Feasibility Study of a Low Heat Rejection Generator Set Engine

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FEASIBILITY STUDY OF A LOW HEAT REJECTION GENERATOR SET ENGINE

By

Oliver Patrick Jordan

A THESIS

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the requirements for the degree of
Master of Science

FEASIBILITY STUDY OF A LOW HEAT REJECTION GENERATOR SET ENGINE

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Internal combustion engine powered generator sets are vastly used worldwide, usually with minimally modified automobile engines. The automobile engine is designed for a very different operation than that required in a generator set, resulting in poor efficiencies when compared to those possible with a more thorough redesign. This research effort investigates the feasibility of using engine parts from multiple engines (principally a Nissan KA24 block and a Nissan Z20 cylinder head) to create a new engine designed specifically for use in a generator set, with a constant engine speed of 1800 RPM. The engine, dubbed the Nissan ZKA26, is fueled by a lean mixture of LPG and has a high compression ratio of 14.6:1. The combustion chamber is hemispherical, resulting in low surface area to volume ratios, which combined with the lack of swirl and squish result in minimal heat losses compared with previous research in this field.
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Chapter 1

Introduction

Internal combustion engine powered generator sets are the world’s number one electrical energy source for off-grid locations such as homes and small businesses, as well as the number one emergency power source for critical demand locations such as hospitals and airports. The cost of running such generator sets is significantly reduced with an increase in the engines’ brake thermal efficiencies. An increase in thermal efficiency also reduces fuel storage mass and volume for a required output of energy in locations without gas pipeline access.

Before introducing ways of increasing the thermal efficiencies of such engines, an analysis is made to investigate the theoretical limit of the brake thermal efficiency of the engine chosen for this study, being that 100% efficiency would violate the second law of thermodynamics. Reasons for choosing this particular engine will be discussed in the next chapter.

The Theoretical Limit

Figure 1 on the following page shows the theoretical pressures in each cylinder as a function of volume for a set of four cycles, where after the fourth cycle is complete the set repeats itself 900 times per minute, for an engine speed of 1800 revolutions per minute (RPM). This diagram is referred to as a P-V diagram. In this idealized cycle, the fuel/air mixture is ignited when the piston is at top dead center (TDC) and instantaneously converts the chemical energy from the fuel to thermal energy, immediately raising the temperatures and pressures in the cylinders. Equation 1 on the following page shows the balanced stoichiometric equation with dry air, using 20.948% oxygen in air and the rest as nitrogen (even though 0.968% of air is neither oxygen nor
nitrogen, it is mostly inert like nitrogen, so it will be factored into the chemical equations as nitrogen), in which the fuel (C\textsubscript{3}H\textsubscript{8}) is then multiplied by the equivalence ratio of 0.625

\[ C_3H_8 + 5O_2 + 18.87N_2 \rightarrow 3CO_2 + 4H_2O + 18.87N_2 \]

Equation 1

\[ 0.625C_3H_8 + 5O_2 + 18.87N_2 \rightarrow 1.875CO_2 + 2.5H_2O + 18.87N_2 + 1.875O_2 \]

Equation 2

(\(\lambda = 1.6\)), and the right side of the equation balanced in order to obtain the actual lean combustion chemical equation, shown above in equation 2. The lb\textperiodcentered mol of fuel (0.625\times44 lb\textperiodcentered mol C\textsubscript{3}H\textsubscript{8}) is then multiplied by the lower heating value of propane (19944 Btu/lb) to obtain the total amount of thermal energy converted from the chemical energy of the fuel.
q = 548,460 Btu. Using an ideal, constant ratio of specific heats for air of 1.4 (equation 3) and the definition of the universal gas constant (equation 4) the specific heat at constant volume \( C_V \) is calculated and used in equation 5 (where \( n \) is the total number of moles of products, summed from the right hand side of equation 2) to find the rise in temperature due to combustion of the fuel/air mixture. The compression ratio is 14.6:1, to match the compression ratio of the engine used in this research. The displacement of each cylinder

\[
\frac{C_p}{C_v} = k = 1.40 \\
C_p - C_v = R = 1.987 \frac{Btu}{lb \cdot mol^\circ R} \\
\frac{q}{C_v n} = \Delta T = 4395^\circ R
\]

**Equation 3** **Equation 4** **Equation 5**

is calculated with the bore and stroke, as the volume of a cylinder \( (\text{stroke} \times \pi \times \text{bore}^2 \div 4) \), from which the clearance volume can be calculated (equation 6), which is the volume of the air/fuel mixture when the cylinder is at the top dead center (TDC) position. It is

\[
\text{ClearanceVolume} = \frac{\text{Displacement}}{CR - 1} = 47.10\text{cm}^3 = 2.874\text{in}^3
\]

**Equation 6**

also assumed that none of the heat gained from combustion is transferred out of the cylinder, but instead is partially converted to mechanical work and partially remains in the exhaust gases as thermal energy. The work is done on the piston, as the high pressures

\[
P_2 = P_1 \left( \frac{V_1}{V_2} \right)^k \\
T_2 = T_1 \left( \frac{V_1}{V_2} \right)^{k-1}
\]

**Equation 7** **Equation 8**

push it down, rotating the crankshaft. This expansion process is completely adiabatic and isentropic, as is the compression done to the gases before ignition, allowing for temperatures and pressures before or after expansion or compression to be calculated.
with the isentropic relations in equations 7 and 8. With the temperature, volume, and pressure known, the number of moles of molecules, \( n \), can be calculated with the ideal gas law (equation 9). After combustion, \( n \) is calculated by multiplying the previous value

\[ PV = nRT \]

Equation 9

by the molar ratio of combustion, calculated by dividing the number of moles of products by the number of moles of reactants, summed from the right and the left side of equation 2, respectively. Finally, the air/fuel mixture in each cylinder is assumed to be completely homogeneous and also zero variations exist cycle-to-cycle and cylinder-to-cylinder. Labeling four points in the cycle as demonstrated below in figure 2, the total energy input from the fuel (\( E_{in} \)) can be calculated as such: \( q \times n_1 \) divided by the number of moles of reactants (summed from the left of equation 2). The work done during compression and expansion can be seen in equations 10 and 11 on the next page. Work done during the pumping loop is zero, because the engine is at wide open throttle (WOT) for best efficiency, and so the
intake and exhaust manifold pressures are both equal to the ambient atmospheric pressure. The thermal efficiency for the cycle is calculated with equation 14.

\[
W_{\text{compression}} = \frac{P_2 V_2 - P_1 V_1}{1 - k} = -2965in \cdot lb \\
W_{\text{expansion}} = \frac{P_4 V_4 - P_3 V_3}{1 - k} = 11590in \cdot lb
\]

Equation 10  
Equation 11

\[
\text{IMEP} = \frac{W_{\text{compression}} + W_{\text{expansion}}}{\text{displacement}} = 220.6 \text{ psi} \\
\text{BMEP} = \text{IMEP} - \text{FMEP} = 206.6 \text{ psi}
\]

Equation 12  
Equation 13

\[
\eta_i = \frac{W_{\text{compression}} + W_{\text{expansion}}}{E_{\text{in}}} = 66.32\% \\
\text{BTE} = \eta_i \times \frac{\text{BMEP}}{\text{IMEP}} = 62.11\%
\]

Equation 14  
Equation 15

Taking into account 14 psi of frictional mean effective pressure (FMEP), the ideal brake thermal efficiency (BTE) of this engine (seen above in equation 15) is 62.11%.

Despite a theoretical limit of over 60% for the BTE, most generator engines have a BTE of approximately 25%, with the highest efficiencies from the most expensive engines reaching 31% as of 2013. This is because of three primary reasons, which include time losses, heat losses, and the decrease in the ratio of specific heats of the gasses \((k)\) as temperature increases. [1, 2, 3, 4, 5]

The Practical Limit

In a real spark ignition engine, when the fuel is ignited it takes a finite time for the fuel/air mixture to go through combustion. This amount of time is long enough that the piston moves an appreciable distance between the time combustion starts and the time it ends. The proposed way to make this time insignificant is with a Homogeneous Charge Compression Ignition (HCCI) design, which currently is not sophisticated enough and
has not yet been developed enough to be controlled for use in a practical application (generators, automobiles, etc.). The alternative is to find ways to speed up combustion so these time losses are reduced. An inexpensive and effective way to do this is with squish and swirl, as can be seen in Development of a Propane Fueled 1.6 L Genset Engine. Squish and swirl make the mixture in the combustion chamber move faster, causing small scale turbulence which speeds up combustion. This decreases time losses but adds to another problem: heat losses. [6]

No engine contains truly 100% adiabatic processes. Heat transfer is always prevalent when there is such a large difference in temperature (over 3000° C in the combustion chamber versus ambient conditions just a few centimeters away.) Even though peak temperatures only last for less than a hundredth of a second at a time, the movement of the gasses enables large heat losses through convection. If the combustion chamber has a large surface area to volume ratio, then the problem becomes worse because there is more available area for heat transfer to occur. Therefore, the minimization of heat losses involves relatively quiescent gasses and a small surface area to volume ratio of the combustion chamber.

If the molecules of a gas have more degrees of freedom, which is a direct result of having more atoms in each molecule, then some of that internal energy is stored in the actual molecule as it vibrates, as opposed to just being stored as translational kinetic energy of the whole molecule, as would be expected in a monatomic gas. Therefore, the more heat is added to this non-monatomic gas, the more of the heat is stored in the gas’s molecules, and the less of it is available to do work. This is directly represented and quantified in the value of the ratio of specific heats, $k = \frac{C_p}{C_v}$, of that gas. So the lower
the ratio of specific heats of the working fluid the less work can be produced by the compression-ignition-expansion process, and the ratio of specific heats of a non-monatomic gas lowers significantly as that gas’s temperature rises.

Since air is over 99% diatomic (78.084% N\textsubscript{2} and 20.948% O\textsubscript{2}) and the fuel has eleven atoms per molecule (C\textsubscript{3}H\textsubscript{8}), the ratio of specific heats of the mixture is directly related to temperature, so one way of keeping this ratio low is by having lower peak temperatures in the cylinder. When this is done, less power is available from the engine. However, if this is done by leaning out the mixture (having it contain more air and less fuel) then less energy is also put into the engine, therefore preventing any decreases in efficiency, in addition to decreasing heat losses since the ΔT between the air/fuel mixture and the environment is smaller. For a generator, a large power to weight ratio for the engine is not as necessary as it is in an automobile, which has to take the engine with it wherever it goes. Lean mixtures will be used in this research effort to increase the ratio of specific heats of the working fluid and reduce the temperature gradient driving the heat losses. Unfortunately, lean mixtures do not burn rapidly, as can be seen on a plot of flame speed versus percent fuel in air/fuel mixture in figure 3 on the following page. Therefore, lean mixtures increase time losses. Notice the particularly low flame speeds of propane (C\textsubscript{3}H\textsubscript{8}). [1, 7]
Figure 3

Flame velocity, ft/s

m/s (ft/s x 0.3048)

Per cent gas in primary air-gas mixture

0 10 20 30 40 50 60 70

C₃H₈

C₂H₂

C₂H₄

CH₄

Carbureted water gas

Natural gas

Coke oven gas

CO
Chapter 2

Engine Choice

The three previously mentioned problems that affect efficiency need to be juggled in the best way possible in order to approach the theoretical value for BTE. This was done in Development of a Propane Fueled 1.6 L Genset Engine through the approach of reducing time losses to a reasonable level, and then doing whatever is feasible to reduce the other problems. This study is taking the other logical approach: reducing the heat losses as much as possible, then doing whatever is feasible to reduce time losses. [6]

It wouldn’t make sense to work on obtaining gains in one area knowing that it would be counteracted by substantial losses in another. Therefore, a new method of reducing time losses had to be available instead of utilizing swirl and squish which increase heat losses. This is where the Nissan Z family of cylinder heads came into play. These cylinder heads are available brand new, for a very low cost. The brand new head used in this engine can be seen above in figure 4. These cylinder heads have a hemispherical combustion chamber with a low surface area to volume ratio. Most
importantly, they have a location that can be drilled and tapped to fit in a second spark plug, as can be seen below in figure 5. Having multiple ignition sites decreases the combustion time without adding appreciable extra motion to the mixture, allowing time losses to be reduced without affecting heat losses substantially. Most of today’s available cylinder heads were not designed to accommodate multiple spark plugs, so adding a second spark plug to another cylinder head would require the addition of more material through welding. [8]

The Z blocks that were designed to be paired with the Z cylinder heads are difficult to obtain because they are over 30 years old. Also, they are bulky and overdesigned, and thus produce more friction. The KA block, however, is very easy to obtain; available in great numbers from most local junkyards at a very low price. It is also less bulky and has a more efficient design with smaller bearings, producing less friction. The KA head, however, has three valves per cylinder, and no room for a second spark plug, as the Z head does. Fortunately, the Z head has the same bore spacing as the KA
block, and this study will determine if this hybrid ZKA engine is feasible for the application to an electrical generator set. The block used in this engine can be seen in figure 6 below, without the cylinder head. [9]

![Figure 6](image)

**Engine Design**

Because of the low knock characteristics of lean propane and other gaseous fuels, an extremely high compression ratio can be used to obtain better efficiencies. This engine was bored from 88.9 mm to 90.2 mm and stroked from 96.0 mm to 100.25 mm for a displacement of 2562.4 cm\(^3\) (2.6 L), versus the 2383.6 cm\(^3\) (2.4 L) of the original KA24 engine. [5, 10]

Three different types of pistons that match the desired size were ordered for analysis: the hypereutectic Sealed Power Federal Mogul piston, the forged Mahle piston, and the hypereutectic Silvolite piston. The Mahle pistons were forged out of #4032 Aluminum, which contains a high concentration of Silica so, like the cast hypereutectic pistons, don’t have as high a coefficient of thermal expansion, while still not being so
brittle that they can’t be forged. Normally, forged pistons are made out of a #2000 series aluminum, like the #2618 aluminum used in a different Mahle forged racing piston, which have a much higher coefficient of thermal expansion, often requiring upwards of 0.004 inches of piston to cylinder wall clearance: four times what is intended to be used in this engine. The smaller clearances (0.001 inches) used in this engine aid in efficiency and most importantly reduce unburnt hydrocarbon emissions. The three pistons can be seen below in figure 6. Their weights are 356.4, 388.2, and 371.9 grams, respectively. Despite being the heaviest of the three, the inexpensive 90.2 mm Mahle PowerPak pistons, used in the Ford Modular V8 engines, were chosen for this engine. The Mahle piston has a flat top and the other two have a 3 cm³ dish, as can be seen in figure 7 above.
This means that if material is machined off the top of the pistons to make them shorter and flat (to reach the desired high compression ratios), the most weight reduction is achieved with the Mahle pistons. Also, since the Mahle pistons are stronger than the other pistons, less material is needed at the top, especially for the lean burn, low RPM usage they will be subjected to. The pistons were machined to 337.3 grams (down from 388.2 grams). One of the machined pistons used in the engine can be seen above on the left part of figure 8, next to one of extra original Mahle pistons, unmachined.

The camshaft used was a modified 1989 Nissan Pathfinder Z24i camshaft. It was reground to provide approximately half of the original lift. Since this engine’s operating speed will be a low, constant 1800 RPM, the stock, higher-lift cam would cause the air to flow into the cylinders much more slowly, compared to the modified cam profile. The higher stock cam lift also requires higher spring pressures along with stronger retainers, all causing more engine friction and a lower brake thermal efficiency. The reground cam produces a maximum lift of 0.2022 inches at 111 degrees after top dead center, which is
still large enough to keep the air velocities well below a Mach index number of 0.5, and allows for the use of lighter moving parts in the cylinder head.

The camshaft was then further modified to be used with the KA24 block and crankshaft. A KA24 cam sprocket is being used on a Z24 camshaft, on a Z20 cylinder head, on a KA24 block, so the stock Z24 sprocket location had to be changed for use with the cylinder block and cylinder head desired. The cam sprocket sits 0.090 inches too far toward the front of the engine (the Z24 camshaft is 0.090 inches too long). Therefore, 0.090 inches was machined off of the camshaft, because the sprocket is too thin to be machined to fit. The problem with machining the cam was that the pin needs to remain where it is so the sprocket can be correctly phased with the cam. This was done with a prototype CNC machine. A lip of metal to fit into the center hole in the sprocket was also
left unmachined to account for the difference in inner diameter of this cam compared to the KA24 cam. The sprocket fits the cam with a light press fit. The ground and machined camshaft can be seen on the previous page in figure 9, next to another Z24 camshaft on the right, with more traditional lobe heights and the stock length. Notice the lip left on the machined camshaft, so it can fit the key way in the KA24 sprocket shown on the lower left. The valve lift produced by the cam profile versus crank angle can be seen above in figure 10.

![Valve Lift vs. Crank Angle](image)

The valve lift versus crank angle data was used to calculate the minimum necessary spring pressures needed to prevent valve float. This was done by taking two derivatives of the smoothed curve fit (with a cubic spline) of the lift data. Since the RPM is constant, the results were easily converted from the angle domain into the time domain, then multiplied by the total mass of the moving parts (including the effective mass of the moment of inertia of the rocker arms) to get the force curve. This came out to be 22.5 pounds of force for the 204.3 grams of effective mass. Springs that provided the
necessary force at the maximum lift, along with the necessary force at zero lift to crush any potential deposits that would otherwise keep the valves from closing, were chosen rather conservatively. The data curves were also smoothed very conservatively. The

Melling VS-920 springs, pictured above in figure 11, were the chosen springs for the engine. They are a bit stronger than the stock inner springs of the Nissan Z20, which are pictured on the left part of figure 12 above (the outer Z20 springs are on the right). The stock retainers were then machined to fit the new springs. The original and modified retainers can be seen below in figure 13.
The stock adjusters are machined spherically at the end (by the original manufacturer) where they come into contact with the valve stems. This becomes a problem if the head is removed or if the adjusters are rotated, even the slightest bit. When placed back, the wear that was caused on the adjuster is now out of its "track" and this will cause grooves to be worn on the end of the valve stems. Therefore, instead of using the stock adjusters, the articulated Porsche Rhino-Feet adjusters were used, which distribute the load over the entire end of the valve stem. The surface that comes into contact with the valve stem is flat and slides across the end of the valve stem. A stock adjuster, along with its wear pattern, can be seen above in figure 14. Figure 15 shows a Rhino-Foot adjuster, like the ones that were used.

The stock rocker arms were modified so that they could be used with the Rhino-Feet adjusters. The articulation needs lubrication, so oil holes had to be drilled in the rocker arms, as can be seen on the next page in figure 16.

The overall friction of the engine was reduced by reducing valve spring pressures (45 lbs versus 164 lbs), using lighter connecting rods, lighter pistons with shortened skirts, shorter valve lifts (0.202” versus 0.380”), Rhino Feet rocker arm adjusters, and a
low friction ring package. An assembled rocker arm, adjuster, retainer, and spring can be seen below in figure 17. The CAD drawing of the custom made 6.6 inch long connecting rods, weighing only 497 grams, can be seen on the next page in figure 18, followed by a picture of one them before being installed in the engine in figure 19.
Before a second spark plug is added to each combustion chamber, data must be acquired with one spark plug to create a baseline for comparison. Once the hole is made for the second spark plug, unburnt hydrocarbon emissions will increase because there will be extra crevices in the chamber to trap unburnt fuel. Having data from the same engine with a single spark plug will determine the real difference for potential future production, as opposed to just not firing the second spark plug. Once the feasibility of this engine has been determined, a future project may demonstrate the actual reduction in time losses via more ignition sites.

**Potential Problems in Need of Testing for Feasibility**

This study is designed to determine if this hybrid engine, which is the first of its kind, can function in a practical manner. Many different problems may come up during testing of a new engine, but the following four are addressed in this feasibility analysis.

The first, which may show up during the first few rotations of the engine’s crankshaft, is the integrity of the camshaft lobes. As mentioned earlier, the valve lifts are significantly lower (47% less) than stock. This means that the cam lobes were ground
down past their hardened surface, making them relatively soft and more susceptible to wear from the rocker arm pushing down on them with the force of the springs.

Iskederian, a well-known and respected camshaft manufacturer, refused to grind the cam lobes to the specifications desired for this engine, for this very reason. Crane, another well-known and respected camshaft manufacturer, only agreed to do this work when they were assured that the cam would be used in a research engine, where failure would be categorized as educational and caused by design error as opposed to faulty machine work or bad decisions made on their part. Additionally, since the lobes are shorter, they may not dip into the oil that sits under the camshafts, making them inadequately lubricated, adding to the problem. If the softer surface isn’t hard enough to withstand the forces even with the weaker springs, and/or the lobes don’t get adequately lubricated, then a different set of cam shafts will need to be used. The starting load divided by the projected area of a journal bearing is required to be less than 300 psi to prevent premature wear. A cam lobe should have a higher limit because of the approach angle between the cam lobe and the surface of the rocker arm is much greater than that of the journal bearings, so a lot more oil is available between the surfaces. However, conservatively using the journal bearing standards we obtain:

\[
\frac{W_{st}}{lD} = \frac{61 \text{ lbs}}{(0.5 \text{ in})(1.2 \text{ in})} = 102 \text{ psi} \leq 300 \text{ psi}
\]

where \( W_{st} \) is the starting load on the bearing (cam lobe), \( l \) is the length of the bearing (or width of the rocker arm contact surface), and \( D \) is the diameter of the bearing (cam lobe). The starting load was calculated by multiplying the maximum spring force (45 lbs) by the rocker arm ratio, which was conservatively estimated as the maximum lift in the stock
cam (0.38 inches) divided by the measured lobe lift in the stock cam (0.28 inches). Equation 16 on the previous page shows that this engine’s design lies well under the lubrication limit of the cams, therefore it was hoped to avoid valvetrain lubrication problems. [11]

The next potential problem is the temperature distribution in the engine. The KA and Z family of engines have the coolant flowing in different directions. This may create a situation in which the coolant goes through one side of the block, into the cylinder head, and out the same side of the cylinder head, leaving one side of the block, or even one side of the whole engine with insufficient coolant flow. To monitor this, four thermocouples have been installed in the coolant passageways, in four separate sections of the engine: in the coolant inlet, the coolant outlet, on the right side of the block, and on the left side of the block. As soon as the engine starts warming up any potential temperature gradient will be clearly visible, and the engine can be turned off long before any significant deformation or engine damage can occur. If this is a prevalent problem, the cylinder block, cylinder head, and/or the cylinder head gasket will need to be machined to allow an even flow of coolant throughout the engine.

Another potential problem is the fuel distribution in each cylinder. A modified stock intake manifold is being used, with the coolant passages removed to be optimized for gaseous fuels. With no heat being added to the intake manifold, the temperature of the charge is reduced, raising the ratio of specific heats, volumetric efficiency, and the knock limit. However, there is no pre-chamber in the intake manifold to fully mix the air and fuel, as in the Mazda 1.6L generator set engine, so there might be cylinder-to-cylinder
air/fuel ratio variations that lower efficiency and require a richer mixture to be used. If this is prevalent, a new intake manifold will have to be designed to fix the problem. [12]

Finally, the oil pump may not provide the necessary oil pressure to keep the engine lubricated. Restrictions on oil flow might be different on the KA engine relative to the Z engine, requiring a different pump on each engine, so the oil pressure will be closely monitored with a pressure sensor installed near the oil filter. If insufficient oil pressure is created then a different oil pump will have to be acquired.
Chapter 3

Engine Installation and Setup

Once all the engine parts were acquired and/or modified (or created) and assembled, the engine was bolted onto a steel stand. After, all the stand parts were wire brushed and painted to avoid any corrosion problems. This included the custom made engine mounts, which were machined and welded to accommodate this new engine. Then a Superflow water brake dynamometer was installed on a 1981 Nissan 510 bell housing, and connected to the crankshaft.

This dynamometer has two piston diaphragms connected to it filled with oil, which produce one psi of pressure in the oil tubing for every foot pound of torque that is exerted on the dynamometer. This makes it easy to read how much torque the engine is producing, allowing for an easy calculation of power, using equation 17 below. The total

\[
Power_{in\, HP} = \frac{(Torque_{in\, ft-lbs})(RPM)}{5252}
\]

Equation 17

amount of air going into the engine is calculated by taking the pressure differential across the square edge flow meter in the inlet system, measured with an inclined water manometer. The amount of energy going into the engine is easily calculated with the amount of fuel that is being permitted into the engine, calculated with the air/fuel ratio measured by the lambda meter in the exhaust pipe, and the lower heating value of pure propane (19944 Btu/lb). Brake thermal efficiency can then be calculated by dividing the power output by the rate of total fuel energy going into the engine, which takes into consideration all the losses not accounted by equations 14 and 15. While this is being done, initially for very low loads and then for increasingly higher loads, all of the
different temperatures in the engine will be closely monitored and recorded, along with the oil pressure and the exhaust temperature at the exhaust port of each of the four cylinders, to analyze cylinder-to-cylinder air/fuel distribution. If there are problems at light loads, they will need to be addressed before proceeding on to higher loads, to avoid damaging the engine. Figure 20 below shows the right side of the engine, with the dynamometer installed and the thermocouples visible that are connected to the right side block coolant passageway and the water inlet. [2]

![Figure 20](image)

**Data and Data Analysis**

The Nissan ZKA26 engine was very easily started once it was appropriately set up. The oil pressure initially was 60 psi at 1500 RPM. As the oil heated up and lowered
in viscosity it went no lower than 40 psi. This indicates that the oil pump is a good match for the engine, as one that produces less than 30 psi will allow the engine to damage itself because of lack of lubrication, and one that produces a much higher oil pressure would simply make the engine less efficient from the added friction and energy required for the extra pumping.

The problem which was first apparent, seconds after the engine started running, was the cylinder-to-cylinder air/fuel ratio distribution. A good distribution is represented by no more than a few degrees difference between the exhaust port temperatures from each cylinder. The recorded exhaust port temperatures can be seen below in table 1, for each minute the engine was running. The data represents an 80°F difference between cylinders after the engine was running for one minute. This represents bad distribution, meaning a different intake manifold must be used in order to achieve competent efficiencies and to meet emission standards.

The next problem was that the temperature difference between the coolant in each side of the block increased rapidly, noticeable during operation. After only two minutes running, there was a 45.3°F difference between each side of the block, and a 65.2°F difference at four minutes. The increase in magnitude of the block’s temperature gradient

<table>
<thead>
<tr>
<th>time (minutes)</th>
<th>exhaust port 1 (°F)</th>
<th>exhaust port 2 (°F)</th>
<th>exhaust port 3 (°F)</th>
<th>exhaust port 4 (°F)</th>
<th>water inlet (°F)</th>
<th>right block water (°F)</th>
<th>left block water (°F)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
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<tr>
<td>3</td>
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<td>1213</td>
<td>1214</td>
<td>1187</td>
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<td>159.7</td>
<td>106.3</td>
</tr>
<tr>
<td>4</td>
<td>1217</td>
<td>1213</td>
<td>1193</td>
<td>117.3</td>
<td>193.4</td>
<td>128.2</td>
<td></td>
</tr>
</tbody>
</table>

Table 1
can be seen in figure 21 below. About thirty seconds later a rapid increase in water inlet temperature was observed, and so the engine was shut off at 208°F. Some of the water started to boil a moment after the engine was turned off. All of this happened with no

![Figure 21](image)

more than 20 ft•lbf of torque being exerted on the crankshaft, which is approximately 17% load. This indicates that the coolant flow in the engine must be changed, by at least modifying the head gasket and the intake manifold.

Finally, after the engine was turned off and allowed to cool enough that it could be safely touched, the rocker arm cover was removed for inspection of the valve train. Wear typically occurs following initial startup during the first minutes of operation when the cam lobe surface has its most prominent irregularities. The camshaft lobes had no visible wear on them at all, as can be seen in figure 22 on the next page. This indicates that despite them being shorter and softer, they were able to sufficiently dip into the oil
underneath them, and the 73% weaker springs (45 lbs compared to 164 lbs stock) were weak enough that they did not push the rocker arms hard enough into the cam lobes to penetrate through the oil film and damage them.

Figure 22

Extended Analysis and the Engine’s Future

One of the two remaining difficulties of this engine is the large temperature gradient in the coolant passageways. This is the only problem that makes further operation of the engine destructive, as the bad cylinder-to-cylinder distributions don’t physically hurt the engine. Before investigating ways to solve this problem, an analysis of the two engines from which this one was constructed from is useful, as both of those engines function without cooling problems, even at much higher loads and RPM values.

The water pump on the Z family of engines pumps the coolant through the right side of the engine (passenger side in the USA), up into the cylinder head, and out through the left side (driver side in the USA) of the block. The KA family of engines, however, is designed such that the coolant flows in the opposite direction as the Z family of engines.
The KA engine’s water pump sends coolant through the left side of the block, up into the cylinder head, and out through the right side. Because the ZKA engine uses the block from a KA engine, the coolant flows first through the left side of the block, then into the cylinder head. This Z engine cylinder head is designed to have the coolant come in through the right side and out the left, so the water jackets on the right side are more restricted by its cylinder head gasket, and the water jackets on the left side are less restricted by the cylinder head gasket. This would indicate that the gasket should be modified, or a new gasket should be created so that the right side coolant passageways are larger and the left side passageways are smaller. It should be noted that two years after Nissan started producing the dual spark plug Z engines, they changed the left side spark plugs to a higher heat range spark plug. The fact that they would produce a dual spark plug engine with two different spark plugs indicates that there was an inherent temperature gradient in the engine. If the factory engine already had a design problem with its temperature distribution, it would indicate that there might be a limit as to how uniform the temperature distribution could get with the Z cylinder head when designed for multiple RPMs. Since this research effort is for only one RPM (1800), a uniform temperature distribution should be attained with the proper modifications to the coolant passageways.

In addition to the restrictions in the coolant passageways in the head gasket, this engine’s intake manifold had all of the water jackets removed from it, because it will only be used with gaseous fuel. Automobiles use liquid fuels, which need to be vaporized before going into the cylinders, a process that requires or is greatly aided by heat being added to the fuel during or right before the vaporization process. When gaseous fuels are
used, heat addition to the fuel is unnecessary and only reduces efficiency, as the initial temperature of the mixture becomes higher resulting in a smaller $\Delta T$ in the thermodynamic cycle. However, when the water jackets are removed from the intake manifold, an exit pathway from the cylinder head is removed and a stagnation area is caused where the intake manifold meets with the cylinder head’s coolant passageways, restricting some of the flow that would be going to the right side of the engine, the side that now gets too hot. The intake manifold can be seen on figures 23 and 24 on the next page, before and after it was modified, respectively. When a new intake manifold is created, it should take this into consideration and perhaps allow coolant flow from the cylinder head to go to the engine’s coolant outlet. This will prevent the unwanted heat to be added to the fuel/air mixture while preventing coolant flow necessary for the right side of the engine.

The coolant passageways in the intake manifold will be one of the features of the new intake manifold that will optimize the performance and durability of the engine. The next major aspect of this manifold will be its air and fuel mixing capabilities. The stock manifold has poor air and fuel mixing capabilities, which cause the poor cylinder-to-cylinder distributions of air and fuel, limiting how lean the engine can run and how close the engine could run to the optimal air/fuel ratio. A new manifold should be designed to get the air/fuel mixture as close to homogeneous as practical before the mixture reaches the intake ports. One way this can be done is with a pre-chamber, as it was done in Design of 1.6 Liter Genset Engine. [12]
Chapter 4

Conclusions

1. Hybrid engine Nissan ZKA26 is not feasible for use with a stock head gasket.

2. This engine needs redesigned cylinder head gasket to accommodate sufficient coolant flow on the right side (passenger side in the USA) of the block, to prevent cylinder distortion and engine failure.

3. The stock intake manifold with coolant passageways machined off is incapable of producing enough fuel/air mixing to obtain an acceptable cylinder-to-cylinder distribution of air and fuel.

4. The stock oil pump provides sufficient oil pressure to keep engine safely lubricated during its intended operation.

5. The stock valvetrain springs that exert 164 lbs on the rocker arms may be replaced with ones that only exert 45 lbs (73% weaker) with no valve float for a generator set application with a constant engine speed of 1800 RPM.

6. The stock camshaft that produces 0.380 inches of lift may be grinded down to produce only 0.202 inches of lift without lubrication problems and without unhardened metal wear problems when used with 45 lbs springs in a generator set application with a constant engine speed of 1800 RPM.
 References


Appendix

Superflow water brake dynamometer:
Inlet system:
Cooling tower:
Installed gauges:
Modified oil pan with insert for oil temperature sensor:
Welded and painted exhaust headers:
Stainless steel plate and gaskets for exhaust port (to insert thermocouples to measure exhaust gas temperatures):

CNC machined stainless steel plates:
The 1981 Nissan 510 bell housing:

New Z20 cylinder head flow vs. lift (measured in flowbench):

<table>
<thead>
<tr>
<th>Lift in inches</th>
<th>Flow in CFM</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
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<tr>
<td>0.15</td>
<td>73.675</td>
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<tr>
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