Proceedings of the Fourth International Conference
On Energy Sustainability ES2010 May 17-22, 2010, Phoenix, Arizona
Paper ES2010-90058, revised

A COMPACT EXTERNAL COMBUSTION ENGINE WITH HIGH PART-LOAD EFFICIENCY

Robert Bourque, Ph.D., P.E.
Los Alamos, New Mexico
bob@rfbourque.net

ABSTRACT

An external combustion engine design using steam is described which has good efficiency at full power and even better efficiency at the low power settings common for passenger vehicles. The engine is compact with low weight per unit power. All of its components fit in the engine compartment of a front-wheel drive vehicle despite the space occupied by the transaxle. It readily fits in a rear-drive vehicle.

Calculated net efficiencies, after accounting for all losses, range, depending on engine size, from 28-32% at full power increasing to 33-36% at normal road power settings. A two-stage burner, 100% excess air, and combustion temperature below 1500°C assure complete combustion of the fuel and negligible NOx. The engine can burn a variety of fuels and fuel mixes, which should encourage the development of new fuels.

Extensive software has been written that calculates full power and part-load energy balances, structural analysis and heat transfer, and performance in specified vehicles including using SAE driving cycles. Engines have been sized from 30 to 3200 hp. In general, fuel consumption should be at least 1.5 times lower than gasoline engines and about the same as diesels operating at low to moderate load settings. From this extensive analysis, a prototype, when built, should perform as expected.

BACKGROUND

Steam engines in automobiles were eclipsed in the mid-1920's by the simpler, more compact, and easier-starting internal combustion (IC) engines (1). This occurred even though IC engines required refined fuel, while steam engines did not, and complex transmissions for good performance.

Over the years, attempts have been made to revive steam engines by hobbyists, corporations, and governments (2,3,4). The efforts had some success in improving performance. However, the large improvement that would be required to retire the IC engine has not happened.

MOTIVATIONS FOR THIS ENGINE

There is an urgent need for a vehicle engine that can efficiently and cleanly burn other fuels besides refined petroleum. Only then will the incentive arise to develop such fuels on a commercial scale. The Age of Petroleum will end in about 40 years at current consumption (5,6,7), unless new deposits are discovered or consumption is reduced.

Petroleum used in transportation accounts for 31% of the carbon dioxide emissions in the US (8). Engines with higher efficiency under normal operation could reduce that. Road vehicles generally run at only a small fraction of their peak power. The extra power is needed for acceleration and grade climbing. At low loads internal combustion engines have throttling and pumping losses that reduce efficiency. Steam engines avoid this, and the engine presented here actually has higher efficiency at part load than at full power.

The fuel for external combustion engines needs less processing and refining. This ultimately reduces total energy use and overall emissions from the source (e.g., the oil well) to the tailpipe. There are other options such as plug-in hybrids or pure electric vehicles. But up to 80% of the fossil-fueled electricity in the US is generated by burning coal (8). It can be shown (9) that charging vehicle batteries with electricity from coal plants actually increases carbon emissions.

There is also the problem of localized pollution due to nitrous oxides, unburned fuel emissions, and carbon monoxide. This is especially serious in urban areas, and ways to reduce it must be sought.

Considering all of the above, external combustion engines appear especially attractive. That appeal is even greater for the engine presented here, as will be shown in detail below.

In external combustion engines the working fluid and combustion gases are separate and can be individually optimized. In Rankine-type external combustion engines, the fluid being recompressed is liquid rather than gas. This reduces the work of recompression, which increases net output and thereby reduces the peak temperature needed for a given efficiency. Many different working fluids have been explored by others (10) and, to a lesser extent, by the author. But, *all things considered*, steam turns out to be the best.

REQUIREMENTS AND CONSTRAINTS

In order to maximize net, i.e. useful, efficiency for any power cycle, the average steam heat addition temperature must be maximized and the average heat rejection temperature minimized. The key word here is *average*. Peak temperatures are useful only in as much as they increase the average.

Maximum temperatures are limited by materials. These ought to have the following properties:

- Capable of carrying the required internal stress at maximum operating temperature
- Forgiving (i.e. ductile)
- Rated by ASME Boiler and Pressure Vessel Code
- Affordable
- Available in large quantities
- Capable of being mass-produced and joined

To be sure, exotic materials can be considered, but their use should be minimized.

Minimum temperatures are limited by the medium to which heat is rejected. Here it is ambient air, taken for design purposes to be 75°F (24°C). There are also limits to the size of the condenser. This requires steam heat rejection temperatures to be much higher than the ambient air.

It is also necessary to maximize expander isentropic and mechanical efficiencies, and minimize parasitic losses from pumps, air blowers, flow pressure drops, and heat conduction. As discussed below, these have been done here.

In order to be competitive, a steam engine for automotive use must achieve the following:

- In-use efficiency must be significantly greater than a gasoline engine
- It must be truly multifuel and capable of accepting any fuel mixture
- It must fit entirely in the engine compartment with access for maintenance
- It must condense *all* the steam using ambient air
- The condenser must fit into the usual radiator space
- The combustion temperature must stay below the level where NOx forms
- The fuel must be completely burned
- There must be rapid response to power changes
- It must be fairly quick starting

In summary, for an automotive application with all its constraints, it is necessary to “pull out all the stops” and do everything possible to achieve the above requirements.

PROGRESS

This project began around 1973. A computer code was started in 1977, written in Fortran 77. Initially intended to analyze the thermodynamic cycle, it grew, as codes tend to do, so that today it is over 6500 lines and has 117 input variables.

The code performs the following functions:

- One full power and 16 part-load energy balances
- Structural analysis including use of the ASME Code using material properties at temperature
- Expander design including piston analysis, bearing sizing, and internal friction, heat and valve losses
- Heat exchanger heat transfer coefficients, core sizing, metal temperatures and stresses, steam and air pressure drops, and air pumping power
- Performance in specified vehicles including acceleration, grade climbing, and fuel consumption at steady-speed and also over urban, suburban, and 55 and 70 mph highway driving cycles specified by the Society of Automotive Engineers (SAE).

In addition, 2-D layouts have been made for several engine sizes. A patent application was filed in 1976 and a US Patent (11) was awarded in 1978.

PROTOYPE DEVELOPMENT

An operating prototype has *not* been built. That is the current objective. The rest of this paper describes the considerable analytical and design work that has been done. This provides the extensive foundation needed for a minimum risk, successful prototype.

DESCRIPTION OF THE BOURQUE CYCLE

Figure 1 shows the Bourque Cycle in block diagram. The air side flow path is shown in green and the steam side in blue. **Figure 2**, taken from the original patent, shows the cycle in the temperature-entropy plane. The left side of **Figure 2** shows the full-power steam cycle and an approximate isobar showing (not to scale) the air side. The right side of **Figure 2** shows the steam cycle only with a

Throttle Valve between the Steam Generator and the Expander (path **G-G′**).

On the steam side in **Figures 1 and 2**, starting with the Feedpump, water enters at **D** from the Condenser and is pressurized to the range of 2800-3500 psia (19-26 MPa) depending on application; but is *not* preheated. The feedwater then enters the first stage of the Steam Generator - the Boiler (also called the Water Heater) - that has finned tubes, at **E**. Here the water is brought to vapor and in this case is raised to about 800°F (427°C) at **F**. Limiting this temperature at this point allows use of lower-cost chrome-moly steels in this section.

The steam then enters the bare-tubed Superheater where it is raised to about 1100°F (593°C) at **G**. It then passes through a Throttle Valve, shown on right side of **Figure 2** (more on this later) and then to the Expander at **G′**. The Expander has four stages with Reheats between each. The multiple Reheats help to maximize average heat addition temperature.

When the steam leaves the last Expander stage at **N** it is still highly superheated. This energy is exploited in the Regenerator by passing it to the incoming air to the Burner. (The figure shows a state **O**, which is hotter than **N**. This was an earlier attempt to pick up friction heat, but has since been abandoned. States **N** and **O** now coincide.)

In the Regenerator the steam is cooled to saturation and even slightly condensed at **P**. Eliminating dry steam, which has poor heat transfer, in the Condenser makes it much easier to design that heat exchanger to fit in a vehicle. The steam is then condensed, and slightly subcooled at **D**.

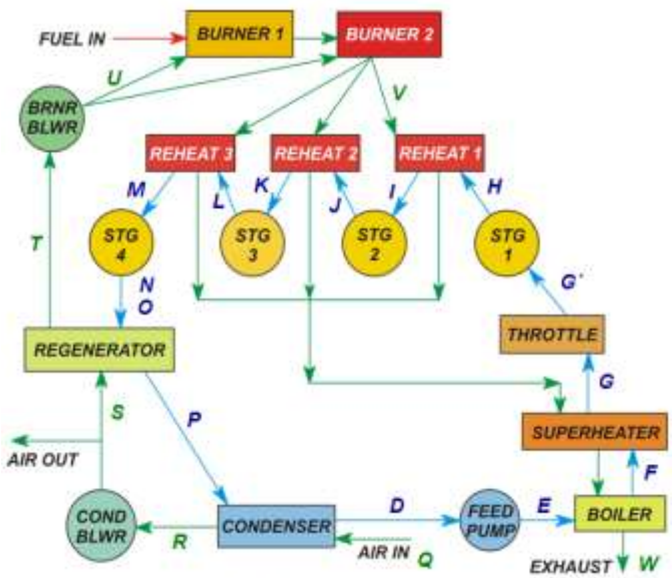

Figure 1. Block diagram of Bourque Cycle

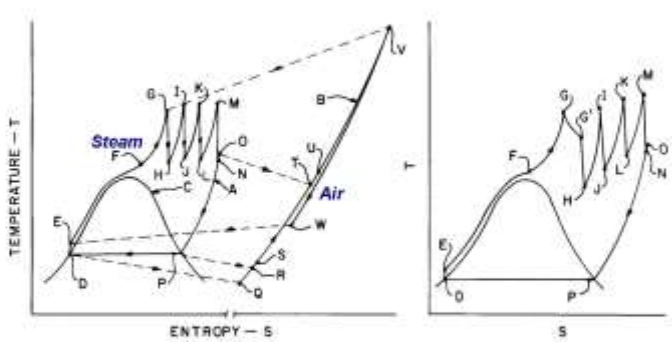

Figure 2. Cycle shown in temperature-entropy planes

The condensing pressure at full power is quite high, typically 50 psia (0.34 MPa), giving a full-power condensing temperature of around 280ºF (138ºC). This provides enough temperature difference with the incoming air to condense all the steam with a rather compact heat exchanger. The high Expander exhaust pressure also reduces the size of the last expansion stage; and it reduces the pressure drops in the Regenerator and last Reheat to acceptable levels. Condensing pressure and temperature are lower at part load.

On the air side, seen in green in **Figure 1** and on the left side of **Figure 2** as an isobar, outside air enters the Condenser at **Q** and leaves it heated to **R**. It is drawn through the Condenser Blower and heated slightly to **S**. It is then substantially heated in the Regenerator by the desuperheating steam to **T** and slightly more to about 600°F (316°C) by the Burner Blower to **U**. It then enters the Combustor where adding fuel heats it to its final combustion temperature of 2700°F (1482°C) at **V**. It passes through the Steam Generator and out the exhaust at **W**.

FEATURES OF THE CYCLE

The somewhat unusual features of this Cycle, particularly with respect to earlier engines, warrant some discussion and justification. This is presented below.

There are low expansion ratios per stage, about 2.3 to one. This has three advantages:

1. High Expander adiabatic efficiency. Losses tend to come from inlet and exhaust turbulence, which is minimized because of more relaxed valve operation, and from thermal losses, which are much less due to reduced temperature swings. The actual expansion is low velocity and essentially isentropic.
2. High ratio of Mean Effective Pressure to Peak Pressure. This reduces loads on bearings and other drivetrain components, and should be smoother. In general, it allows for higher Expander speeds and smaller bearings, cranks, and connecting rods.
3. High pressure is confined to small sizes. The Expander cylinder size depends on the volume of its *exhaust* steam. Smaller size means lighter weight and greater safety.

Note that total expansion ratio across all four stages is very high, about 26 to one. Therefore the total work per unit mass extracted from the steam is also high.

The Expander exhaust steam preheats air to the Burner. Less fuel is therefore needed to reach a given combustion temperature. And there are other advantages:

1. It makes it easier to burn difficult fuels. This engine is intended to be truly multifuel.
2. There is 100% excess air, which assures complete combustion, and combustion temperature is controlled to stay below that which forms nirtous oxide.
3. It preheats the fuel so that it can ignite more readily.
4. As mentioned, the steam entering the Condenser is wet, reducing Condenser core size.

5. Net efficiency is high even though the condensing temperature is quite high.
6. Sufficient Regenerator and Steam Generator driving temperature differences arise automatically from the relative air and steam mass flows.
7. There is *no* feedwater preheating. It is replaced by the burner air preheating, which is more effective. The exhaust gas is therefore at the minimum possible temperature giving the minimum heat loss out the tailpipe.
8. At very low load settings, water vapor in the combustion products condenses in the Boiler, allowing use of the fuel higher heating value. If the feedwater were preheated, an exhaust gas recuperator would be needed or there would be insignificant, if any, efficiency gain.

There is a simple Throttle Valve between the Steam Generator and the Expander. Many steam engines use variable cutoff for control, to maximize expansion ratio at part-load. But the advantages of a simple Throttle Valve are:

1. All downstream components are at reduced pressure during normal operation.
2. The lower pressure carries all the way to the Condenser, resulting in lower heat rejection temperature. This increases efficiency at part load.
3. Steam Generator pressure is constant. There is therefore a reserve of pressure just behind the Expander for rapid response. Also, the Steam Generator is not subject to pressure cycling that could cause fatigue failure.
4. Heat addition in the Reheats increases, which also increases efficiency. The increased reheat temperatures are tolerable because the reduced pressure at part load reduces metal stresses. This stress reduction is greater than the reduction in metal strength.
5. The loss due to throttling is only that from liquid recompression. This is very small, is a unique advantage of the Rankine Cycle.

There is a two-stage Combustor. This is not part of the invention but is a worthwhile feature. As discussed later, it has been shown to improve combustion. The fuel first mixes with a small amount of the preheated air, giving a very rich air/fuel ratio. There it is heated, vaporized, and partially burned. Then it is mixed with about 100% excess air for very complete combustion at temperatures below that where NOx forms.

THE COMPLEXITY ISSUE

One may argue that this cycle is too complex for vehicle applications. But the complexity is needed to achieve the desired high efficiency. In general, high performance of any system usually requires complexity. In mass production, complexity is less of an issue than total weight, type of materials, and type of fabrication. As seen below, the result is very good performance and low weight and bulk volume.

It is believed that this complexity does not reduce reliability. In fact, reliability may be greater because of the smaller heat exchangers and smaller high-pressure components.

FUEL REQUIREMENTS

The fuel requirements are simply that it can be stored on the vehicle and delivered in metered fashion to the Burner. Beyond that, almost any combustible material will work. The engine provides a way to bypass nearly all fuel preparation processes. Refining requirements are minimal. Cetane or octane ratings, and perhaps hydrogenation, are not needed. This increases "oil well-to-tailpipe" efficiency beyond what the engine itself delivers.

In addition, the clean burning and high part-load efficiency reduce not only NOx, CO and unburned hydrocarbons, but also carbon dioxide emissions. Total fuel supply requirements are reduced; and the development of alternative fuels, such as biomass, methanol, biodiesel, etc, are encouraged because an engine would existed to burn them.

Multiple fuels can be used in a single tankful; and they do not have to be miscible. As discussed later, the Control System operates by feedback; and fuel and air flow are adjusted to maintain fixed combustion and superheater outlet temperatures as fuels with different heating values pass through. With the right delivery system, powder/liquid slurries or pressurized gases could also be used.

FIRST EXAMPLE ENGINE IN A VEHICLE

Several examples are given below of engines in specified vehicles. These vehicles were used in the software written by the author to estimate performance. They were specified either by weight and coastdown runs performed by the author to get the road power curve, or by weight, frontal area, and drag coefficient that was used in a standard formula.

Figure 3 shows the first example, a 140 hp (104 kW) engine, as it would be installed in the author's 4200 lb (1900 kg) minivan. Shown are the two Burners, Steam Generator, Expander, Regenerator, two Blowers, and Condenser. The Expander, which is discussed in detail later, is a tilted 'V' configuration, making it look odd in the top view. Also shown is the outline of the existing transaxle. This presented a challenge because it takes up considerable engine compartment space. (With rear-wheel drive it is behind the engine compartment and out of the way.) Nevertheless, as seen, the front-wheel-drive engine does fit.

One interesting item, shown in tan at the bottom, is a braking blower. To compensate for lack of compression braking, an air blower can be attached to the Expander. Air flow through it is zero until the brakes are applied. Then air is admitted to provide the equivalent drag as compression braking.

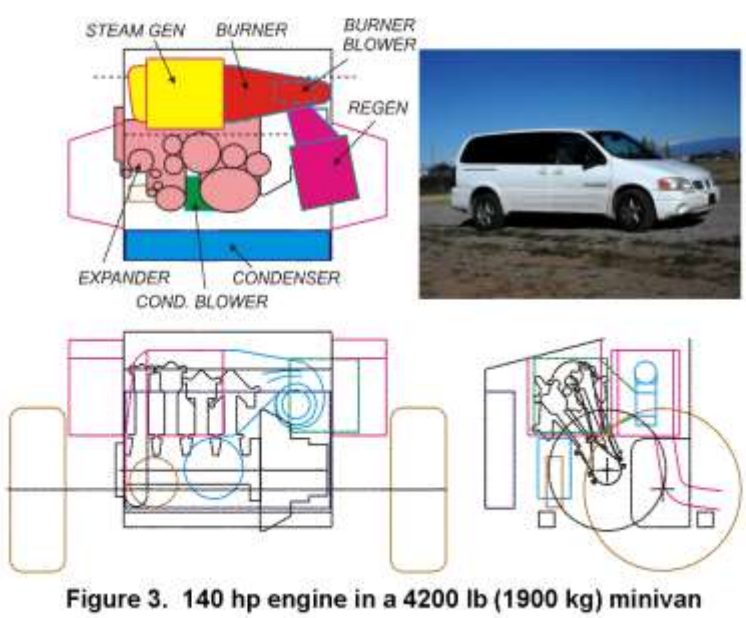

Figure 3. 140 hp engine in a 4200 lb (1900 kg) minivan

Figure 4 shows the calculated power and torque for this engine at wide-open throttle. The very flat torque curve permits reduced peak power for a given overall performance. The slight dropoff is due to increasing pressure drops and mechanical friction as engine speed is increased.

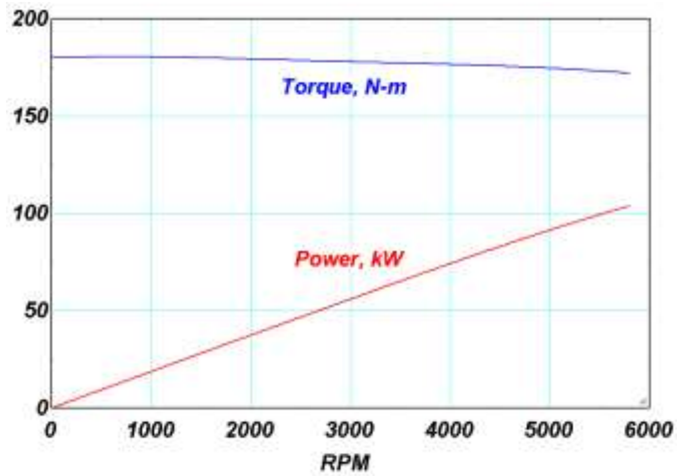

Figure 4. 140 hp (104 kW) engine power and torque

Figure 5 shows calculated net efficiency and specific fuel consumption (SFC) with gasoline. Although any fuel can be used, these plots have been made using the most familiar fuels: gasoline for cars, diesel for trucks, buses and locomotives.

The right ends of the curves in **Figure 5** indicate power at wide-open throttle at that rpm. As seen, efficiency is about 28% at maximum full power. Although quite good, it gets even better: It increases at lower speeds and throttle settings, reaching 32% at half speed and 33% at quarter speed. Specific fuel consumption is between 0.42 and 0.47 lb/hp-hr (256-286 gm/kW-hr) at normal load settings. The odd jump in SFC at 1450 rpm appears to be from software convergence tolerance.

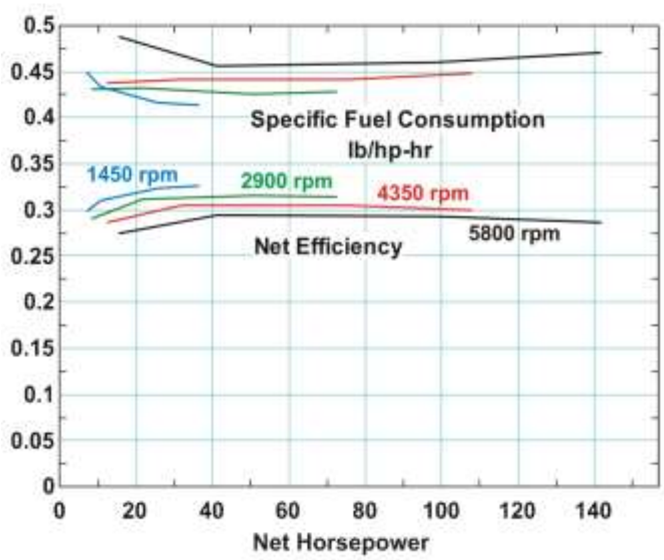

Figure 5. Calculated 140 hp engine net efficiency and SFC

These results are compared in **Figure 6** to IC engines, taken from a variety of sources (12,13,14). The curves show SFC for 50% rpm. This time the steam engine is installed in a 3200 lb passenger car. In that vehicle the engine needs to supply 25 kW or less at steady speed. The 10.8 kW shows how little power is needed to propel it on a level road at 50 mph. Note that the two gas engines, shown in red and blue, have around 50% higher SFC in the normal operating range (lower SFC is better). Only the diesel matches the steam engine. This suggests that passenger vehicles should have about 50% higher fuel economy if their gas engine were replaced with the steam engine. Replacing diesel engines should give about the same fuel economy but would have cleaner burning, multifuel capability, and quieter operation.

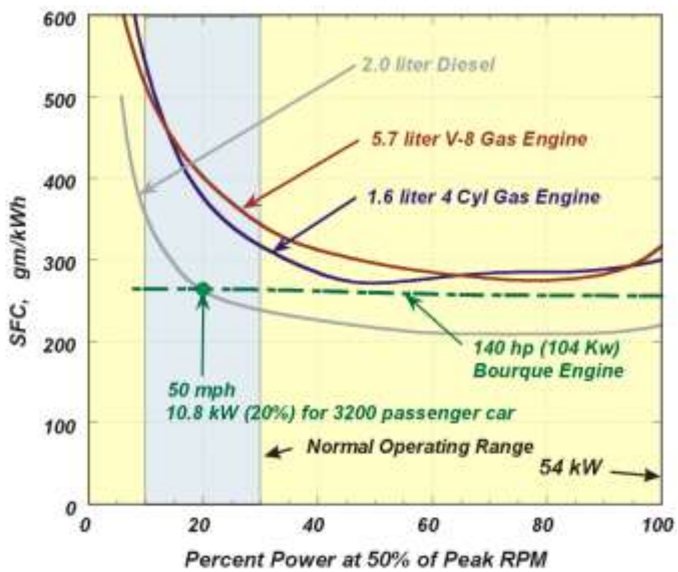

Figure 6. Comparison with other engines

Figure 7 shows calculated steady-speed performance of the 140 hp engine in the minivan. The road power curve was determined from extensive coastdown runs. As seen, miles per gallon is very sensitive to speed, which reflects the fact that engine efficiency increases with decreasing load.

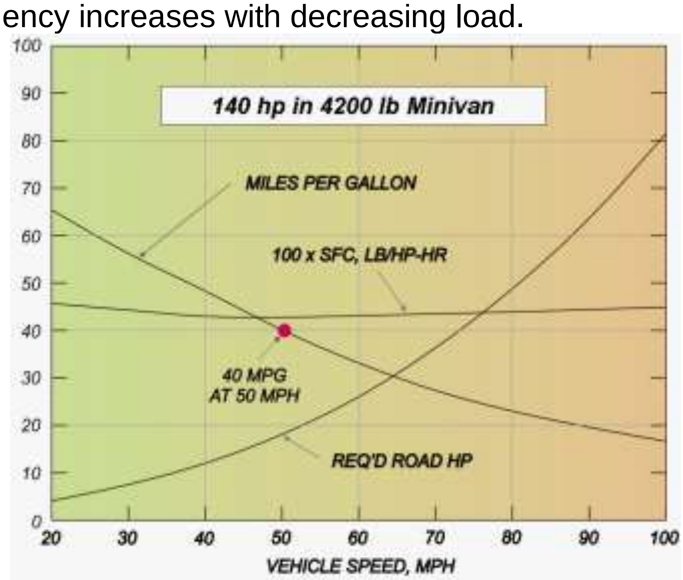

Figure 7. Calculated steady speed performance

Of course, steady speed results do not reflect actual driving. For this reason, the software written by the author includes the SAE Standard Driving Cycles. These cycles include starts and stops, and acceleration to speed, as well as periods of steady speed. **Table 1** shows calculated fuel economy, taken from the code output, over the four SAE Driving Cycles. As seen, fuel economy is very high even under these conditions. These values compare very favorably with the nominal 22-25 mpg reported by owners (15) and by the author.

Table 1. 140 hp engine in 4200 lb Minivan

FUEL ECONOMY OVER STANDARD DRIVING CYCLES
SAE ROAD TEST PROCEDURE - SAE J1082 SEP80

DRIVING CYCLE	KM/LITER	MPG	MAX MPH
URBAN	14.5	34.2	30.0
SUBURBAN	16.1	37.8	45.0
INTERSTATE 55 MPH	15.4	36.2	60.0
INTERSTATE 70 MPH	11.7	27.6	75.0

The engines have been sized to provide enough power for good acceleration for a street vehicle. It also provides for ample grade climbing. **Table 2** shows code output of acceleration for the minivan. A four-speed torque convertor is used.

Although fewer speeds would work because of the flat torque curve, a transmission is still needed to get high power, that comes only at the high rpm's, to the ground at low vehicle speed. The instantaneous accelerating force equals the surplus power (KW GOT - KW NEED in the table) divided by the vehicle velocity.

DESCRIPTION OF THE EXPANDER

Many types of expanders were examined for this engine: turbines, rotaries, and several types of reciprocating. In the end the last was chosen using the standard connecting rod/crank configuration and with flathead valves. The others seemed appealing at first. But all had flaws when examined in detail. For example, steam turbines are too small at vehicle power levels. Rotaries, and sliding valves, have difficulty dealing with very high steam pressure.

Figure 8 shows a longitudinal cut through the expander. An inline version is shown for convenience, although V and horizontally-opposed options are certainly possible.

At the bottom is an oil-filled crankcase at near room temperature. The connecting rod and crank bearings are journal, while the main bearings are roller. (Roller bearings are usable here because the oil should remain contamination-free for a very long time. In IC engines, stress risers for combustion particulates can shorten roller bearing life.) Long pushrods are shown. These provide thermal isolation and space for several seals. As shown later, each seal has its own purpose.

The two high-pressure pistons are solid while the other two are conical, a stiff geometry suitable for thin walls. Each cylinder head is independent to avoid stresses due to thermal expansion. The back side of each piston is at its exhaust pressure. Because of the low expansion ratios, this noticeably reduces the load on the pushrods and crank.

Specifications, reformatted from the code output, are given in **Table 3** for the 140 hp Expander. The Expander is discussed in more detail in the following paragraphs.

Table 2. Calculated Minivan Acceleration

MPH	SEC	GEAR	KW NEED	KW GOT
20	1.8	1	2.9	56.3
30	3.2	1	5.3	83.6
40	4.9	2	8.5	79.8
50	7.1	2	12.9	98.8
60	10.3	3	18.5	85.2
70	13.9	3	26.0	98.8
80	19.7	4	34.8	81.3
90	26.7	4	45.3	91.0

QUARTER MILE: 76.7 MPH/17.6 SEC

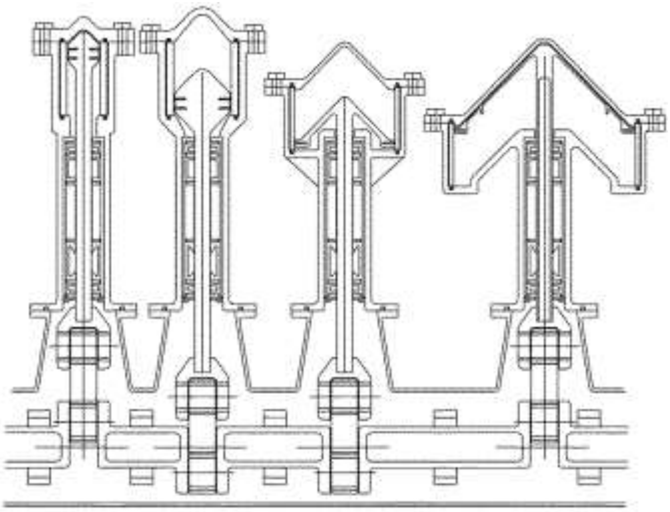

Figure 8. Expander cross section (Valves not shown)

Some comments on **Table 3**:

(1) The peak rpm is chosen so that the full power piston force on the crank is about the same as the rotary force at peak rpm. These forces never act at the same time.
(2) The valve losses of 3-4% are less than the assumed adiabatic losses of 10%, allowing room for other losses.
(3) The total force on each piston is about the same, which should enhance smooth operation.

Table 3
140 hp Engine Expander Specifications

Max rpm	5800
Stroke	49 mm (1.93 in.)
Bores - Stages 1,2,3,4	32, 57, 101, 182 mm (1.26, 2.24, 4, 7.2 in.)
Peak pressures – 1,2,3,4	25.5, 9.2, 3.2, 1.0 MPa (3700, 1334, 464, 145 psia)
Peak piston forces – 1,2,3,4	21, 23, 26, 26 kN
Expansion ratio – 1,2,3,4	2.34, 2.26, 2.26, 2.26
Overall expansion ratio	25.8
Valve Δp/p – 1,2,3,4	4, 4, 3, 3 %
Assumed adiabatic eff	90 %
Max force due to pressure	26 kN (5900 lb)
Max force due to rpm	27 kN (6100 lb)
Bearing journal diameter	37 mm (1.46 in.)
Bearing width	36 mm (1.42 in.)
Piston rod diameter	18 mm (0.71 in.)
Rod buckling safety factor	26
Piston speeds	1200 ft/min at 50 mph 12.3 m/sec at max rpm

Figure 9 shows transverse cuts through each Expander stage with one of the two valves shown. The sliding crossheads, visible near the bottom, eliminate any side thrust on the piston and piston rods.

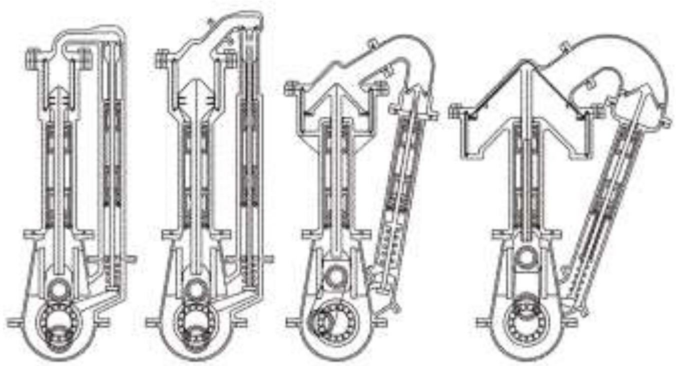

Figure 9. Sections though each Expander stage

The camshaft runs at crank speed. Actually only two cams, intake and exhaust, are needed; and they can be placed on the crank. Pushrod (and pullrod) mechanisms can deliver the motion to the valves. The intake valves open outward, allowing steam pressure to enhance their seal during expansion. However, as is shown below, there is no pressure differential across them when they open.

Shown in **Figure 9** are the flathead valves and high clearance volumes. The latter would be difficult with high expansion ratios; but pose a minor inconvenience with low ratios. **Figure 10** shows, in a P-V diagram, why this is so. Flathead valves eliminate mechanisms and bearings in the hot steam region, which would be difficult to lubricate.

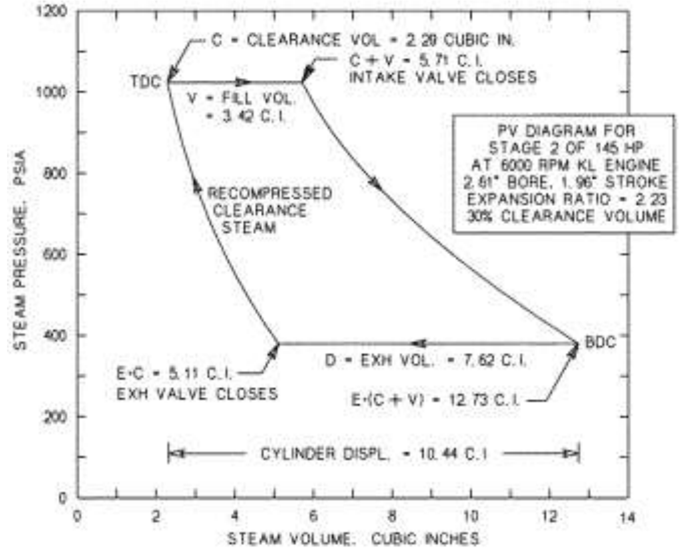

Figure 10. P-V Diagram

Although the engine shown in **Figure 10** is slightly different, the features are the same. Because of the 30% clearance volume, with the expansion ratio of 2.23 the cylinder displacement must be increased from 7.62 cubic inches, as would be needed with zero clearance volume, to 10.44 cubic inches, a 37% increase. This requires an 11% increase in bore and stroke, not a serious issue. With higher expansion ratios, the required increase in displacement would be much greater.

The resulting recompression at the end of the exhaust stroke places the steam at the fill pressure, equalizing it across the intake valve when the valve opens. This eliminates turbulent losses across the valve.

More Expander details are shown in **Figure 11**. Four seals are shown, which are intended to keep steam and oil separated.

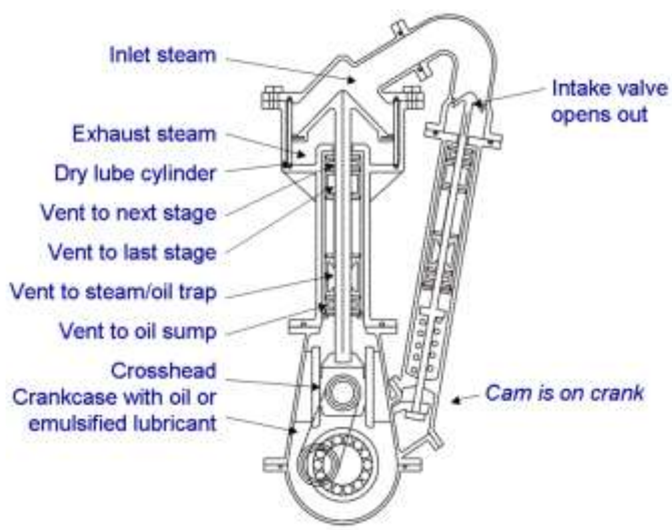

Figure 11. Expander details (Stage 3 shown)

In fact, some mixing is inevitable, and this is directed to a trap that is periodically emptied. This is a close as it comes to changing oil. Very high quality oil can be used because there is no source of contamination other than water, which should be minor because of the seals, and will likely evaporate. Oil change intervals can therefore be very long, if ever.

EXPANDER HOT CYLINDER LUBRICATION

The key uncertainty in the entire engine is the hot cylinder and its lubrication, which will require tests. Even the highest quality oils cannot take the high time-average temperatures. They range from about 1000ºF (540ºC) on the cylinder wall, to 1100ºF (590ºC) in the Expander inlet steam, to up to 1200ºF (650ºC) should the lubricant contact the inside walls of the Steam Generator tubes. Therefore the cylinders are lubricated by a combination of powdered high-temperature lubricant, cylinder sleeves impregnated with these lubricants, and hard surface coatings. **Figure 12** shows the general arrangement.

NASA did pioneering work on high-temperature lubricants at the beginning of the space program (16). Lubricants they identified, such as calcium fluoride, lead oxide, and boric oxide are possible candidates here. There are others, which are discussed below.

The first dry lubricant to be examined will likely be graphite powder and graphitized metals (17). Graphite lubricity requires water or water vapor, which is in abundance here. Fine graphite powder would circulate through the entire steam circuit. Key issues are degradation in the hot steam and clumping in the Feedpump.

Other powdered lubricants also look promising. These include tungsten disulfide (WS_2), selenium disulfide (SeS_2) and hexagonal boron nitride (hBN). These materials have very high temperature capability and low friction (18). They can be used alone or in combination (19,20).

There are a variety of hard surface coatings that can be used in conjunction with dry lubricants. These include nickel-boron (21), special chrome platings (22), and titanium carbide or tungsten carbide (23).

The above are just examples of available dry lubricants and wear-resistant coatings. There are other options and a well-established industry available for assistance. This is a selection task, not a research task.

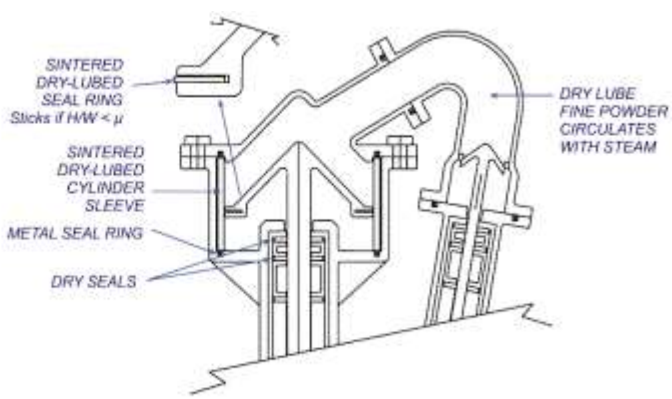

Figure 12. Hot cylinder lubrication

The use of dry rather than liquid lubricants allows one to exploit the fact that static friction is then greater than sliding. **Figure 12** shows how this can be used to reduce seal wear. The primary seal ring, on the periphery of the piston, is deliberately thin (leakage just vents to the next stage). Forces from steam pressure keep it from moving provided its height-to-width ratio is less than the static friction coefficient. Therefore, the seal is stuck in position when there is a differential pressure across the piston, as occurs during the power stroke. The seal then essentially skims along the cylinder wall without wearing. It resets itself only when there is no pressure difference and hence no leakage, that is, during the exhaust stroke. A small spring provides the resetting force.

There are a number of options for the seal itself. One has been patented by the author (24).

EXPANDER PISTON STRUCTURAL ANALYSIS

The ASME Boiler and Pressure Vessel Code Section VIII was used to evaluate stresses at operating temperature for the Expander pistons, cylinders, and cylinder head bolts (25). Bearing sizes were established using applicable vendor load ratings.

An unusual aspect of the Expander is the “loudspeaker” piston shape in the two low-pressure stages, chosen because of their high stiffness despite thin walls. These required finite element analysis to determine stresses, deflection, buckling, and resonant frequency. **Figure 13** shows results for Stage 4. This is the most problematic stage because this piston must be as thin as possible for minimum mass. In fact, to match it for balance, the other stages require ballasting.

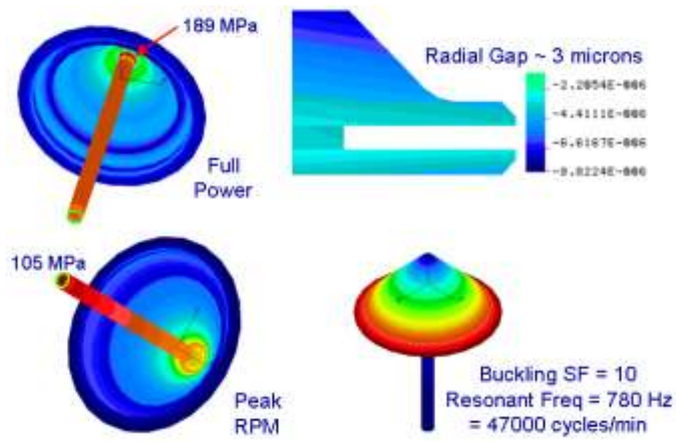

Figure 13. Finite element analysis of Stage 4 piston

The Expander speed is set so that peak acceleration loads roughly match steam pressure loads at full throttle. Fortunately, these loads never occur at the same time and therefore can be evaluated separately.

At the top left of **Figure 13** is stress from the full power load. The maximum is 189 MPa (27 ksi) appearing at the rod/piston junction. This is acceptable even at high temperature because it is very localized bending (it could be designed away). Below that image shows stress from peak acceleration, which occurs at the bottom of the piston rod. The 105 MPa (15 ksi) peak is acceptable, and is in the low temperature region.

Another issue is the possibility of the piston to deflect away from the cylinder wall due to pressure, causing blowby. As seen at top right in a magnified cross section of the seal slot, the gap from inward deflection is only about three microns.

Lastly, problems would occur if the piston resonant frequency were near any operating frequency, or if the cone buckled under full-pressure load. As seen in the bottom right, the resonant frequency is much higher than peak rpm, and there is ample buckling safety factor (SF).

TWO MORE ENGINE AND VEHICLE EXAMPLES

Engines have been sized from 30 hp (22 kW), for a "personal urban car" to 3200 hp (2400 kW) for a locomotive. Two more examples from this database are presented below.

The first, shown in **Figure 14**, is for a semi-truck with 80,000 lb (36,000 kg) gross weight, the maximum allowed on US highways. The engine produces 360 hp (268 kW) at 3600 rpm. The Expander cylinder heads and valves are in light purple (the first stage is under the Burner). As seen, the last Expander stage seems a bit large and can probably be divided in two. The reason for the large size is that the stroke for all stages is the same. Perhaps this feature can be modified in the future for large engines. Note that the condenser readily fits in the existing radiator space.

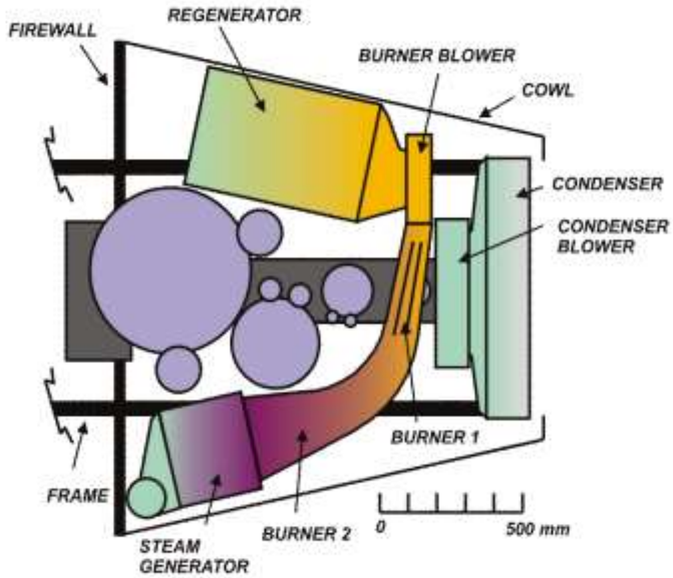

Figure 14. Top view of 360 hp engine for a large truck

The calculated specific fuel consumption and net efficiency are shown in **Figure 15**. There is relatively more room in this truck engine compartment than in smaller vehicles. Therefore one can enlarge the heat exchangers, which reduces pressure drops through them, and use a lower condensing temperature. All this increases efficiency.

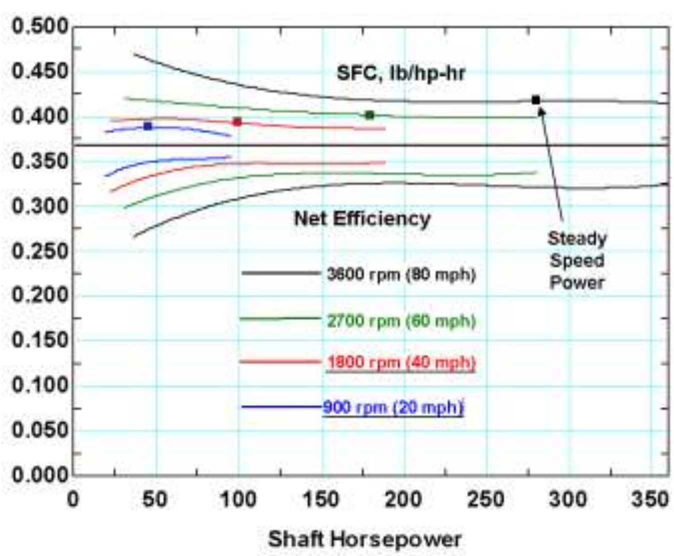

Figure 15. Calculated SFC and efficiency for truck engine

The four points shown in the SFC plots are steady-speed power values needed for this vehicle and indicate SFC values actually accessed during driving. They come fairly close to modern diesel engines. Calculated fuel economy for this 80,000 lb rig with diesel fuel is a nominal 6 mpg (2.6 km/liter) at 60 mph (97 km/hour), about the same as existing rigs.

The second example is an 85 hp (63 kW) engine in a 2180 lb (990 kg including fuel and driver) Saab 96. This popular front-wheel drive car, often used in rallies, was built from 1960 to 1980. Its measured C_D is 0.32, not far above the best sedans today. Despite the light weight, it had excellent crashworthyness and could carry four full-size adults with luggage. A modern version would be an ideal compact car.

The engine installation is shown in **Figure 16**. Here the Expander (in yellow) is horizontally opposed and mounted over the existing fore-and-aft transaxle (purple). The Regenerator (pink) and Steam Generator (red) are mounted side-by-side over the Expander. **Table 4** shows some of the calculated performance.

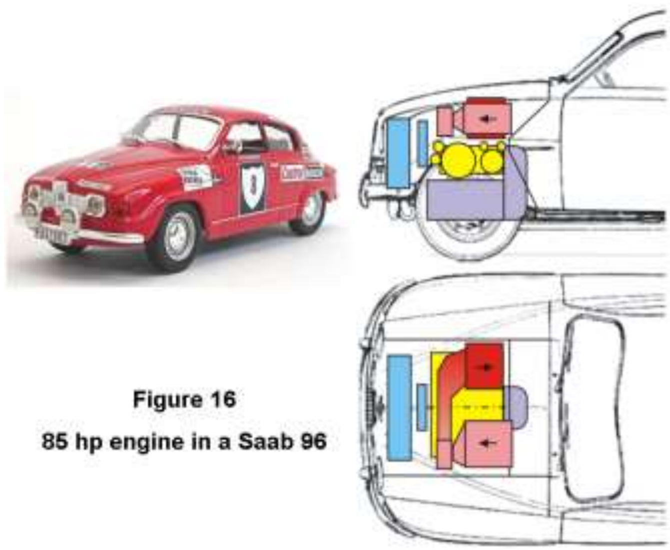

Figure 16
85 hp engine in a Saab 96

Table 4
Calculated Engine Performance in a Saab 96

Engine weight	270 lbs (124 kg)
Zero to 60 mph	9.2 seconds
Fuel economy at steady 40 mph	79.5 mpg
Fuel economy at steady 60 mph	54.2 mpg

DRIVING CYCLE	MPG
Urban	58.3
Suburban	63.0
Interstate 55 mph	60.1
Interstate 70 mph	45.9

This spectacular performance results from the light weight, low drag coefficient and small frontal area of the vehicle, and from the high net efficiency of the engine at normal power settings.

OTHER ENGINE COMPONENTS

The other primary engine components are the Burner, Steam Generator, Regenerator, Condenser, and Feedpump. These are discussed below.

Burner. The two-stage Burner is shown in **Figure 17**. Fuel is injected into Burner 1 with some of the hot air from the Regenerator. There it partially vaporizes and a small amount ignites. This results in a mix of liquid and vaporized fuel, hydrogen and CO, all quite hot. When it enters Burner 2, where it mixes with plenty of also hot air, it ignites quite readily. The air/fuel ratio in Burner 2 is about twice stoichiometric, providing enough air to completely burn the fuel. A combustion temperature sensor keeps the temperature just below that where nitrous oxides form, irrespective of the type of fuel being used. Two-stage burners have already been tested by NASA and DOE and do show very clean burning of the fuel (26).

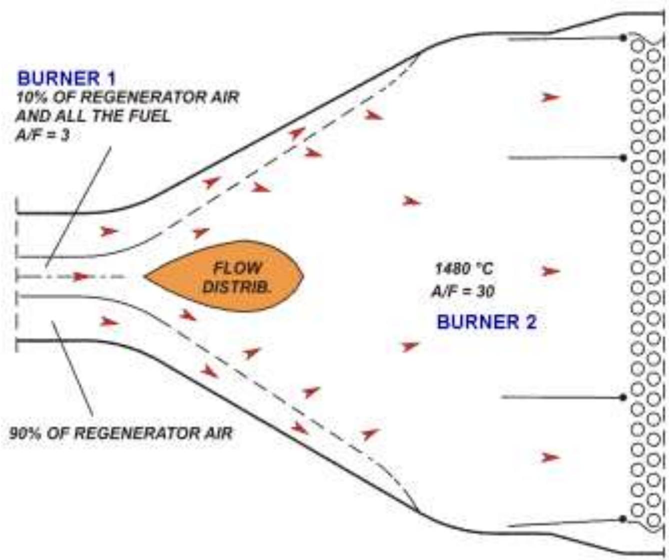

Figure 17. Two-stage burner

Steam Generator. All of the heat exchangers are of conventional design and based on well-established experimental heat transfer and flow friction data (27). These data were imported into the engine software described above.

The Steam Generator is shown in **Figure 18**. It is made up of five sets of tubes, which are discussed below.

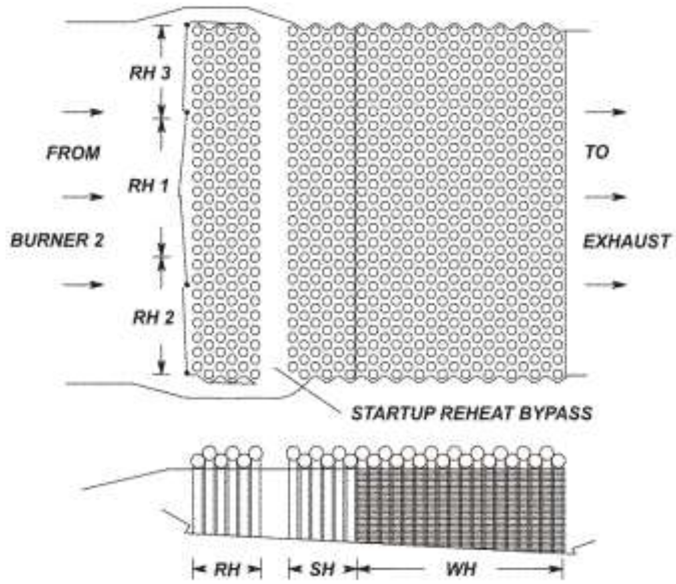

Figure 18. Steam generator

The tubes are connected by manifolds at each end. The manifold arrangement is established by the software to provide enough flow pressure drop though the tubes for reasonably high heat transfer coefficients, but not so much as to adversely affect thermodynamic efficiency.

The Boiler, or Water Heater (WH in the figure) raises the feedwater to saturated vapor if subcritical or about 800ºF if supercritical. It has staggered tubes with sheet fins.

The Superheater (SH) raises the steam to its controlled throttle inlet temperature. It has bare staggered tubes

The three Reheats (RH), placed in parallel, reheat the steam between each Expander stage. They also have bare staggered tubes. The Reheats are bypassed during startup to avoid burnout. Closure plates for this are shown in the figure as well as the bypass channels. The closure plates could also serve as thermal radiation intercepts if needed.

Regenerator. The Regenerator, shown in **Figure 19**, is a plate-fin heat exchanger that is identical on the steam and air sides except that the steam side has four passes while the air side has one. It is a major feature of the thermodynamic cycle and accounts for much of the high net efficiency. In this heat exchanger hot Expander exhaust steam is cooled to the condensing temperature by the incoming burner air.

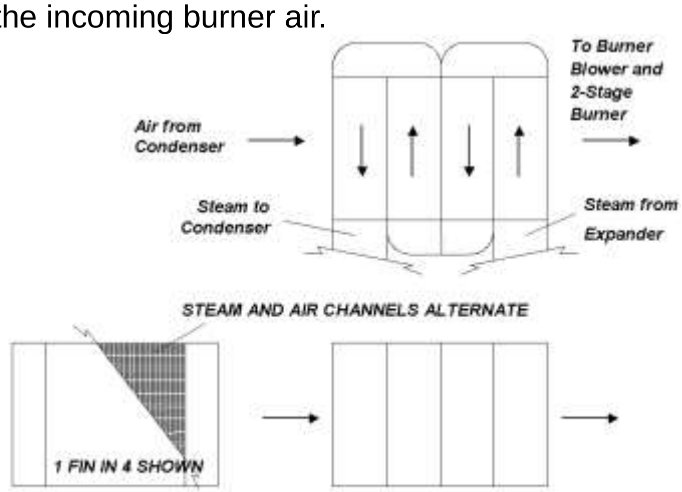

Figure 19. Regenerator

Condenser. The Condenser is shown in **Figure 20**. It looks like a conventional automobile radiator with vertical steam channels and horizontal fins.

It differs in that it has a set of control blinds, or a movable shade, to control air flow through it so that the condensate is

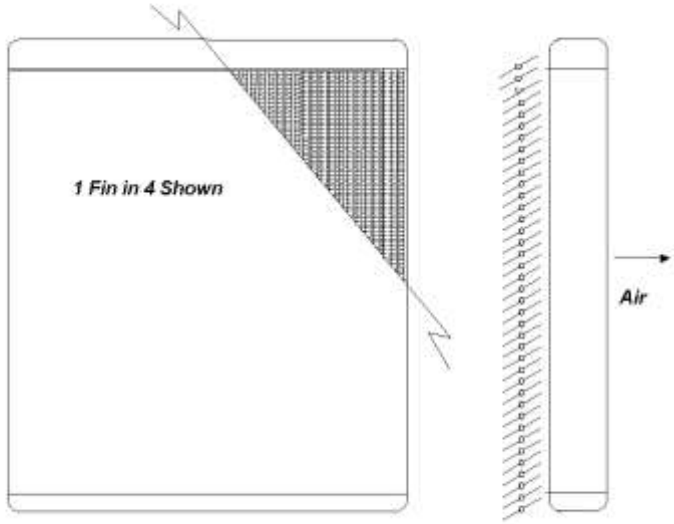

Figure 20. Condenser

subcooled to about 5ºF. Less subcooling and there could be cavitation at the Feedpump inlet. More and efficiency suffers. There may also be a separate condensate channel to separate any air that may have leaked into the system. A small auxiliary vacuum pump would remove the air.

Feedpump and Feed Pressure Control. The Feedpump draws only a few hp but could be a substantial loss at part load. **Figure 21** shows the steps taken to ensure this does not occur.

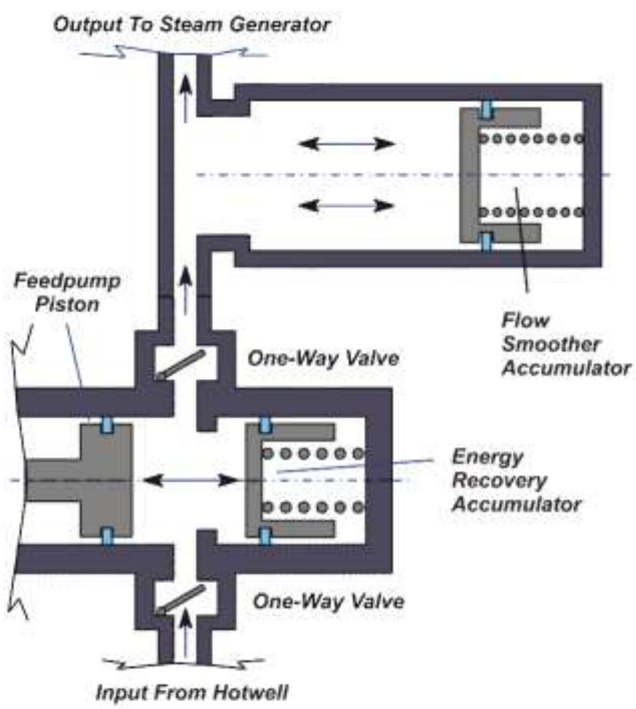

Figure 21. Feedpump and feed pressure control system

As seen, there are two spring-loaded accumulators. These are pistons held in place by long springs nearly fully compressed. Their spring constants are low enough so that the change in force over their piston stroke is small, about 10%.

The Flow Smoother Accumulator serves to smooth out the flow into the

Steam Generator. The Energy Recovery Accumulator comes into play at part load. When the upstream pressure is at or just above the desired value, the force on its piston becomes higher than its spring load and it is pushed back. On the return stroke, some of the stored energy is returned to the Feedpump drive and back to the Expander. Its stroke at least equals to that of the Feedpump piston. Therefore, even if the throttle were fully closed, the stored energy would simply cycle back and forth between the Feedpump drive system and the Energy Recovery Accumulator with little loss.

The Feedpump is also lubricated by the dry lubricant circulating in the system. There will be treated/coated wear surfaces here as well as in the Expander. It is known that tungsten disulfide tends to remain dispersed and does not clump. Tests are needed to determine if other lubricants, for example, graphite, do the same.

MATERIALS

These engines are intended to be mass-produced. Therefore materials used must be readily available, not difficult to fabricate, and reasonable in cost. Yet they must be capable of adequate strength at their operating temperature.

Table 5 shows the primary materials used. The Time-at-Load is used to establish allowable stress using data in the ASME Codes (25).

For engines where fuel consumption issues override initial cost, such as large trucks and locomotives, more expensive materials with higher temperature capability could be used. However, the example truck engine shown above uses the materials in **Table 5**. Therefore, there is room for some efficiency improvement.

Table 5. Materials Used

COMPONENT	MATERIAL	TIME AT LOAD
Reheat Tubes	304 SS	100000 hr
Superheater Tubes	304 SS	100000 hr
Water Heater Tubes	5 Cr - 1 Mo	100000 hr
Hot Expander	304 SS	Short Term
Crankcase	5 Cr - 1 Mo	(Low Temp)
Regenerator	304 SS	Short Term
Condenser	SS Jkt, Cu Fins	(Low Temp)

SAFETY

A steam engine is in fact a collection of pressure vessels. Therefore, as discussed above, the ASME Boiler and Pressure Vessel Codes have been used to maximize safety. In addition, safety is enhanced in this engine by the following:

1. High pressure is confined to small volumes, an advantage of the low expansion ratios per Expander stage.
2. Steam inventory is minimized so that the volume to which all the steam would expand if vented to atmosphere does not exceed the void volume in the engine compartment.
3. Thermal insulation and baffles block lines-of-sight to the pressurized components.

WATER FREEZING

It would be nice to find a working fluid as good as steam but does not freeze. As discussed above, that search has so far been unsuccessful. Therefore, the freezing issue must be addressed. This is done with the following steps:

1. At shutdown, the water is drained into a heavily insulated storage vessel (called a ‘hotwell’). The total liquid inventory is 1-2 liters.
2. To retard freezing, a small amount of battery power is fed to resistance heaters around the storage vessel.
3. Allowance is made for the 10% expansion if it freezes.
4. To restart after a cold shutdown, low temperature combustion gas flows between the thermal insulation and engine components to thaw ice and warm expander oil.

CONTROL SYSTEM

Five parameters are controlled: steam flow rate, feedwater pressure, and three temperatures.

Steam flow rate, which determines power level, is controlled by the Throttle Valve attached to the driver's pedal.

Feedwater pressure is controlled by the Energy Recovery Accumulator in the Feedpump system, discussed above.

Superheat temperature is controlled by the fuel flow.

Combustion temperature is controlled by the Burner air flow using a butterfly valve at the Burner Blower.

Condensate subcool is controlled by the Condenser air flow using the blinds on the Condenser, discussed above.

STARTING TIME

Starting time is an important factor in the engine design. For example, a lower exhaust gas temperature would increase efficiency. But it would also increase Steam Generator mass. This would increase startup time because most of the energy in starting goes into heating the metal, not the steam.

The starting sequence begins with low-temperature combustion gas flowing through the Superheater and Boiler. The Reheats are bypassed with the closure plates shown in **Figure 18**. The Expander and Feedpump begin turning but steam flow bypasses the Expander until vapor starts forming.

The design is adjusted to give calculated startup times of about 30 seconds for private vehicles. Longer times are acceptable for commercial vehicles. These times are for normal low-power operation, such as backing out of a driveway. Full-power capability takes 2-3 minutes, similar to warming up a conventional engine.

SUMMARY

This engine has been the result of several decades of study and analysis. Most of this has been for the development of very sophisticated software, and seeking design solutions that had hindered earlier attempts. No fatal flaws have been found. The engine is an attractive alternative to internal combustion.

However, the opportunity has never arisen to build a prototype. The time now appears right for this challenging task. It is the author’s opinion that no other engine, now in use or under development, can *simultaneously* lay claim to all of the following attributes:

- ✓ High part-load efficiency: high fuel economy, lower carbon dioxide emissions
- ✓ Very wide multifuel capability and simpler fuel processing
- ✓ Negligible NOx, unburned fuel, CO and soot emissions
- ✓ Reasonable weight and compactness
- ✓ Inherently quiet operation
- ✓ Quick starting and rapid response
- ✓ Temperatures low enough so that exotic alloys are not needed

ACKOWLEDGEMENTS

It is a pleasure to acknowledge the following individuals for their technical support, encouragement, and assistance:

Dr. John Clarke, Chief Consulting Engineer (now retired) for the Caterpillar Engine Research Division, whose support coaxed me to revisit this engine after several years' hiatus.

The late Roy Renner (see Ref. 4 below), who encouraged me at the very beginning.

Professor "Lou" London of Stanford University (see Ref. 26) who also provided early advice and encouragement.

Professor James Rizza of California State University at Fullerton, who collaborated with me on a Partially-Condensing Cycle, which preceded this engine.

Dr. Ken Schultz of General Atomics for his ongoing support over all these years.

Mr. Mark Bibeault of Los Alamos National Laboratory for graciously offering to read and critique this manuscript.

NOTES AND REFERENCES

1. For a list of the dozens of steam automobile manufacturers that existed mainly in the early 20th century see: http://www.american-automobiles.com/Steam-Automobile-Manufacturers.html . It also lists most of the known IC engine-driven automobile manufacturers over time.
2. The Steam Automobile Club of America serves as a focus for most current hobbyist activities in steam engines. See: http://www.steamautomobile.com/ .
3. A vintage Popular Mechanics article on Bill Lear's steam car can be found at: http://books.google.com/books?id=KtgDAAAAMBAJ&pg=PA128&lpg=PA128&dq=bill+lear+steam+car&source=bl&ots=SnJXmqlmam&sig=WRvu88Ej1lk4hTRFkNF5g_5V1es&hl=en&ei=G0McS621C9XcnAf1i8zmAw&sa=X&oi=book_result&ct=result&resnum=8&ved=0CB4Q6AEwBw#v=onepage&q=bill%20lear%20steam%20car&f=false .
4. R.W. Hauck, M. Wenstrom, and R. Renner, *The California Clean Car Project*, Final Report, November 15, 1974.
5. The highest estimate for remaining oil reserves is 1342 billion barrels. See the tabulation from the US DOE Energy Information Administration at: http://www.eia.doe.gov/emeu/international/reserves.html .
6. World oil consumption as of July 2009 was 85.47 million barrels per day. See the DOE/EIA tabulation at: http://tonto.eia.doe.gov/energyexplained/index.cfm?page=oil_home#tab2 . This translates to 5500 cubic feet per second!
7. Dividing Ref. 5 by Ref. 6 gives 15701 days = 43 years.
8. http://www.epa.gov/climatechange/emissions/downloads09/GHG2007-ES-

508.pdf .
9. Based on: Gasoline/diesel average lower heating value of 43.2 MJ/kg, average coal heating value of 25.5 MJ/kg, average coal plant heat rate of 10.6 MJ/kWh, electricity transmission and vehicle motor efficiencies of 95%, and vehicle battery charge/discharge efficiencies of 90%. This is compared to an IC engine-powered vehicle with cruise specific fuel consumption of 450 gm/kWh. The end result is that the plug-in vehicle generates about 50% more carbon dioxide if its charging electricity comes from coal plants.
10. DuPont developed many different working fluids for steam power, mainly with the intention of being able to use turbines of reasonable size (steam turbines are too small for automobiles). An interesting article on their efforts is found at: http://books.google.com/books?id=mzWswaAaeEAC&pg=PA56&lpg=PA56&dq=doerner+dupont&source=bl&ots=xOy_Wu6EhP&sig=N6uJ4upjag8OffRkB2ljOB5oGlw&hl=en&ei=ikQcS5SeOM7ZnAf9xaTRAw&sa=X&oi=book_result&ct=result&resnum=2&ved=0CA4Q6AEwAQ#v=onepage&q=&f=false .
11. R.F. Bourque, *External Combustion Power Cycle and Engine with Combustion Air Preheating*, US Patent No. 4,090,362, issued May 23, 1978.
12. R.C. Juvinall, **Fundamentals of Machine Component Design,** John Wiley and Sons, 1983, (Figure 1.10).
13. Composite specific fuel consumption data was derived from http://autospeed.com/cms/A_110216/article.html and www.heat2power.net .
14. G. Genta and L. Morello, **The Automotive Chassis v.2 System Design,** Ch 22 p. 172, Springer Verlag 2009.
15. Taken from consumer reviews reported at: http://www.edmunds.com/pontiac/transsport/1998/consumerreview.html .
16. E.E. Bisson and W.J. Anderson, *Advanced Bearing Technology*, NASA SP-38, 1965.
17. Graphitized metals are used in high-temperature food processing equipment to avoid contamination. One source is seen at http://www.graphalloy.com/ . Peak operating temperatures are claimed to reach 1500 ºF (800 ºC)
18. M.K. Impex in Canada specializes in dry lubricants and could be helpful in the evaluation program. See: http://www.lowerfriction.com/product-page.php?categoryID=1&gclid=CNL0kdbIx54CFQ4NDQodMBRPrA .
19. H.E. Sliney, *Carbide/Fluoride/Silver Self-Lubricating Composite*, US Patent 4,728,448 issued March 1, 1988 and assigned to NASA.
20. C. DellaCorte and J.C. Wood, *High Temperature Solid Lubricant Materials for Heavy Duty and Advanced Heat Engines*, DOE/NASA/50306-5, NASA TM-106570, 1994 Fall Technical Conference, ASME Internal Combustion Engine Division.
21. J.G. Smith, *NiBron Nickel Boron Wear Resistant Coatings*, Amer. Soc. Metals, Metals/Materials Technology Series, Paper 8305-041, October 1983.
22. Armoloy has developed a chrome plating specific for wear resistance at high temperature. See: http://www.armoloycompany.com/wear-resist.htm .
23. P.N. Dyer et al, *Cvd Tungsten Carbide and Titanium Carbide Coatings for Aerospace Components*, SAE Paper 890933, April 1989. Abstract visible at: http://www.sae.org/technical/papers/890933 .
24. R.F. Bourque, *Shaft seal having capacity for substantial wear*, US Patent 5,306,022, issued April 26, 1994 and assigned to General Atomics.
25. American Society of Mechanical Engineers Boiler and Pressure Vessel Code Section VIII Div. 1 and 2, Section II Part D, and Section I.
26. H.G. Lew et al, *Low NOx Heavy Fuel Concept Program, Phase 1: Combustion Technology Generation*, Final Report DOE/NASA/0146-1, NASA CR-165482,

October 1981.
27. W.M. Kays and A.L. London, **Compact Heat Exchangers**, McGraw-Hill 1964.