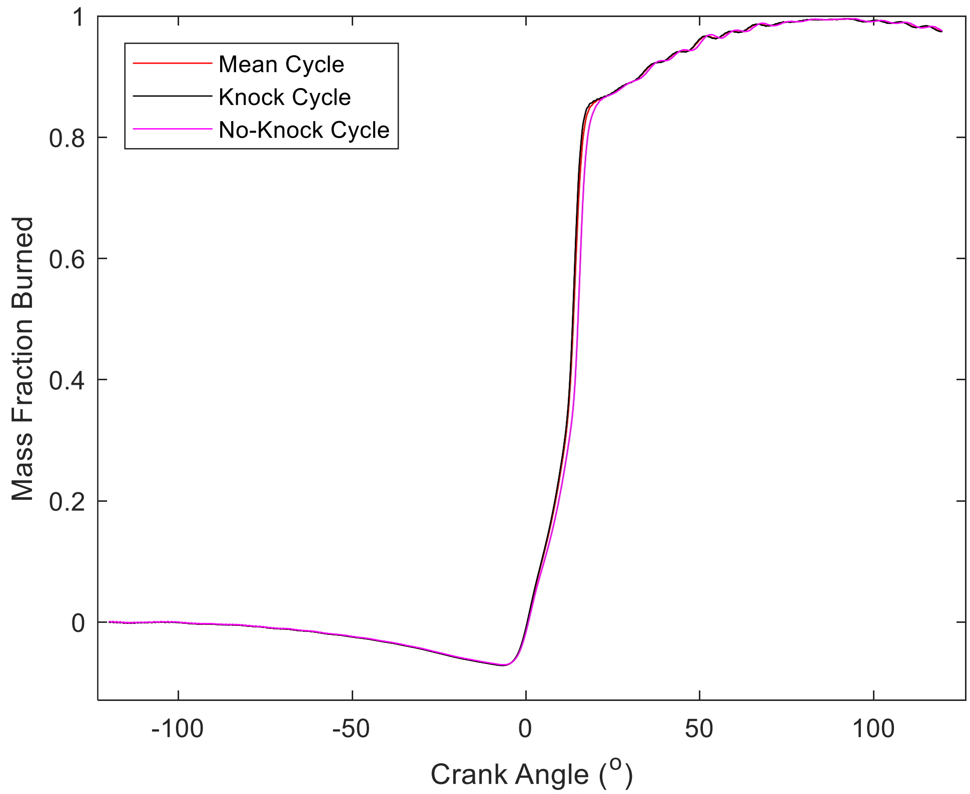

Figure 17. Average Mass Fraction Burned curves for each condition

The mass fraction burn curve shows that, although slight, the average CA50 for the no-knock cases is slightly later than the knock-cases which agrees with theory. The high temperatures and pressures present during knock causes the fuel to burn faster. Because integrating has a filtering effect, the individual cycles will not be graphed as it the variance of the knock case cannot be seen. However, the total heat released by the two conditions is of interest and can be seen in figure 18 on the next page:

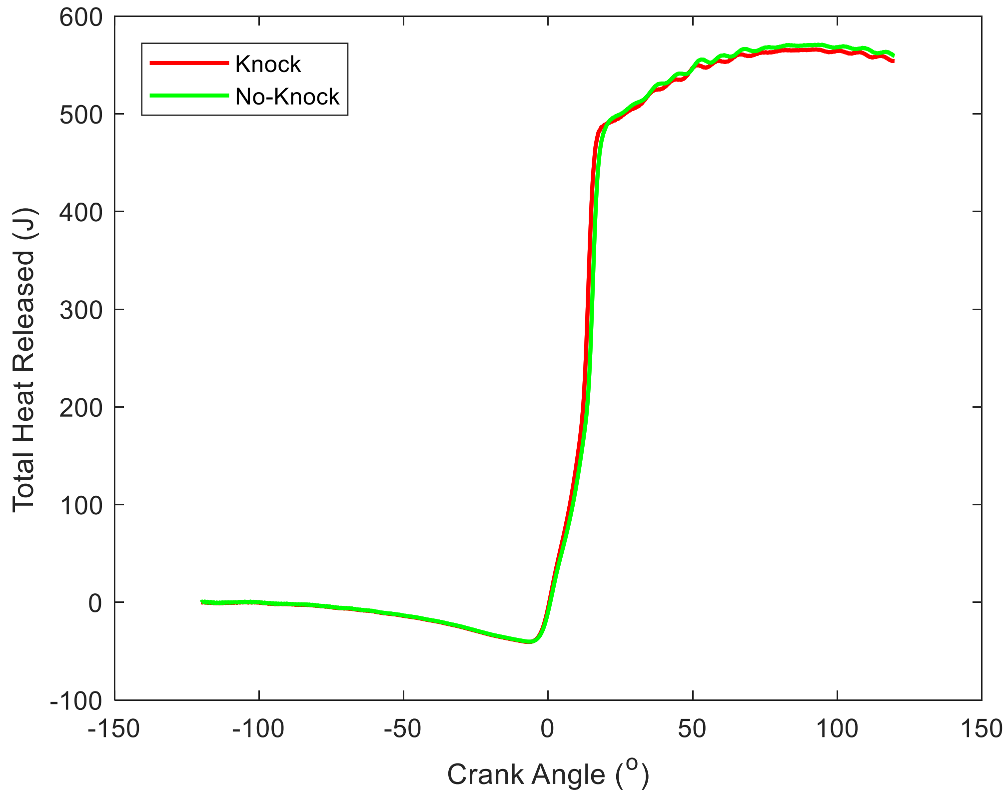

Figure 18. Total Heat Released vs Crank Angle for both knock and no-knock cases

Figure 18 shows that the no-knock cycle releases 5.32 more J of heat by 120° ATDC. The slower burn rate of the no-knock cycles can also be seen as it leads the knock case throughout the flame propagation, indicating a less steep slope. The rate of change of the knocking case starts to decrease to the flat shape around 4° earlier than the no knock case.

The isentropic unburned shows the maximum temperature of the unburned gases inside the cylinder chamber and can be calculated using equation:

$$T_{isen,unburned} = T_{IVC} * \left(\frac{P_{cyl}}{P_{IVC}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}$$

(Lawler 2018) (7)

Where:

T_{IVC}=Bulk Temperature at Internal Valve Close

P_{cyl}=Pressure at point i

P_{IVC}=Pressure at Internal Valve Close

Y=Ratio of specific heats

The calculated values were then graphed, showed in figure 19 below:

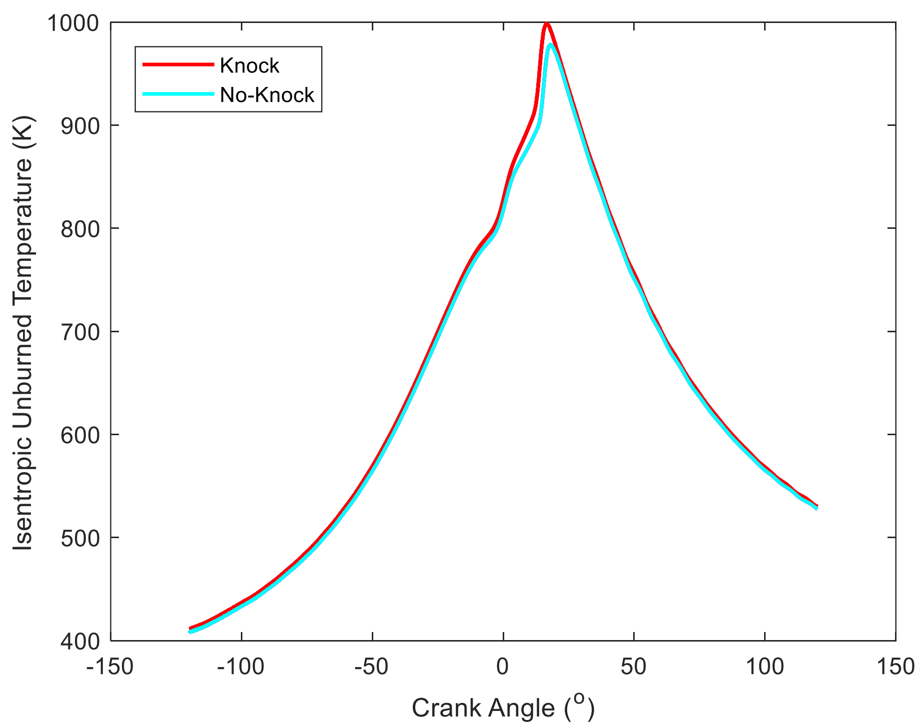

Figure 19. Isentropic Unburned Temperature vs Crank Angle for knock and no-knock cycles

Figure 19 shows that the maximum isentropic temperatures for the knock case is slightly higher at 20.47 K. The auto-ignition and overall higher bulk temperature during combustion leads to higher possible temperatures for the gasses. Unlike with the bulk temperature, the isentropic unburned temperatures are consistently higher for the knock cases than they are for the no-knock cases.

While thermal properties are important as it tells indicates how an engine produces work and where the losses are, in the end efficiencies are the top priority in accordance with emissions. If efficiencies are higher, than engineers will find a way to operate at whatever conditions necessary. The net and gross work done was done by integrating the pressure trace with respect to volume. The plot for the efficiencies can be seen in figure 20 and table 1 below:

Efficiencies			
	Mean	Knock	No-Knock
Indicated Net Thermal	28.59221	28.58865	28.60789
Indicated Gross Thermal	31.9012	31.89527	31.92733
Mehcanical	53.34578	53.3557	53.30214
Net Fuel Conversion	28.15997	28.15647	28.17542

Table 1. Efficiencies of each condition

Table 1 shows the values used to make figure 20 on the next page.

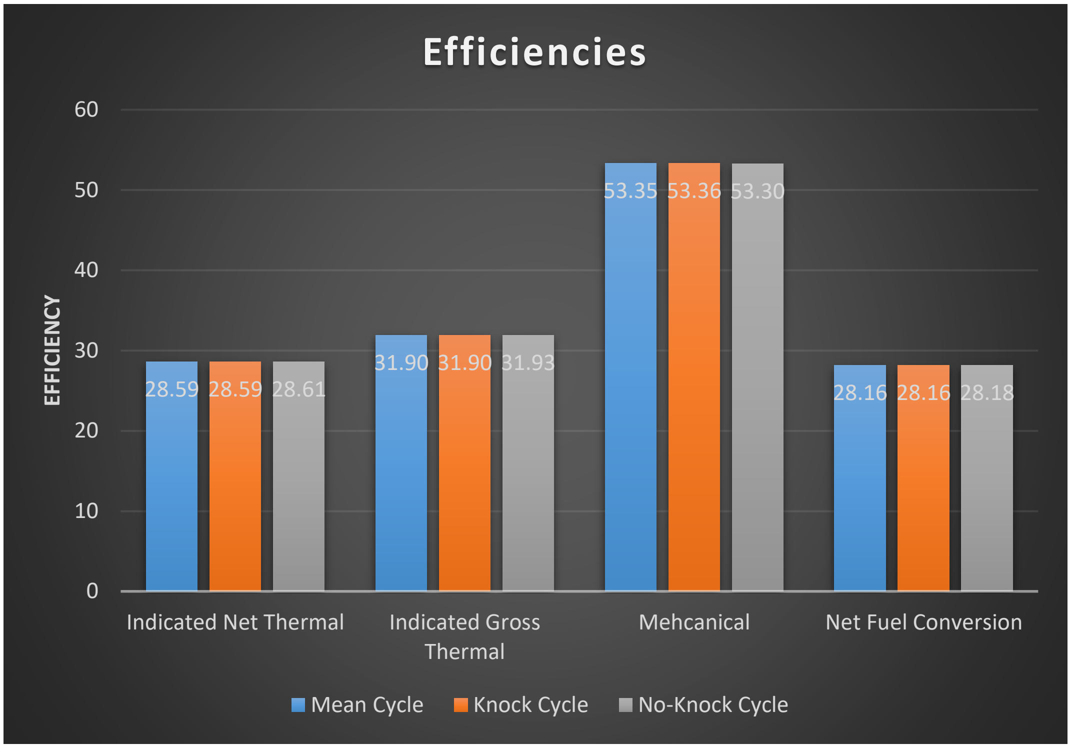

Figure 20. Efficiencies of each condition

Figure 20 shows that, for this analysis, the efficiencies are essentially the same. The difference between them is almost negligible. The results from figure 20 and table one agree with the specific fuel consumption values that can be seen in figure 21 and table 2 below:

Specific Fuel Consumptions			
	Mean	Knock	No-Knock
Net	296.2551	296.292	296.0927
Gross	265.5257	265.575	265.3084

Table 2. Specific Fuel Consumptions for each condition

Table 2 shows the calculated SFC values used to make figure 21 on the next page:

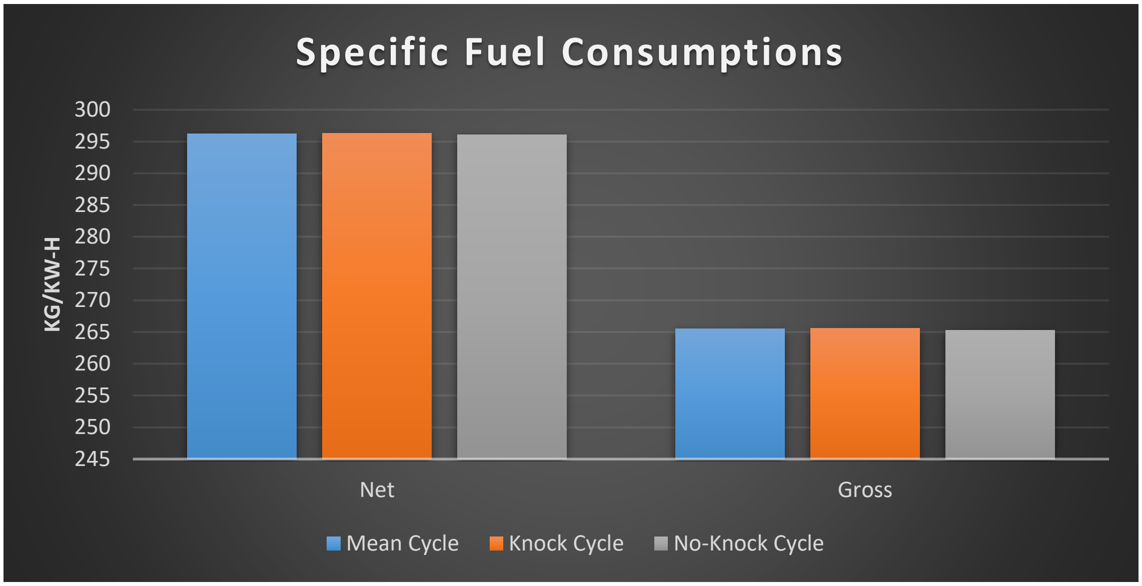

Figure 21. SFC for each condition

Figure 21 shows the amount of fuel needed to produce a kW-h of energy. Again, the values differ by too little to notice by eye so figures 22a and 22b will show the percent deviations of the knock and no-knock cycles from the mean cycle.

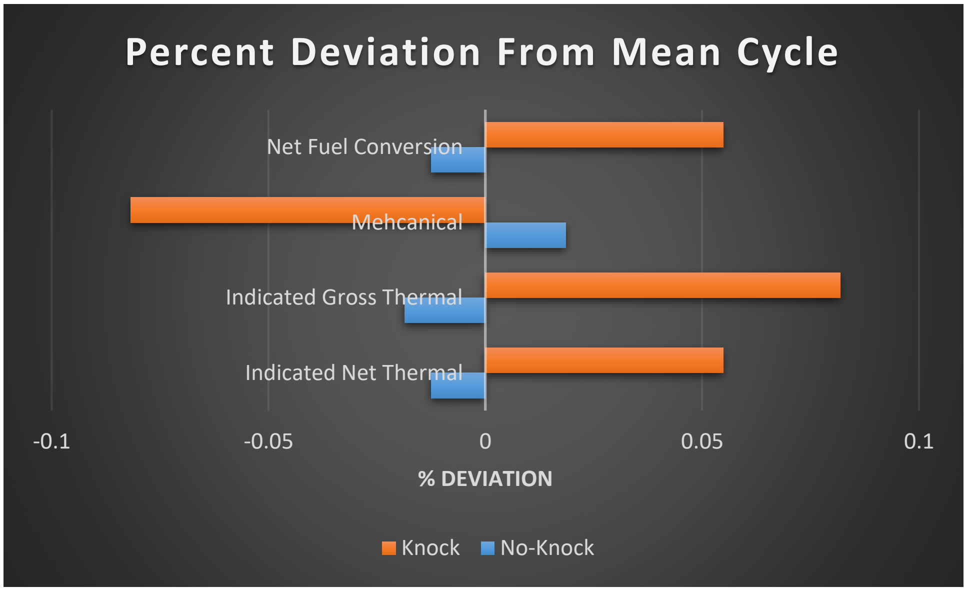

Figure 22a. Percent deviation from the mean cycle efficiency

Figure 22a shows that although small, there is a noticeable trend where the no-knock cycles are more efficient which agrees with intuition and theory. The gross efficiency seems to be the highest positive deviation for the no-knock case. This makes sense as this is where the knock-cases were bleeding off most of the energy while ringing. So, when the no-knock cases are isolated, they should come out at higher gross efficiencies. The mechanical efficiency is lower for the no-knock cases. This makes sense if the gross work is lower for the knocking cycle as mechanical efficiency is defined on next page as:

$$\eta_{eff} = \frac{P_b}{P_{i,g}} \tag{8}$$

Where:

P_b=Indicated Brake Power

$P_{i,g}$=Indicated Gross Power

The bigger denominator term will lead to a lower efficiency.

Figure 22b shows the fuel consumption percent deviations on the below:

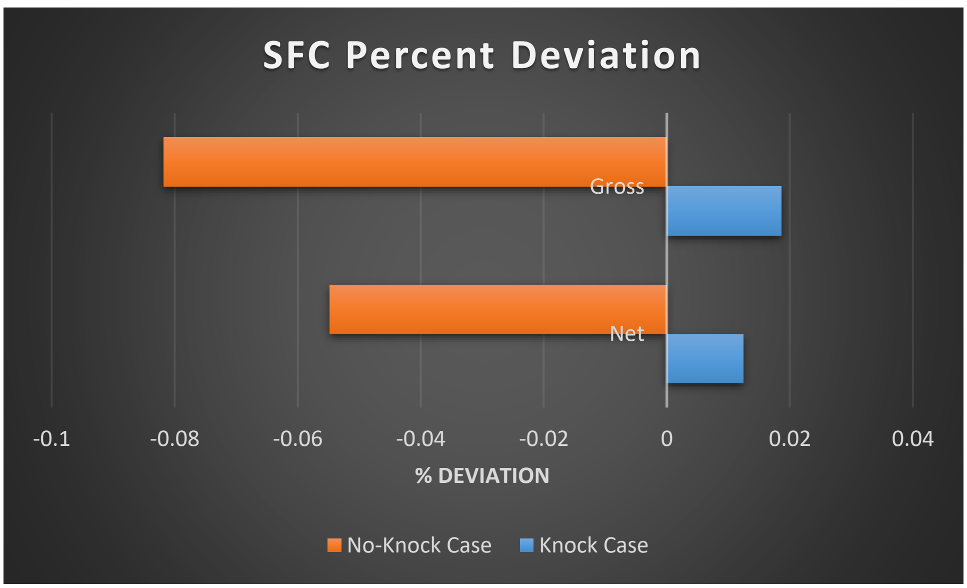

Figure 22b. SFC percent deviations from mean cycle

Figure 22b shows that the non-knocking cycles require less fuel than the knocking cycles albeit a very small difference. This shows that knocking will inevitably lead to lower fuel economy as more fuel is needed to produce the same amount of energy.

Conclusion

The analysis showed that, on average, the peak temperatures and pressures for the knocking cycles were indeed higher than that of the cycles that weren't knocking as expected. These higher pressures and hotter temperatures lead to a narrower gap of heat release. The ringing in the pressures caused high variances in heat release during combustion which is unfavorable as engines cycles should be relatively repeatable because they are design to operate under certain conditions. If the variance is too high, the engine will be operating in sub optimal or even potentially dangerous zones. The efficiency differences between the knocking and no-knocking cases were small but showed a trend in which the cases that don't knock were consistently more efficient. The sample size was fairly low for the cases that didn't knock and could have been skewed by outliers as a large sample size has the benefit of absorbing outliers and mitigating their effect. In closing, it can be seen that engines operating at knock are less efficient and highly chaotic and as such, should be avoided despite the higher peak pressures available to do work.

References

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Appendix

```
clear all

clc

[CRMotored,~,~,~] =

convertTDMS(false,'20181016_SI_CRsweep_01.tdms');

[CRData3,~,~,~] =

convertTDMS(false,'20181016_SI_CRsweep_04.tdms');%10


%% Preliminary

clearvars -except CRMotored CRData3

clc

l=10*0.0254;

B=3.25*0.0254;

L=4.5*0.0254;

a=2.25*0.0254;

Vd=pi/4*B^2*L;


CRUMP=CRMotored.Data.MeasuredData(3648).Data;

CRUL3=CRData3.Data.MeasuredData(3648).Data;


d=0;
```

```
for j=1:200; %200 cycles
for i=1:3600; %span the entire length of the vector
    d=d+1;
    CRSM(i,j)=CRUMP(d);
    CRS3(i,j)=CRUL3(d);
end
end


for i=1:3600
   MCRSM(i)=mean(CRSM(i,:));
      MCRS3(i)=mean(CRS3(i,:));
end


peak=max(MCRSM); %Finding highest pressure value
loc=0;
for i=1:3600;
    id=MCRSM(i); %compares every M element to its peak
value
    if id==peak; %peak value is found
        loc=i %locates index of highest pressure value
        break
```

```
    end

end

CA=[0.2:0.2:720];

TLA = -1.4;

NCA = CA-(CA(loc)-(TLA)); %Professor Lawler

loc=loc-(TLA)/0.2;

IVC=loc-135/0.2;


for i=1:3600

s(i)=a*cos(NCA(i)*pi/180)+(l^2-

a^2*sin(NCA(i)*pi/180)^2)^(1/2);

end


VcM=Vd/7;

Vc=Vd/10;



VM=VcM+(pi*B^2/4)*(l+a-s);

V=Vc+(pi*B^2/4)*(l+a-s);



IP3=mean(CRData3.Data.MeasuredData(3624).Data)*0.01;
```

```
IPM=mean(CRMotored.Data.MeasuredData(3624).Data)*0.01;
for i=1:200
    BPM(i)=mean(CRSM(loc-925:loc-875,i)); %900 indices=180
degrees which is BDC
    BP3(i)=mean(CRS3(loc-925:loc-875,i)); %900 indices=180
degrees which is BDC
end



for i=1:200
    CRSM(:,i)=CRSM(:,i)+(IPM-BPM(i));
    CRS3(:,i)=CRS3(:,i)+(IP3-BP3(i));
end
for i=1:3600
   MCRSM(i)=mean(CRSM(i,:));
      MCRS3(i)=mean(CRS3(i,:));
end

S=loc-120/0.2;
E=loc+120/0.2;
BDC=loc-180/0.2;
```

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```
plot(NCA,CRS3,'-k','Linewidth',0.5)
xlabel('Crank Angle (^{o})')
ylabel('Pressure (bar)')
title('Pressure vs Crank Angle')
hold on
plot(NCA,MCRS3,'-r','LineWidth',2)


SP3=L*mean(CRData3.Data.MeasuredData(3625).Data)/30;

d=0;


for i=S+1:loc-10  %Pressures should be about the same
before combustion
    d=d+1;
I3d(d)=MCRS3(i);
end
j=0;
for k=0:0.01:5 %k will range from 0 to 5 to ensure a
correct value is chosen
    d=0;
```

```
    j=j+1;
for i=S+1:loc-10
    d=d+1;
    Psm3(d)= MCRS3(IVC)*(V(IVC)/V(i))^k;
end
E3(j)=mean(abs((I3d-Psm3)./I3d));
end

best3=min(E3); %Lowest error will be used to create
simulated pressures


CAsim=NCA(loc-120/0.2:loc+120/0.2);

for i=1:length(E3)
    verif3=E3(i);
    if verif3==best3
        verif3=i
        break
    end
end
k3=(verif3-1)*0.01;
```

```
%simulated pressures
d=0;
    for i=loc-120/0.2:loc+120/0.2
    d=d+1;
    Psim3(d)= MCRS3(IVC)*(V(IVC)/V(i))^k3;
    end


figure()
plot(CAsim,Psim3,'-r',CAsim,MCRS3(loc-
120/0.2:loc+120/0.2),'-g')
xlabel('Crank Angle (^{o})')
ylabel('Pressure (bar)')
title('Pressure vs Crank Angle for CR=10')
close all


d=0;
C1=2.28;
C2=0.00324;
Tref3=mean(CRData3.Data.MeasuredData(3617).Data);
```

```
R=0.288;


SP=L*mean(CRData3.Data.MeasuredData(3625).Data)/30

for j=1:200

    d=0;

    for i=S:E

        d=d+1;

        w3s(d,j)=C1*SP+C2*Vd*Tref3*(CRS3(i,j)-
Psim3(d))/(CRS3(BDC,j)*V(BDC));

    end

end


     w3sm=MeanPressures(w3s);




figure() %CR=10

    plot(CAsim,w3s,'-g','LineWidth',0.5)

    hold on

    plot(CAsim,w3sm,'-r','LineWidth',2)

    xlabel('Crank Angle (^{o})')

    ylabel('Velocity (m/s)')
```

```
    title('Gas Velocity vs Crank Angle for CR=10')



AFratio3=mean(CRData3.Data.MeasuredData(3630).Data);
mass=mean(CRData3.Data.MeasuredData(3628).Data)*(1+1/AFrati
o3)*0.001;
mfuel=mean(CRData3.Data.MeasuredData(3628).Data)/AFratio3*0
.001;
fuelflow=mean(CRData3.Data.MeasuredData(3629).Data)*0.001;
mair=mean(CRData3.Data.MeasuredData(3628).Data)*0.001;
Qlhv=42500000;
Supply=Qlhv*fuelflow; %supply is dQ/dt


V_int=mass*R*Tref3/MCRS3(IVC)*0.01;


Vexh=V(IVC)-V_int;



mass_res=(MCRS3(IVC)*Vexh*100)/(R*mean(CRData3.Data.Measure
dData(3614).Data));
```

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```
mass_tot=mass_res+mass;



d=0;
for j=1:200
    d=0;
for i=S:E
    d=d+1;
Tbulk(d,j)=CRS3(i,j)*V(i)*100/(mass_tot*R);
end
end

for j=1:200
    d=0;
    for i=S:E
        d=d+1;
        h(d,j)=3.26*B^(-
0.2)*(CRS3(i,j)*100)^(0.8)*Tbulk(d,j)^(-0.55)*w3s(d,j)^0.8;
    end
end
```

```
d=0;

for i=1:length(h)

    d=d+1;

    hm(d)=mean(h(i,:));

end


figure()

    plot(CAsim,h,'-g','LineWidth',0.5)

 xlabel('Crank Angle (^{o})')

    ylabel('h (W/(m^{2}-K)')

    title('h vs Crank Angle for CR=10')

    hold on

    plot(CAsim,hm,'-r','LineWidth',2)




A=V/(pi/4*B^2)*pi*B; %h=V/A, A=pi*B*h

Twall=400


h=h*1/(1000/60*360); %conversion factors

hm=hm*1/(1000/60*360);
```

```
d=0;

for j=1:200

    d=0;

    for i=S:E

        d=d+1;

       hLoss(d,j)=h(d,j)*A(i)*(Tbulk(d,j)-400);

    end

end

d=0;

hLossm=MeanPressures(hLoss);



figure()

plot(CAsim,hLoss,'-g','LineWidth',0.5)

xlabel('Crank Angle (^{o})')

ylabel('Heat Loss Rate to Wall (J/CA)')

title('Heat Loss to Wall Rate vs Crank Angle for CR=10')

hold on

plot(CAsim,hLossm,'-r','LineWidth',2)


    for j=1:200 %indefinite integral
```

```
        d=0;
        hRelease(1,j)=0;

        for i=2:length(hLoss)-1
    hRelease(i,j)= hRelease(i-
1,j)+0.2*(hLoss(i+1,j)+hLoss(i,j))/2;
        end
    end
hReleasem=MeanPressures(hRelease);



     figure()
    plot(CAsim(1:1200),hRelease,'-g','LineWidth',0.5)
    xlabel('Crank Angle (^{o})')
    ylabel('Heat Lost (J)')
    title('Heat Lost to Wall for CR=10')
    hold on
    plot(CAsim(1:1200),hReleasem,'-r','LineWidth',2)

d=0;
g=1.33; %heat release rates
for j=1:200
```

```
    d=0;
  for  i=S:E
    d=d+1;
    Loss(d,j)=(g*100000*CRS3(i,j)*((V(i+1)-V(i-
1))/0.4))/(g-1)+(V(i)*100000*((CRS3(i+1,j)-CRS3(i-
1,j))/0.4))/(g-1);
  end
end
Lossm=MeanPressures(Loss);

figure()
plot(CAsim,Loss,'-g','LineWidth',0.5)
xlabel('Crank Angle (^{o})')
ylabel('Heat Release Rate (J/CA)')
title('Heat Release Rate vs Crank Angle')
hold on
plot(CAsim,Lossm,'-r','LineWidth',2)

figure() %Verifying that COV is greatest after combustion
for i=1:length(Loss)
    COVLoss(i)=abs(std(Loss(i,:))/mean(Loss(i,:)));
```

```
end

plot(CAsim,COVLoss,'-r')

xlabel('Crank Angle (^{o})')

ylabel('Coefficent of Variation')

title('Coefficient of Variation vs Crank Angle')




for j=1:200 %Heat released

    Release(1,j)=0;

  for i=2:length(Loss)-1

      Release(i,j)= Release(i-
1,j)+0.2*(Loss(i,j)+Loss(i+1,j))/2;

  end

end


Releasem=MeanPressures(Release);


TotalHeat=Release+hRelease; %total heat released

TotalHeatm=MeanPressures(TotalHeat);

figure()

plot(CAsim(1:1200),TotalHeat,'-g','LineWidth',0.5)
```

```
xlabel('Crank Angle (^{o})')

ylabel('Heat Released (J)')

title('Net Heat Released vs Crank Angle')

hold on

plot(CAsim(1:1200),TotalHeatm,'-r','LineWidth',2)

for j=1:200 %normalize by max to creat MFB Curve

MFB(:,j)=TotalHeat(:,j)/max(TotalHeat(:,j));

end




MFBm=MeanPressures(MFB);

figure()

plot(CAsim(1:1200),MFB,'-g','LineWidth',0.5)

xlabel('Crank Angle (^{O})')

ylabel('Mass Fraction Burned')

title('Mass Fraction Burned vs Crank Angle')

hold on

plot(CAsim(1:1200),MFBm,'-r','LineWidth',2)


w=mean(CRData3.Data.MeasuredData(3625).Data)/60; %omega

T=mean(CRData3.Data.MeasuredData(3626).Data); %Torque
```

```
BrakePower=T*2*pi*w;

NetWorkMean=0;
for i=1:3599
    NetWorkMean=NetWorkMean+(V(i+1)-
V(i))*100000*(MCRS3(i)+MCRS3(i+1))/2;
end
GrossWorkMean=0;
for i=loc-900:loc+900
    GrossWorkMean=GrossWorkMean+(V(i+1)-
V(i))*100000*(MCRS3(i)+MCRS3(i+1))/2;
end

INPMean=NetWorkMean*w/2;
IGPMean=GrossWorkMean*w/2;
FPMean=IGPMean-BrakePower;

INEMean=INPMean/Supply*100;
IGEMean=IGPMean/Supply*100;
BEMean=BrakePower/Supply*100;
MEMean=BrakePower/IGPMean*100;
```

```
NFCEMean=NetWorkMean/(Qlhv*mfuel)*100;

MeanEfficiencies=[INEMean IGEMean BEMean MEMean NFCEMean]

BSFC=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/B
rakePower;
NSFC=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/I
NPMean;
GSFC=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/I
GPMean;

MeanSFC=[NSFC GSFC BSFC]
```

```
%% Identifying Knock
close all



for j=1:200
    for i=1:length(MFB)
            H=MFB(i,j);
        if H>0.5
            z(j)=1/(MFB(i,j)-MFB(i-1,j))*(0.5-MFB(i-
1,j))+i-1; %Linear Interpolation
            break
        end
    end
end

for i=1:length(z)
CA50(i)=(loc-600+z(i))*0.2-360; %Converting to CA
end
```

```
min50=min(CA50)

max50=max(CA50)

for i=1:200

    idB=CA50(i);

    if idB==max50;

        Rb=i; %R is the most retarded case

        break

    end

end

for i=1:200

    idS=CA50(i);

    if idS==min50;

        Ra=i; %Ra is the advanced case

        break

    end

end

figure()

plot(NCA(loc-30/0.2:E),CRS3(loc-30/0.2:E,Ra),'-r',NCA(loc-
30/0.2:E),CRS3(loc-30/0.2:E,Rb),'-k')

xlabel('Crank Angle (^{o})')

ylabel('Pressure (bar)')
```

```
title('Pressure vs Crank Angle for Most Retarded and
Advanced Cases')
legend('Advanced Case','Retarded Case')


for j=1:200
    d=0;
for i=S:E
    d=d+1;
    TI(d,j)=Tbulk(1,j)*(CRS3(i,j)/CRS3(S,j))^((g-1)/g);
%Isentropic Unburened Temps
end
end


for j=1:200
    d=0;
    for i=S:E
        d=d+1;
        tau(d,j)=(1.3*10^(-4))*(CRS3(i,j)*0.986923)^(-
1.05)*20.5^(-1.41)*exp(33700/(1.987*TI(d,j)));
    end
end
```

```
CF=w*360/1000

tau=tau*CF; %convert tau

for j=1:200

    for i=1:length(tau)

        invTau(i,j)=1/(tau(i,j)); %inv tau will be
numerically integrated

    end

end


for j=1:200

    KO(1,j)=0;

   for i=2:length(tau)-1

    KO(i,j)=KO(i-1,j)+0.2*(invTau(i,j)+invTau(i+1,j))/2;
%integrate over CA space

   end

end


plot(CAsim(1:1200),KO,'-r')

xlabel('Crank Angle ^({o})')

ylabel('Integral of 1/\tau)')
```

```
r=1;

q=1;

for j=1:200

    id=max(KO(:,j)); %Find all knocking and not knocking
cases

    if id >= 1

        Knock(r)=j;

        r=r+1;

    else

        Safe(q)=j

        q=q+1;

    end

end

for j=1:length(Knock)

    for i=1:3600

CRKnock(i,j)=CRS3(i,Knock(j)); %Isolating Knock Cycles

    end

end

for j=1:length(Safe)
```

```
        for i=1:3600
        CRSafe(i,j)=CRS3(i,Safe(j)); %Isolating No-Knock Cycles
        end
    end
    plot(NCA,CRSafe,'-r')
    xlabel('Crank Angle (^{o})')
    ylabel('Pressure (bar)')
    title('Non-Knocking Pressures vs Crank Angle')
    xlim([0 35])
    ylim([16 35])



    plot(NCA,CRKnock,'-r')
    xlabel('Crank Angle (^{o})')
    ylabel('Pressure (bar)')
    title('Knocking Pressures vs Crank Angle')
    xlim([0 35])
    ylim([16 35])

    MCRKnock=MeanPressures(CRKnock);
    MCRSafe=MeanPressures(CRSafe);
```

```
%% Knock and No-Knock Cases


% all processes repeated for isolated knock and no knock
cases
d=0;
for j=1:163
    d=0;
for i=S:E
    d=d+1;
TbulkK(d,j)=CRKnock(i,j)*V(i)*100/(mass_tot*R);
end
end


d=0;
for j=1:37
    d=0;
for i=S:E
    d=d+1;
TbulkS(d,j)=CRSafe(i,j)*V(i)*100/(mass_tot*R);
end
end
```

```
d=0;

C1=2.28;

C2=0.00324;

Tref3=mean(CRData3.Data.MeasuredData(3617).Data);

R=0.288;


SP=L*mean(CRData3.Data.MeasuredData(3625).Data)/30

for j=1:163

    d=0;

    for i=S:E

        d=d+1;

        wK(d,j)=C1*SP+C2*Vd*Tref3*(CRKnock(i,j)-
Psim3(d))/(CRKnock(BDC,j)*V(BDC));

    end

end

wKm=MeanPressures(wK);


for j=1:37

    d=0;

    for i=S:E
```

```
        d=d+1;
        wS(d,j)=C1*SP+C2*Vd*Tref3*(CRSafe(i,j)-
Psim3(d))/(CRSafe(BDC,j)*V(BDC));
    end
end
wSm=MeanPressures(wS);

for j=1:163
    d=0;
    for i=S:E
        d=d+1;
        hK(d,j)=3.26*B^(-
0.2)*(CRKnock(i,j)*100)^(0.8)*TbulkK(d,j)^(-
0.55)*wK(d,j)^0.8;
    end
end
hKm=MeanPressures(hK);

for j=1:37
    d=0;
    for i=S:E
```

```
        d=d+1;
        hS(d,j)=3.26*B^(-
0.2)*(CRSafe(i,j)*100)^(0.8)*TbulkS(d,j)^(-
0.55)*wS(d,j)^0.8;
    end
end
hSm=MeanPressures(hS);




A=V/(pi/4*B^2)*pi*B;
Twall=400

hS=hS*1/(1000/60*360);
hSm=hSm*1/(1000/60*360);

hK=hK*1/(1000/60*360);
hKm=hKm*1/(1000/60*360);

d=0;
for j=1:163
    d=0;
```

```
        for i=S:E
            d=d+1;
           hLossK(d,j)=hK(d,j)*A(i)*(TbulkK(d,j)-400);
        end
    end
    d=0;
    hLossKm=MeanPressures(hLossK);


    d=0;
    for j=1:37
        d=0;
        for i=S:E
            d=d+1;
           hLossS(d,j)=hS(d,j)*A(i)*(TbulkS(d,j)-400);
        end
    end
    d=0;
    hLossSm=MeanPressures(hLossS);
```

```
figure()
plot(CAsim,hLossK,'-g','LineWidth',0.5)
xlabel('Crank Angle (^{o})')
ylabel('Heat Loss Rate to Wall (J/CA)')
title('Heat Loss to Wall Rate vs Crank Angle for CR=10')
hold on
plot(CAsim,hLossKm,'-r','LineWidth',2)

    for j=1:163
        d=0;
        hReleaseK(1,j)=0;

        for i=2:length(hLossK)-1
    hReleaseK(i,j)= hReleaseK(i-
1,j)+0.2*(hLossK(i+1,j)+hLossK(i,j))/2;
        end
    end
hReleaseKm=MeanPressures(hReleaseK);

    for j=1:37
        d=0;
```

```
        hReleaseS(1,j)=0;


        for i=2:length(hLossK)-1
    hReleaseS(i,j)= hReleaseS(i-
1,j)+0.2*(hLossS(i+1,j)+hLossS(i,j))/2;
        end
    end
hReleaseSm=MeanPressures(hReleaseS);



     figure()
    plot(CAsim(1:1200),hReleaseS,'-g','LineWidth',0.5)
    xlabel('Crank Angle (^{o})')
    ylabel('Heat Lost (J)')
    title('Heat Lost to Wall for CR=10')
    hold on
    plot(CAsim(1:1200),hReleaseSm,'-r','LineWidth',2)



d=0;
g=1.33;
for j=1:163
```

```
    d=0;
  for  i=S:E
    d=d+1;
    LossK(d,j)=(g*100000*CRKnock(i,j)*((V(i+1)-V(i-
1))/0.4))/(g-1)+(V(i)*100000*((CRKnock(i+1,j)-CRKnock(i-
1,j))/0.4))/(g-1);
  end
end
LossKm=MeanPressures(LossK);

for j=1:37
    d=0;
  for  i=S:E
    d=d+1;
    LossS(d,j)=(g*100000*CRSafe(i,j)*((V(i+1)-V(i-
1))/0.4))/(g-1)+(V(i)*100000*((CRSafe(i+1,j)-CRSafe(i-
1,j))/0.4))/(g-1);
  end
end
LossSm=MeanPressures(LossS);
```

```
figure()

plot(CAsim,LossK,'-g','LineWidth',0.5)

xlabel('Crank Angle (^{o})')

ylabel('Heat Release Rate (J/CA)')

title('Heat Release Rate vs Crank Angle')

hold on

plot(CAsim,LossKm,'-r','LineWidth',2)



for i=1:length(LossK)

    COVLossK(i)=abs(std(LossK(i,:))/mean(LossK(i,:)));

end


for i=1:length(LossS)

    COVLossS(i)=abs(std(LossS(i,:))/mean(LossS(i,:)));

end

figure()

plot(CAsim,COVLossS,'-r')

xlabel('Crank Angle (^{o})')

ylabel('Coefficent of Variation')

title('Coefficient of Variation vs Crank Angle')
```

```
for j=1:163
    ReleaseK(1,j)=0;
  for i=2:length(Loss)-1
      ReleaseK(i,j)= ReleaseK(i-
1,j)+0.2*(LossK(i,j)+LossK(i+1,j))/2;
  end
end

ReleaseKm=MeanPressures(ReleaseK);

for j=1:37
    ReleaseS(1,j)=0;
  for i=2:length(Loss)-1
      ReleaseS(i,j)= ReleaseS(i-
1,j)+0.2*(LossS(i,j)+LossS(i+1,j))/2;
  end
end

ReleaseSm=MeanPressures(ReleaseS);
```

```
TotalHeatK=ReleaseK+hReleaseK;

TotalHeatKm=MeanPressures(TotalHeatK);

TotalHeatS=ReleaseS+hReleaseS;

TotalHeatSm=MeanPressures(TotalHeatS);


figure()

plot(CAsim(1:1200),TotalHeatS,'-g','LineWidth',0.5)

xlabel('Crank Angle (^{o})')

ylabel('Heat Released (J)')

title('Net Heat Released vs Crank Angle')

hold on

plot(CAsim(1:1200),TotalHeatSm,'-r','LineWidth',2)

for j=1:163

MFBK(:,j)=TotalHeatK(:,j)/max(TotalHeatK(:,j));

end


for j=1:37

MFBS(:,j)=TotalHeatS(:,j)/max(TotalHeatS(:,j));

end



MFBKm=MeanPressures(MFBK);
```

```
MFBSm=MeanPressures(MFBS);


figure()

plot(CAsim(1:1200),MFBS,'-g','LineWidth',0.5)

xlabel('Crank Angle (^{O})')

ylabel('Mass Fraction Burned')

title('Mass Fraction Burned vs Crank Angle')

hold on

plot(CAsim(1:1200),MFBSm,'-r','LineWidth',2)


T=mean(CRData3.Data.MeasuredData(3626).Data);

BrakePower=T*2*pi*w;


NetWorkMeanK=0;

for i=1:3599

    NetWorkMeanK=NetWorkMeanK+(V(i+1)-

V(i))*100000*(MCRKnock(i)+MCRKnock(i+1))/2;

end

GrossWorkMeanK=0;

for i=loc-900:loc+900
```

```
    GrossWorkMeanK=GrossWorkMeanK+(V(i+1)-
V(i))*100000*(MCRKnock(i)+MCRKnock(i+1))/2;
end

NetWorkMeanS=0;
for i=1:3599
    NetWorkMeanS=NetWorkMeanS+(V(i+1)-
V(i))*100000*(MCRSafe(i)+MCRSafe(i+1))/2;
end
GrossWorkMeanS=0;

for i=loc-900:loc+900
    GrossWorkMeanS=GrossWorkMeanS+(V(i+1)-
V(i))*100000*(MCRSafe(i)+MCRSafe(i+1))/2;
end

INPK=NetWorkMeanK*w/2;
IGPK=GrossWorkMeanK*w/2;
FPK=IGPK-BrakePower;

INEK=INPK/Supply*100;
IGEK=IGPK/Supply*100;
```

```
BEMean=BrakePower/Supply*100;

MEK=BrakePower/IGPK*100;

NFCEK=NetWorkMeanK/(Qlhv*mfuel)*100;


KnockEfficiencies=[INEK IGEK BEMean MEK NFCEK]


BSFCK=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/
BrakePower;

NSFCK=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/
INPK;

GSFCK=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/
IGPK;


KnockSFC=[NSFCK GSFCK BSFCK]



INPS=NetWorkMeanS*w/2;

IGPS=GrossWorkMeanS*w/2;

FPS=IGPS-BrakePower;


INES=INPS/Supply*100;

IGES=IGPS/Supply*100;
```

```
BEMean=BrakePower/Supply*100;

MES=BrakePower/IGPS*100;

NFCES=NetWorkMeanS/(Qlhv*mfuel)*100;


NonKnockEfficiencies=[INES IGES BEMean MES NFCES]


BSFCS=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/
BrakePower;

NSFCS=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/
INPS;

GSFCS=mean(CRData3.Data.MeasuredData(3629).Data)*3600*1000/
IGPS;


NonKnockSFC=[NSFCS GSFCS BSFCS]


Efficiencies=[MeanEfficiencies' KnockEfficiencies'
NonKnockEfficiencies']


SFC=[MeanSFC' KnockSFC' NonKnockSFC']


close all
```

```
for j=1:163
    d=0;
for i=S:E
    d=d+1;
    TIK(d,j)=TbulkK(1,j)*(CRKnock(i,j)/CRKnock(S,j))^((g-
1)/g);
end
end
TIKm=MeanPressures(TIK);

for j=1:37
    d=0;
for i=S:E
    d=d+1;
    TIS(d,j)=TbulkS(1,j)*(CRSafe(i,j)/CRSafe(S,j))^((g-
1)/g);
end
end

TIm=MeanPressures(TI);
```

```
TISm=MeanPressures(TIS);


plot(CAsim,TIKm,CAsim,TISm,'-r',CAsim,TIm,'-g')

close all


%% Graphs

figure()

p1=plot(NCA,CRS3,'-b','LineWidth',0.3) %PvsCA for all

hold on

p2=plot(NCA,MCRS3,'--r','LineWidth',1.5)

hold on

p3=plot(NCA,MCRSafe,'-k','LineWidth',2)

hold on

p4=plot(NCA,MCRKnock,'-y','LineWidth',2)

hold on

legend([p1(1) p2(1) p3(1) p4(1)],'All cycles','Mean

Cycle','No-Knock Cycle','Knock Cycle')

xlabel('Crank Angle (^{o})')

ylabel('Pressure (bar)')

xlim([5 30])
```

```
figure()
p1=plot(NCA,CRSafe,'-g','LineWidth',0.3) %PvsCA noKnock
hold on
p2=plot(NCA,MCRSafe,'-r','LineWidth',2)
xlabel('Crank Angle (^{o})')
ylabel('Pressure (bar)')
xlim([10 30])
ylim([20 32])
legend([p1(1) p2(1)],'All Cycles','Average Cycle')

figure()
p1=plot(NCA,CRKnock,'-g','LineWidth',0.3) %PvsCA Knock
hold on
p2=plot(NCA,MCRKnock,'-r','LineWidth',2)
xlabel('Crank Angle (^{o})')
ylabel('Pressure (bar)')
xlim([10 30])
ylim([20 35])
legend([p1(1) p2(1)],'All Cycles','Average Cycle')
```

```
figure()

loglog(V,MCRS3,'--r') %PV

hold on

loglog(V,MCRSafe,'-k')

hold on

loglog(V,MCRKnock,'-b')

hold on

legend('Mean Cycle','No-Knock Cycle','Knock Cycle')

xlabel('Volume (m^{3})')

ylabel('Pressure (bar)')


figure()

plot(CAsim,Psim3,'-r',CAsim,MCRS3(loc-

120/0.2:loc+120/0.2),'-g') %Psim

xlabel('Crank Angle (^{o})')

ylabel('Pressure (bar)')

title('Pressure vs Crank Angle for CR=10')


figure()

p1=plot(CAsim,w3s,'-c','LineWidth',0.3) %Gas Velocities

hold on
```

```
p2=plot(CAsim,w3sm,'--r','LineWidth',1.5)
hold on
p3=plot(CAsim,wKm,'-k','LineWidth',2)
hold on
p4=plot(CAsim,wSm,'-y','LineWidth',2)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Gas Velocity (m/s)')
legend([p1(1) p2(1) p3(1) p4(1)],'All Cycles','Mean
Cycle','Knock Cycle','No-Knock Case')

figure()
Tbulkm=MeanPressures(Tbulk);
TbulkKm=MeanPressures(TbulkK);
TbulkSm=MeanPressures(TbulkS);

figure()
p1=plot(CAsim,Tbulk,'-m','LineWidth',0.3) %T bulk
hold on
p2=plot(CAsim,Tbulkm,'--c','LineWidth',1.5)
hold on
```

```
p3=plot(CAsim,TbulkKm,'-k','LineWidth',2)

hold on

p4=plot(CAsim,TbulkSm,'-y','LineWidth',2)

hold on

xlabel('Crank Angle (^{o})')

ylabel('Bulk Temperature (K)')

legend([p1(1) p2(1) p3(1) p4(1)],'All Cycles','Mean
Cycle','Knock Cycle','No-Knock Case')


figure()

p1=plot(CAsim,TbulkK,'-m','LineWidth',0.3) %T bulk K

hold on

p2=plot(CAsim,TbulkKm,'c','LineWidth',1.5)

hold on

xlabel('Crank Angle (^{o})')

ylabel('Bulk Temperature (K)')

legend([p1(1) p2(1)],'All Cycles','Average Cycle')


figure()

p1=plot(CAsim,TbulkS,'-m','LineWidth',0.3) %T bulk S

hold on
```

```
p2=plot(CAsim,TbulkSm,'c','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Bulk Temperature (K)')
legend([p1(1) p2(1)],'All Cycles','Average Cycle')


figure()
p1=plot(CAsim,h,'-m','LineWidth',0.3) %h
hold on
p2=plot(CAsim,hm,'--c','LineWidth',1.5)
hold on
p3=plot(CAsim,hKm,'-k','LineWidth',2)
hold on
p4=plot(CAsim,hSm,'-y','LineWidth',2)
hold on
xlabel('Crank Angle (^{o})')
ylabel('h (J/(CA-m^{2}-K')
legend([p1(1) p2(1) p3(1) p4(1)],'All Cycles','Mean
Cycle','Knock Cycle','No-Knock Case')
```

```
figure()
p1=plot(CAsim,hK,'-k','LineWidth',0.3) %T bulk S
hold on
p2=plot(CAsim,hKm,'y','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('h (J/(CA-m^{2}-K')
legend([p1(1) p2(1)],'All Cycles','Average Cycle')
xlim([0 30])

figure()
p1=plot(CAsim,hS,'-k','LineWidth',0.3) %T bulk S
hold on
p2=plot(CAsim,hSm,'y','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('h (J/(CA-m^{2}-K')
legend([p1(1) p2(1)],'All Cycles','Average Cycle')
xlim([0 30])

figure()
```

```
p1=plot(CAsim,hLossm,'-r')
hold on
p2=plot(CAsim,hLossKm,'-k')
hold on
p3=plot(CAsim,hLossSm,'-m')
xlabel('Crank Angle (^{o})')
ylabel('Heat Transfer Rate (J/CA)')
legend([p1(1) p2(1) p3(1)],'Mean Cycle','Knock Cycle','No-
Knock Cycle')

figure()
p1=plot(CAsim(1:1200),hReleasem,'-r')
hold on
p2=plot(CAsim(1:1200),hReleaseKm,'-k')
hold on
p3=plot(CAsim(1:1200),hReleaseSm,'-m')
xlabel('Crank Angle (^{o})')
ylabel('Heat Released (J)')
legend([p1(1) p2(1) p3(1)],'Mean Cycle','Knock Cycle','No-
Knock Cycle','Location','northwest')
```

```
figure()
p1=plot(CAsim,Lossm,'-r')
hold on
p2=plot(CAsim,LossKm,'-k')
hold on
p3=plot(CAsim,LossSm,'-m')
xlabel('Crank Angle (^{o})')
ylabel('Heat Loss Rate (J/CA)')
legend([p1(1) p2(1) p3(1)],'Mean Cycle','Knock Cycle','No-
Knock Cycle','Location','northeast')

figure()
p1=plot(CAsim,LossK,'-r','LineWidth',0.3) %T bulk S
hold on
p2=plot(CAsim,LossKm,'y','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Heat Release Rates (J/(CA-m^{2}-K')
legend([p1(1) p2(1)],'All Cycles','Average Cycle')
xlim([0 30])
```

```
figure()
p1=plot(CAsim,LossS,'-r','LineWidth',0.3) %T bulk S
hold on
p2=plot(CAsim,LossSm,'y','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Heat Release Rates (J/(CA-m^{2}-K')
legend([p1(1) p2(1)],'All Cycles','Average Cycle')
xlim([0 30])

figure()
p1=plot(CAsim,COVLossK,'-r','LineWidth',1.5) %T bulk S
hold on
p2=plot(CAsim,COVLossS,'-k','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Coefficient of Variation')
legend([p1(1) p2(1)],'Knock','No-
Knock','Location','northwest')
xlim([0 30])
```

```
figure()
p1=plot(CAsim(1:1200),MFBm,'-r')
hold on
p2=plot(CAsim(1:1200),MFBKm,'-k')
hold on
p3=plot(CAsim(1:1200),MFBSm,'-m')
xlabel('Crank Angle (^{o})')
ylabel('Mass Fraction Burned')
legend([p1(1) p2(1) p3(1)],'Mean Cycle','Knock Cycle','No-
Knock Cycle','Location','northwest')

figure()
p1=plot(CAsim(1:1200),TotalHeatKm,'-r','LineWidth',1.5) %T
bulk S
hold on
p2=plot(CAsim(1:1200),TotalHeatSm,'-g','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Total Heat Released (J)')
legend([p1(1) p2(1)],'Knock','No-
Knock','Location','northwest')
```

```
figure()
p1=plot(CAsim,TIKm,'-r','LineWidth',1.5) %T bulk S
hold on
p2=plot(CAsim,TISm,'-c','LineWidth',1.5)
hold on
xlabel('Crank Angle (^{o})')
ylabel('Isentropic Unburned Temperature (K)')
legend([p1(1) p2(1)],'Knock','No-
Knock','Location','northwest')


c=input('Would you like to see all plots? \nif yes enter
1\nif no enter 2\n')
if c==2
    close all
end
```