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Design of 1.6 Liter Genset Engine

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UNIVERSITY OF MIAMI

DESIGN OF 1.6 LITER GENSET ENGINE

By
Hasitha Samarajeewa

A THESIS

Submitted to the Faculty
of the University of Miami
in partial fulfillment of the requirements for
the degree of Master of Science

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UNIVERSITY OF MIAMI

A thesis submitted in partial fulfillment of
the requirements for the degree of
Master of Science

DESIGN OF 1.6 LITER GENSET ENGINE

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Generators are widely used across the world as portable power units in case of power outages, used for emergency services and are also used in rural areas without access to electricity. The majority of commercially available generators use internal combustion engines designed as automobile engines with little or no optimization for use in generators. With operating conditions vastly different than that of automobile engines, they can be re-designed to operate much more efficiently as generator engines. The development objective here was to design a low cost, 1.6L, lean burn, internal combustion engine which minimizes heat losses, time losses and frictional losses to improve thermal efficiency. Various high swirl, high squish, easily CNC’d combustion chambers were created in the re-design process. A computer model was used to provide insight into the trade-off between time losses and heat losses. A maximum brake thermal efficiency of 37.2% was achieved.
Acknowledgement

I would like to thank my advisor, Dr. Michael Swain. Without his help, this would have never been possible.

I would like to thank Mr. Mathew Swain for his immense help with setting up experiments and insight into tackling problems.

I would like to thank my parents, who encouraged me to do my best in school and life.
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**Definitions**

BMEP: Measured brake or dynamometer work output per unit swept volume.

Compression Ratio: The volume ratio between the piston and cylinder head before and after a compression stroke.

IMEP: Indicated work output per unit swept volume.

Mach Index Number: Ratio of port gas velocity relative to speed of sound.

PMEP: Pumping work per unit swept volume.

Ratio of Specific Heats (k): ratio of the heat capacity at constant pressure \( (C_p) \) to heat capacity at constant volume \( (C_v) \).

Spark Advance: The timing of ignition relative to the top dead center of the piston given in crank shaft degrees.

Squish: Gas motion caused when the piston is reaching Top Dead Center (TDC). A narrowing area between the piston and head causes flow into the main area of the combustion chamber.

Swirl: Bulk rotation of gasses about the axis of the cylinder.
Chapter 1

Introduction

In times of emergency, common sources of electricity can be interrupted. Under these circumstances electricity is supplied by secondary sources such as electric generator sets (gensets). Gensets are available in a wide variety of sizes ranging from 1 kW to several hundred kilowatts. These units have become necessity throughout the world. This particular project is centered on the development of a genset engine which can be made from rebuilt automobile engines. The advantage would be a lower cost, more efficient engine re-designed and optimized for the generator market rather than for the automobile market.

A 1.6L four cylinder engine suitable to develop 15 kW to 17.5 kW was constructed and tested which showed a maximum brake thermal efficiency (BTE) of 37.2%. A computer model was used to compare the trade-off between time losses and heat losses using the best efficiency spark advance (BESA) data rather than in cylinder pressure data.

Determination of High Efficiency

Of considerable importance is the efficiency of the gensets. This affects the cost of fuel during operation to produce electricity. Engine efficiency determines the number of times the unit needs to be refueled under conditions when fuel may be difficult to obtain. These effects place a high premium on improving genset efficiency.

In order to determine the level of performance needed for a genset to be accepted as high-efficiency, a survey was conducted of gensets presently in production. The
gensets used for comparison were operated utilizing propane as a fuel. Propane is the fuel used during this research effort. The efficiency of the gensets was plotted as a function of load (Figure 1.1). Gensets with maximum outputs ranging from 13 kW to 18 kW were used for comparison. It can be seen from this graph that the available gensets produce efficiencies between 10.3% and 13.8% at 25% load and between 19.7% and 24.5% at 100% load.

As gensets grow larger they tend to have higher efficiencies. Internal combustion (IC) engines are heat engines and large IC engines have inherently lower heat losses when compared to small IC engines. The fundamental concept governing this trend is the surface to volume ratio of the combustion chamber. The volume of a combustion chamber increases more rapidly with size than does the surface area of that volume. This
fact gives an inherent advantage to larger IC engines but any technique that reduces the surface to volume ratio of the combustion chamber can increase efficiency.

On the generator side, four pole electric generators which operate at 1800 rpm to produce 60 Hz electricity are more efficient than two pole generators which operate at 3600 rpm to produce 60 Hz electricity. Generator efficiency also tends to increase with generator size. The efficiency does not increase as dramatically with size in generators as it does in IC engines. The increase is principally due to reduction in magnetic field losses which are easier to contain in larger generators.

The least expensive heat engine available to power an emergency genset is an automotive type IC engine. The lowest-cost units are available by rebuilding used automobile engines. Wide selections of potential engines are available as multiple manufacturers have produced engines in the desired range of sizes. Induction generators of this size operate with approximately 85% efficiency and 2% slippage at 100% load. The fan required to cool the engine would require 1.2 kW. Therefore, the desired engine size to produce 17.5 kW of electricity would need to produce 21.7 kW of power at 1836 rpm. An engine of 100 in.³ displacement would need a brake mean effective pressure (BMEP) of 128 psi to produce the desired power. This should be easily accomplished at the low mach index number represented by a 1.6 L automobile engine operating at 1836 rpm.
Design Considerations

An idealized thermodynamic depiction of the Otto Cycle, shown in the following equation, shows high-efficiency can be achieved with a high compression ratio (CR) and high ratio of specific heats (k).

\[ n = 1 - \frac{1}{CR^{k-1}} \]

The 1.6L four-cylinder engine designed by Mazda, manufactured and distributed by Ford, Mazda and Kia is an ideal choice for this purpose due to the wide availability of engines and parts. Also the cylinder head on the 1.3L and the 1.6L can be interchanged with merely simple modifications to the water jacket. The advantage is the ability to increase the compression ratio, without having to weld aluminum to the combustion chamber, by simply swapping to the 1.3L head. Furthermore the smaller intake system, 26 mm intake port diameter and 31.75 mm valve head, yields a mach index number of 0.20 to 0.22 at 1800 rpm which allowed modification to inlet swirl (Figure 1.3).

For this study, a compression ratio of 12.2 was chosen. This compression ratio was found to be the critical for air-propane mixtures utilized in an engine with a greater flame front travel distance than this engine (Obert). The test showing the results were conducted by General Motors.

The increase in compression ratio inherently increases the surface-to-volume ratio which in-turn increases the heat losses. Table 1.1 shows the effect on surface-to-volume ratio relative to the compression ratio and stroke of the engine for a 1.6L four-cylinder engine with hemispherical combustion chambers. For this study, a stroke of 83.6 mm and a bore of 79 mm was selected, which produces approximately 1.6L. Flat-top pistons with
79.0 mm bore and 20 mm wrist pins which weighed 280 g were used to minimalize the surface-to-volume ratio. 0.5 cm³ was cut into the piston head valve relief (Figure 1.2).

<table>
<thead>
<tr>
<th>Stroke (mm)</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
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<tbody>
<tr>
<td>85.0</td>
<td>2.12</td>
<td>2.33</td>
<td>2.55</td>
<td>2.88</td>
<td>2.99</td>
<td>3.22</td>
<td>3.44</td>
</tr>
<tr>
<td>80.0</td>
<td>2.21</td>
<td>2.45</td>
<td>2.68</td>
<td>2.92</td>
<td>3.15</td>
<td>3.39</td>
<td>3.63</td>
</tr>
<tr>
<td>75.0</td>
<td>2.33</td>
<td>2.57</td>
<td>2.83</td>
<td>3.08</td>
<td>3.34</td>
<td>3.59</td>
<td>3.85</td>
</tr>
<tr>
<td>70.0</td>
<td>2.46</td>
<td>2.73</td>
<td>3.00</td>
<td>3.27</td>
<td>3.55</td>
<td>3.83</td>
<td>4.10</td>
</tr>
<tr>
<td>65.0</td>
<td>2.61</td>
<td>2.90</td>
<td>3.20</td>
<td>3.50</td>
<td>3.80</td>
<td>4.10</td>
<td>4.40</td>
</tr>
</tbody>
</table>

Table 1.1 – Surface-to-Volume Ratio

The increase in the ratio of specific heats was achieved by reducing the temperatures of the working fluids. A lower ratio of specific heats reduces both the specific heat at constant volume (Cᵥ) and specific heat at constant pressure (Cₚ). This in turn reduces the pressure increase during combustion thus decreasing the efficiency of the engine. This was mitigated by diluting the fuel with air (lean mixtures), thus reducing the
temperature. Lean mixtures also decrease the pumping mean effective pressure (PMEP) by producing a high intake manifold pressure from the same fuel flow rate, therefore increasing the mechanical efficiency of the engine. An inherent disadvantage of lean mixtures is the reduction in flame speed and increase in time losses.

The effects of swirl and squish (Figure 1.4) can also be considered to improve the efficiency (Nagayama). The advantage of swirl is, the high swirl rates increases the small scale turbulence and flame speed. However, it is disadvantageous that the higher swirl rates increase heat loss at the cylinder walls. For this study, four heads were tested on a flowbench to compare swirl and flow rate and a given configuration (Table 1.2). As the piston moves up, squish forces the air-fuel mixture in the chamber towards the spark plug which increases flame propagation rates, however, too much squish directed at the developing flame kernel can quench flame initiation. Squish was held constant at 18.4% for the research effort. By combining the advantages of the squish, swirl and high compression ratio the flame speed of lean mixture was increased.

<table>
<thead>
<tr>
<th>Combustion Chamber Configuration</th>
<th>Combustion Chamber Wall Shroud Angle</th>
<th>Lift (mm)</th>
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</thead>
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<tr>
<td></td>
<td></td>
<td>2.54 5.08 7.62 10.16</td>
</tr>
<tr>
<td>I</td>
<td>65°</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Airflow rate (m$^3$/min) at 711.2 mmHg</td>
<td>1.31 2.31 2.71 2.79</td>
</tr>
<tr>
<td></td>
<td>Swirl rate (rpm) at 711.2 mmHg</td>
<td>622 770 724 751</td>
</tr>
<tr>
<td>II</td>
<td>70°</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Airflow rate (m$^3$/min) at 711.2 mmHg</td>
<td>1.27 2.28 2.71 2.79</td>
</tr>
<tr>
<td></td>
<td>Swirl rate (rpm) at 711.2 mmHg</td>
<td>630 1584 1440 1272</td>
</tr>
<tr>
<td>III</td>
<td>75°</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Airflow rate (m$^3$/min) at 711.2 mmHg</td>
<td>1.25 2.24 2.65 2.72</td>
</tr>
<tr>
<td></td>
<td>Swirl rate (rpm) at 711.2 mmHg</td>
<td>634 2407 1791 1623</td>
</tr>
<tr>
<td>IV</td>
<td>90°</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Airflow rate (m$^3$/min) at 711.2 mmHg</td>
<td>0.97 1.88 2.44 2.60</td>
</tr>
<tr>
<td></td>
<td>Swirl rate (rpm) at 711.2 mmHg</td>
<td>2320 4232 2912 2763</td>
</tr>
</tbody>
</table>

Table 1.2 – Combustion Chamber Configurations
IC engines designed for automotive purposes operate in a wide rpm band ranging from around 700 rpm to 7000 rpm. One of the design considerations is durability throughout rpm range for the life of the engine. Mechanical moving components such as bearings, connecting rods, valve springs and pistons are designed to withstand the high inertial forces at higher rpm in an automotive engine. In an IC engine on a genset, running at a low speed with minimal rpm fluctuations, these forces are considerably lower and therefore, an automotive engine redesigned for a genset can be improved with the use of smaller, lighter components. This can reduce the inertial forces in moving parts and also the friction, quantified by frictional mean effective pressure (FMEP). (Bishop).

For this research effort crankshaft main bearing and journal bearing sizes were 49.9 mm diameter/17.6 mm length and 39.9 mm diameter/17.1 mm length respectively. Cast iron connecting rods were 132.9 mm long and weighed 418 g. The low friction ring package consisted of a 1 mm steel gas nitrated barrel-face top compression ring and a 1.2 mm cast iron phosphate coated tapered-face napier scraper ring. The three piece oil ring was 2.8 mm wide and used a stainless steel flex vent spacer with gas nitrated rails.

The fuel delivery system was composed of a standard minor diameter radial inlet venturi gaseous fuel mixer. To provide excellent cylinder-to-cylinder fuel distribution, the gas mixture entered a 175 cm³ pre-chamber before passing through a 2 cm diameter orifice into the intake manifold. This intake geometry produced 20 mm of mercury intake manifold vacuum at wide open throttle.
Figure 1.3 - Swirl

Figure 1.4 - Squish
Chapter 2

Data Acquisition

A water-brake dynamometer (Figure 2.1) attached to the crankshaft, absorbs energy creating torque at a given rpm. The rpm itself was accurately found with the use of a strobe (Figure 2.2). The spark advance (SA) was set by the Haltech E8 computer in conjunction with a crankshaft position sensor incorporated in the distributor. The air-fuel ratio ($\lambda$) was measured with the use of an oxygen sensor in the exhaust stream. The oxygen sensor was calibrated specifically for this engine by comparing with the air flow rate, measured with the laminar flow element (Figure 2.3) and fuel flow rate. The air pressure drop before the carburetor was measured with the use of an inclined manometer (Figure 2.4) while the manifold vacuum was directly measured by a pressure gauge attached to the manifold. The exhaust temperature for each cylinder was measured by Super OMEGAACLAD, KMQXL-032G-6 thermal-couples attached to a stainless steel plate between the exhaust ports and exhaust manifold (Figure 2.5). The water temperature, oil temperature and pressure were measured directly off the block. Refer to Appendix for further details.
Figure 2.1 – Water-brake Dynamometer

Figure 2.2 – Strobe
Figure 2.3 – Laminar Flow Element

Figure 2.4 – Inclined Manometer
Warm-up

In order to gather consistent, accurate data it is important to wait until the engine is warmed up and running under stable conditions. A good benchmark is the oil temperature and pressure. As the oil temperature rises, the viscosity of the oil decreases causing a change in friction between lubricated moving parts. Therefore after start-up the oil temperature and pressure were monitored until they become stable.

The test engine was started with 25 degrees SA and the rpm approximately set at 1800 rpm with the throttle. As the oil pressure decreases, an increase in rpm was noticed which is caused by the reduction in friction. Oil temperature of approximately 185°F and pressure of approximately 27 psi were observed as stable conditions for the test configurations. Warm-up phase usually takes around 15-20 minutes. The ambient air temperature and relative humidity were noted before and after warm-up.
Configurations

Four combustion chamber configurations (I, II, III, and IV) listed in Table 1.2 were formed by rotating the combustion chamber a total of 9° at the cylinder head gasket surface. This allowed the swirl intensity to be varied while holding compression ratio, squish percent and surface area constant. Combustion chamber configuration I (Figure 2.6) had the combustion chamber positioned to create a 65° angle between the gasket surface and the combustion chamber wall near the intake valve. As the combustion chamber was rotated the combustion chamber wall moved closer to the intake valve. After 9° of rotation, combustion chamber configuration IV (Figure 2.7), the wall was parallel to the valve stem and 1.14 mm from the head of the valve. The rotation of the chamber created increases in swirl, first noted at high valve lifts (combustion chamber configurations II and III) and finally at 2.54 mm valve lift (combustion chamber configuration IV). The airflow capacity was reduced as swirl increased.
Figure 2.6 – Combustion Chamber Configuration I

Figure 2.7 – Combustion Chamber Configuration IV
Two engine configurations were tested in this research effort. Each configuration was constructed utilizing the same short block (Figure 2.8) and the combustion chamber configurations I and IV listed in Table 1.2. The water return holes for each of the selected cylinder heads were machined to align with the water return holes of the block. The heads were mounted using studs and nuts (Figure 2.9). The original bolts were replaced with studs to increase the longevity of the blocks bolt threads over multiple head swaps. The mating surface was thoroughly cleaned and new head gaskets were used to ensure good sealing when swapping heads.

Figure 2.8 – Four-cylinder Short Block
Procedure

As for any test engine, the goal was to find the maximum torque using the least amount of fuel at a given rpm. The $\lambda$ and SA which produces the highest efficiency must be determined experimentally. It is an iterative process, first choosing a reasonable $\lambda$ and measuring torque output versus spark advance until optimum spark advance is determined for the chosen $\lambda$. The process was repeated until the best efficiency $\lambda$ and corresponding SA were determined for a given load and rpm. For this research effort, four loads (25%, 50%, 75%, and 100%) that operate at 1800 rpm were chosen. These are the industry accepted operating points used to determine genset efficiency.
Analysis and Results

The first test utilized combustion chamber configuration I. This represents the baseline configuration and incorporated the minimum swirl tested. As a result the best efficiency equivalence ratio fell between $\lambda = 1.23$ and $\lambda = 1.31$. Figure 2.10 shows BTE vs. Load. Maximum BTE was 37.2%, while producing 19.3 kW of power at $\lambda = 1.29$. Maximum power was 21.5 kW with a BTE of 36.8% and $\lambda = 1.23$. Even though higher BMEP would have been possible with richer mixtures, a significant decrease in BTE was observed under these conditions. The target 21.5 kW output was achieved with minimum enrichment.

Figure 2.10 – BTE vs. Power Output
Plots of energy balance values versus $\lambda$ for 20, 40 and 60 foot-pounds torque are shown in Figures 2.11, 2.12 and 2.13. They were calculated as a percent of the heating value of the fuel. Percent heat in exhaust was calculated using exhaust gas temperature and flow rate. This was the exhaust gas temperature measured between the cylinder head and exhaust header. The heat losses in the exhaust gases while traveling past the exhaust valve and through the exhaust port were not included in these measurements of percent heat in exhaust. Nor was the kinetic energy in the exhaust gases. Figure 2.14 shows the energy balance values versus load at $\lambda = 1.25$. Percent energy in the exhaust remained relatively constant over the load range. Best efficiency air-fuel ratio at light loads tended towards $\lambda = 1.25$. As load increased BTE remained relatively constant even as the $\lambda$ approached 1.30.

Figure 2.11 – 20 lb-ft Low Swirl
Figure 2.12 – 40 lb-\text{ft} Low Swirl

Figure 2.13 – 60 lb-\text{ft} Low Swirl
The second test utilized combustion chamber configuration IV (Table 2). This configuration was identical to the first engine configuration except the combustion chamber was as previously mentioned, rotated 9°. This produced a large increase in swirl by allowing the combustion chamber wall next to the intake valve, to shroud the passage of air past that side of the valve. The best BTE for the second engine configuration fell between \( \lambda = 1.40 \) and \( \lambda = 1.49 \) air fuel ratios. Figure 2.15 shows BTE vs. Power Output. Maximum BTE for this configuration was 35.9% with 17.9 kW of power at \( \lambda = 1.40 \). Maximum power was 21.6 kW, with a BTE of 33.4% because an air fuel ratio of \( \lambda = 1.09 \) was required to reach this power output.
Plots of energy balance values versus $\lambda$ for 20, 40 and 60 foot-pound torque are shown in Figures 2.16, 2.17 and 2.18. Figures 2.19 and 2.20 show energy balance values versus load at $\lambda = 1.25$ and $\lambda = 1.45$.

Figure 2.15 – BTE vs. Power Output

Figure 2.16 – 20 lb-ft High Swirl
Figure 2.17 – 40 lb\text{f}-ft High Swirl

Figure 2.18 – 60 lb\text{f}-ft High Swirl
It can be seen that for loads between 25 lb\textsubscript{f}-ft and 55 lb\textsubscript{f}-ft of torque on both configurations, the BTE is virtually identical. The second configurations, using leaner mixtures, demonstrated as high as a 12% increase in BTE at loads below 25 lb\textsubscript{f}-ft. For Figure 2.19 – \( \lambda = 1.25 \) High Swirl

Figure 2.20 – \( \lambda = 1.45 \) High Swirl
loads of 70 lb\text{-}ft the mixture had to be much richer than $\lambda=1.40$ and as the load increased the BTE rapidly decreases.

In general, the addition of swirl increases heat transfer to the coolant. It is of interest to note that for the light load points (20 lb\text{-}ft) the first engine configuration has higher portion of heat in the coolant than the second engine configuration. In this light load case the second engine configuration which employs additional swirl allowed best BTE operation at a much leaner mixture. The second configuration was then able to operate leaner, due to the increased swirl, that the combustion duration was longer than that in the first configuration. This increase in burn duration reduced the heat losses due to rapid gas motion during combustion, enough to produce less heat transfer to the coolant in the second configuration even though there was more swirl. This phenomenon only occurred at light load operation. See computer model section for additional analysis.
Chapter 3

Computer Model

The proper balance between heat losses and time losses must be designed into an Otto cycle engine to improve efficiency. If the air-fuel mixture takes a long time to burn (large burn duration) the piston, on average, is further from top dead center during the combustion process. This reduces the effective expansion ratio for the burning gases and lowers cycle efficiency. Lean air-fuel mixtures burn slowly and therefore flame speed was increased with swirl and squish for this research effort. Swirl and squish add turbulence to the air-fuel mixture to increase flames speed, increase gas velocities relative to the walls of the combustion chamber and also increase heat losses.

The following modeling was done to identify key variables affecting performance and not to predict engine behavior. The data showed the engines exhibiting dramatic decreases in BTE at high loads. It is clear that, as load increases the burn duration decreases until time losses have been virtually eliminated. A further decrease in burn duration does not show efficiency gains due to decreased time losses and, in fact, shows decreased thermal efficiency due to increased heat losses. A limited scope thermodynamic computer model of the cycle was constructed to investigate the change in heat transfer with change in burn duration. Woschni's equation $h_c = f(P, T, V)$ defines a heat transfer coefficient for heat transfer from the combustion gases to the chamber wall that is a function of pressure $P$, temperature $T$ and gas velocity $V$. This model assumes Woschni's pressure and temperature dependency $P^{0.8}T^{-0.55}$ and then solves for the gas velocity dependent term. An underlying assumption is that 10% - 90% burn duration lies between 65% and 110% of the BESA value. The model was limited in the scope of its
applicability to reduce the complexity of the model. The model was developed for air-fuel mixtures between \( \lambda = 1.54 \) and \( \lambda = 1.23 \). This is the range of air-fuel ratio of interest in designing a high efficiency genset engine. The restriction of, no mixtures richer than \( \lambda = 1.23 \) allows the model to neglect the effects of dissociation which occurs at high temperatures. Dissociation produces delayed and reduced energy release from the fuel.

The restriction of no mixtures leaner than \( \lambda = 1.54 \) allows the assumption of complete combustion in the model. It should be pointed out that the model is only used to analyze mixtures as lean as \( \lambda = 1.54 \) for the engines employing squish and swirl. This model was specific to the research engine in this work. The scope of the model was limited to a bore of 79 mm, stroke 83.6 mm, 1800 rpm, compression ratio of 12.2, squish of 18.4\%, and the computer analysis was done on data taken at BESA. The model predicted the work loop in the pressure versus volume (PV) diagram (Figure 3.1). The model began with the compression of an experimentally determined amount of air and propane in the cylinder. Isentropic compression continued utilizing a constant ratio of specific heats for the given air-propane mixture until the crank angle at which combustion began. The crank angle at which combustion begins is calculated from the combustion duration and Wiebe function.

To model combustion the gases in the cylinder were separated into a series of "slices" with the volume of each slice determined from the mass fraction Wiebe function. The "slices" were created such that each "slice" represented the volume of gases burned during 0.3° crankshaft rotation. Each slice was combusted as follows.
First, a new temperature and volume was calculated for the slice assuming combustion at constant pressure. The heat released by the slice was computed using its lower heating value of propane (19,916 BTU/lb.). An average specific heat at constant pressure was calculated from known adiabatic flame temperatures for air-propane mixtures at the given air-fuel ratio. The average specific heat at constant pressure was used to calculate the new temperature of the slice. The ideal gas law was then used to determine the new volume of the slice after constant pressure combustion. An approximation for the thermodynamic state of the gases at a volume equal to the cylinder volume was made by adiabatically compressing them using an average value of the ratio of specific heats for the burned and unburned gases. Heat transfer out of the gases into the combustion chamber wall was computed at this point using a modified Woschni equation and the approximate thermodynamic state of the gases. The modified Woschni
The equation used for simulating heat transfer was applied to each individual slice assuming the surface area available for heat transfer from the slice, relative to the total surface area of the combustion chamber, was linearly proportional to its relative volume in the combustion space. The final thermodynamic state of the gases, prior to burning the next slice, was determined by iteration using adiabatic compression of each slice with the appropriate value of ratio of specific heats for the burned and unburned gases and matching the total volume of the gases to the volume of the combustion chamber at the appropriate crankshaft position. Woschni’s equation was modified for constant 1800 rpm operation. Woschni’s equation for heat transfer coefficient was as follows:

\[
h = P^b \left( c_1 V_p + c_2 \frac{V_s T_r}{P_r V_r} (P - P_m) \right)^b x^{b-1} T^{(0.75 - 1.62b)}
\]

Where:

- \( P \) = cylinder pressure
- \( b = 0.8 \)
- \( x \) = cylinder bore diameter
- \( T \) = cylinder gas temperature
- \( P_m \) = motoring cylinder pressure without combustion
- \( V_s \) = swept volume
- \( V_r, T_r, P_r \) = volume, temperature, and pressure at intake valve closing
- \( V_p \) = mean piston speed

For gas exchange: \( c_1 = 6.18 \quad c_2 = 0 \)

For compression: \( c_1 = 2.28 \quad c_2 = 0 \)

For combustion and expansion: \( c_1 = 2.28 \quad c_2 = 3.24 \times 10^{-3} \)
Since this equation is meant for use with a variety of engines at a variety of rpms it can be simplified when used with a single engine at a set rpm. The $c_1V_p$ term depicts gas velocity dependency on rpm and stroke and the $c_2\frac{V_sT_r}{P_rV_r}(P - P_m)$ term is meant to depict gas velocity dependency on gas motion produced by combustion. The $x^{b-1}$ term is dependent on bore size. For this work the bore, stroke, and rpm are constant. The model therefore used a heat transfer coefficient of the following form.

$$C_{kv} P^{0.8} T^{-0.55}$$

Where $C_{kv}$ replaces:

$$\left( c_1V_p + c_2\frac{V_sT_r}{P_rV_r}(P - P_m) \right)^b x^{b-1}$$

in the Woschni formulation. The model then calculated a value for $C_{kv}$ to allow the model to match the experimentally determined IMEP. A different value for $C_{kv}$ was calculated for each experimental data point. The value of $C_{kv}$ is a measure of the heat loss due to gas velocity at that particular operating condition. The model requires an assumed value for 10% to 90% mass burn duration and Wiebe function variables $a$ and $m$. Ultimately, analysis of the measured data using the model allowed $C_{kv}$ to be replaced with the following term which achieved a good match to the experimental data.

$$10.01 \left( \frac{46}{dur} \right)^{\frac{46 - dur}{24}} - 3.597 \left( \frac{46 - dur}{24} \right)^{12.2} - 0.8238$$

Where: $dur =$ combustion duration from 10% mass burn to 90% mass burn

After combustion was complete, the mass average temperature of the burning gases in the combustion chamber was calculated using the ideal gas law. The temperature and pressure during the expansion stroke was also computed in 0.3°
increments. At each increment the state of the gases was calculated including heat losses to the combustion chamber until the crankshaft reached 54° before bottom dead center (BBDC). At 54° BBDC, modeling of the blow-down process was initialized. Gases were modeled as leaving the combustion chamber at the speed of sound calculated from the ratio of specific heats and temperature of the gases ($c = \sqrt{kRT}$). At each 0.3° increment in crankshaft rotation the mass of gases leaving the cylinder, together with a reduction in temperature and pressure due to the isentropic expansion of the gases remaining in the chamber was calculated. Because of the small size and slow opening rate of the exhaust valve, the reduction in Indicated Thermal Efficiency (ITE) due to blow-down never exceeded 0.5%. The model then uses iteration to match the calculated IMEP determined by the methods presented in "Development and Evaluation of a Friction Model for Spark - Ignition Engines" (Patton). This method relies on the earlier work of I. Bishop and was chosen because it predicts the FMEP of a running engine as opposed to a motored engine. Input data were, experimentally determined BMEP, intake manifold pressure, atmospheric pressure, and rpm. Additionally, engine geometry was defined by bore, stroke, the number, diameter and length of all bearings, valve dimensions, and compression ratio.

The sensitivity analysis showed that the conclusions drawn from the model were independent of the following assumptions:

- Piston, ring and auxiliary load assumed.
- 10% to 90% mass burn durations derived from 65% of BESA to 110% of BESA.
- Wiebe function parameters from (4, 4) to (12, 1)
Sensitivity Analysis of Model Parameters

Friction was initially modeled using the method described in “Development and Evaluation of a Friction Model for Spark Ignition Engines” (Patton). However, the mass of the intake and exhaust valve assumed by the authors was changed to the actual masses of the valves used in the research engine. The authors assumed typical valve sizes for a 1.6L automotive engine. The valves in the research engine were smaller and weighed 27.6% less than the valves described in the paper. The value of the valve mass friction terms were therefore reduced by 27.6%. The model was used to predict $C_{kv}$ versus 10% to 90% mass burn duration. The results shown in green in Figure 3.2 were for the original friction equation, together with a best estimate Wiebe function, and 10% to 90% mass burn duration equal to 95% of BESA. It can be seen that as burn duration decreases the heat losses increase by slightly more than a factor of two. This explains the rapid loss in BTE at high loads. The larger the proportion of the combustion process occurring within 30° of top dead center (TDC) the more the high gas velocities created by squish increase the heat losses.

Additional modifications were then made to the friction equation. The piston/ring combination used in the short block produced less friction than the standard 1989 piston. The piston skirt hydrodynamic friction and piston ring friction without gas pressure were therefore reduced by 10%. Additionally, the calculations of power loss due to auxiliaries in the original method assumed standard oil pump, water pump, and nonfunctioning alternator. The lab engine had no alternator, a nine lobe instead of ten lobe oil pump and the water pump rotational speed was reduced 7%. The auxiliary friction term was therefore reduced by 20%. The result for the new friction model is shown in red in Figure
3.2. These assumptions increase the maximum mechanical efficiency from 88.5% to 89.0% while reducing ITE from 41.5% to 41.3%. Both the original and new friction model results were similar and display the characteristic increase in $C_{kv}$ at low burn duration as shown in Figure 3.2. The new friction model was used for all subsequent calculations.

The model uses the Wiebe equation

$$f(x) = \frac{1 - e^{(-a(x)^m)}}{1 - e^{-a}}$$

to define mass fraction burned versus normalized burn time ($dur$). Figure 3.3 shows Wiebe curves for various values of "a" and "m". The range depicted from $(a,m) = (12, 1)$ to $(4, 4)$ covers the vast majority of combustion events. The values $(a,m) = (5, 2)$ are commonly used (Rogers). Figure 3.4 shows the results of using the model to predict $C_{kv}$.
versus burn duration. Plotted with the other curves is a curve listed as “Best estimate”. It is an experimentally derived curve for a previously tested 1.6 L engine with similar swirl and squish to the first engine tested. The results calculated with “Best estimate” are virtually identical to Wiebe (6, 1). Notice also that the results for "Best estimate", (6, 1), (5, 2), and (5.5, 1.5) are all quite similar.

Figure 3.3 – Wiebe Curve for common “a” and “m” values.
The Wiebe parameters "a" and "m" required to match the real mass burn curve can change with air-fuel ratio. As a worst-case scenario, if we assume the mass burn curve changes from a Wiebe (12,1) at 46° burn duration to a Wiebe (6,1) at 20° burn duration it can be seen that the value for $C_{kr}$ has still almost doubled. The model is therefore relatively insensitive to the choice of Wiebe parameters. The computer model adjusts the timing of the provided Wiebe curve to yield the best efficiency. The 10% to 90% burn duration is held constant regardless of the values specified for the Wiebe equation.

The last and most significant assumption needed as input to the model is burn duration. Burn duration is herein defined as the number of crank angle degrees for 10% to 90% mass burn. Inspection of data taken on the other lean burn engines (Nakamura) employing swirl and squish, showed burn duration to fall between 65% and 110% of

Figure 3.4 – Heat Transfer Coefficient with common Wiebe values.
BESA. This broad range does not exist in a single engine. Values for a single engine did not vary more than 15% of BESA as air-fuel ratios were varied from $\lambda = 1.0$ to the lean limit of combustion. Figure 3.5 shows the result of varying the ratio of burn duration to BESA from 65% to 110%. It can be seen that a larger range of $C_{kv}$ values exist than that found when varying the Wiebe equation parameters. The literature shows that the burn duration: BESA ratio increases with leaning. Lean mixtures tend to have longer burn durations. If, in the worst case, the burn duration: BESA ratio decreases to 65% at rich mixtures (short burn duration) and increases to 110% for lean mixtures (long burn duration) the heat losses at the short burn durations would increase by more than a factor of two. As the literature shows, the burn duration: BESA ratio will not likely increase more than 15% for rich mixtures which produces a result that is less than the worst case and again yields an approximate doubling of $C_{kv}$ from 46° to 20° burn duration. Figure 3.6 shows a typical plot of $C_{kv}$ versus burn duration for the modified friction model, best estimate Wiebe curve, and burn duration = 95% BESA.
Figure 3.5 – Varying Burn Duration for Leaning Mixtures.

Figure 3.6 – Heat Transfer Coefficient for New Friction Model
Chapter 4

Conclusion

1. Swirl Ratios and squish areas used in automobile engines designed for gasoline operation are not sufficient for lean burn propane fueled engines.

2. High compression ratio (12.2:1) alone do not provide enough increase in flame speed to optimize a $\lambda = 1.0$ gasoline engine for use as lean burn, propane fuel genset engine. This would render the optimized genset engine unsuitable for automotive use.

3. Squish areas large enough to improve combustion chamber geometry for lean burn propane fuel operation can produce large heat losses if mixtures richer than ($\lambda \leq 1.25$) are used.

4. A 20% improvement in fuel consumption across the operating range is possible if combustion chamber geometries and frictional losses are designed for lean burn propane operation as opposed to designs used for stoichiometry.

5. The use of pre-chamber between the gas mixer and intake manifold runners can provide sufficient mixing to virtually eliminate cylinder-to-cylinder fuel distribution problems and still require only 2.8 kPa intake manifold vacuum at 1800 rpm wide-open throttle.

6. Valve spring pressure used in automotive gasoline engines can be reduced at least 50% at all valve lifts without adverse effects. This has reduced FMEP approximately 3 to 4 kPa at 1800 rpm for this application.
7. Durability tests have not been performed with any of the engine configurations tested. It is not anticipated durability issues would arise if the engine was operated with the proper oil type.
References


## Appendix

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