PROPOSAL

LIGHT STEAM POWER SYSTEM

PREPARED FOR PRESENTATION TO JANICKI INDUSTRIES FOR CONSIDERATION IN FULFILLMENT OF GATES FOUNDATION GRANT

May 16, 2012
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INTRODUCTION

Steam enthusiasts tend to have pet projects, cherished ideas and strong opinions, often not compromising or working well with others and striving to prove ideas long after prudence dictates change (see the entire career of Abner Doble). This enthusiasm has created a longstanding cycle of overly optimistic claims followed by inevitable disappointment; perhaps grounded in the belief that a successful demonstration justifies all. "Once you get something running, anyone can take over and get it into production." Experience indicates the opposite; sufficient complexity can achieve results, after a fashion, refining those results into a simple, elegant, reliable, economical and buildable form is often the greater challenge. Fixing the ideas of simplicity and practicality firmly in mind at the project outset improves the likelihood of eventual success.

Universal simple solutions or spectacular results are suspect; often best practice is tailored to the application and ideal theoretical answers may be less practical. I cheerfully admit having pet projects and ideas but refrain from presenting them, as much as I would like to. Advocating untried solutions, when I feel the allotted time is too short for more than direct and abbreviated development, would be unprofessional.

Light steam engines have a long history but, I submit, are not a mature technology because development slowed to a crawl around a century ago with the advent of gasoline and diesel engines; little from that era classifies as "modern". I understand a system must be presented in about 2 ½ years, a very short period for even automotive manufacturers to develop an engine from "the ground up". Engine manufacturers possesses dedicated equipment, trained specialists, enjoy existing vendor support and institutional experience; lacking this full range of resources, a different approach to tackling the problem is necessary. Fortunately, products similar to our needs are readily available for modification; it may be possible to collapse the work load to a manageable level by "borrowing" off-the-shelf hardware **and limiting development to only the most critical 'steam-specific items'**. Even borrowing liberally, a successful outcome depends on disciplined project planning, management and scheduling. The dearth of successful light steam product launches suggests a historical lack of strong planning, discipline and focus; by vigorously adopting project management tools (such as PERT charts) the needed discipline can be imposed. "Modern steam" programs often seem dedicated to proving personal theories rather than producing a marketable product; forging ahead on a fixed course regardless of time or cost until running out of one or both. The successful program will exercise strong discipline and adherence to the critical development path but will also allow a degree of flexibility and encompass "Plan B's" to ensure ongoing progress rather than stagnation when advancement stalls.

More than efficient manufacture is needed; at some point the customer takes possession and this transfer must be planned to promote a happy outcome. To this end extensive testing is necessary to:

- Verify performance or provide the necessary data to correct deficiencies
- Verify reliability or provide necessary data to make corrections
- Aid in establishing warranty policies
- Aid in determining what portion of revenues must be set aside to fund warranty repairs
- Establish correct operating procedures
- Establish effective and simple maintenance routines

Ease of operation and maintenance after the sale are critical to ensuring customer satisfaction; representative test subjects working with the hardware during the test phases may be needed to develop tools and materials suitable to the target customer.

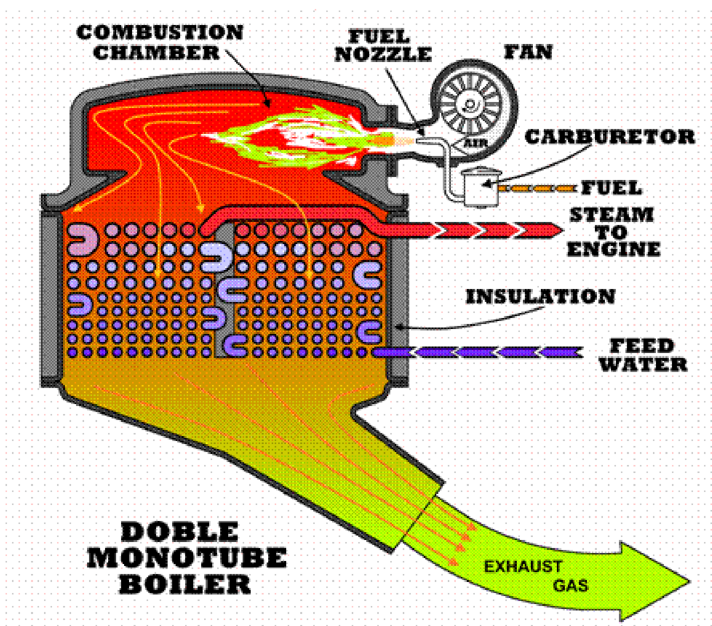
SYSTEM

My initial understanding was that program contemplated converting a diesel engine into a compound steam engine of about 75 horsepower which would turn an electrical generator. The furnace was to be fired by solid fuel, combusted in a fluidized bed. Process waste steam derived from the engine was to provide heat for some unspecified drying or distillation apparatus and the entire system was to be fitted on a trailer. Since that time the information has changed slightly, the output has increased to 100 horsepower and a triple expansion compound with inter-stage reheat is projected to be built up from two joined diesel engines. The following proposal reflects the original goal of 75 horsepower as the changes needed for greater output are immaterial to the discussion.

Steam Generator

The steam generator will be one of the two major technical challenges in the system, the other being the steam expander (engine). The fire tube boiler is inadvisable for any number of reasons such as slow startup, the need for a heavy pressure vessel and incompatibility with the proposed fluidized bed combustion system. It is mentioned merely to acknowledge that the alternative exists and to rapidly dismiss it.

Water tube boilers can be divided into a number of categories; once-through versus recirculation being a good starting point. Once-through boilers admit water in one end and emit steam at the other, the flow being a straight line. The monotube is a once-through boiler with only a single flow path; other once-through boilers may have multiple parallel paths. Recirculation boilers contain a larger weight of steam and water; the fluid travels a loop with water being added, then heat is applied and some steam produced, the steam is then extracted and the remaining water shuttles back to the beginning of the loop for another pass.

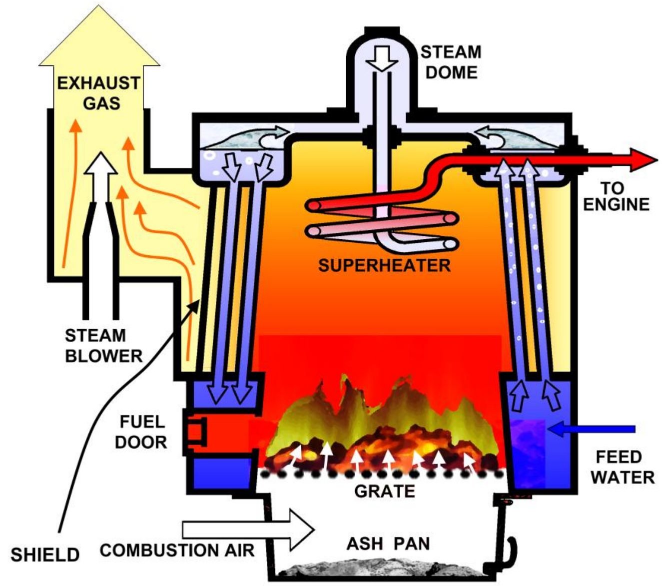


Recirculating boilers have some advantages over once-through designs such as monotubes.

- Greater water weight and flow velocity permit higher safe firing rates.
- Submerged steam generator surfaces make control simple; steam pressure regulates burner firing rates while the water level regulates the water feed rate.
- Saturated circulating water readily turns to steam when pressure falls, dampening pressure fluctuations.
- Non-volatile contaminants remain in solution throughout the circulating loop rather than depositing on the superheater; the risk of tube burnout, corrosion and scale damage to the engine are lessened. Top and bottom blow-downs along with occasional chemical treatments remove contaminants.

Recirculation comes in two flavors, natural and forced. Natural circulation occurs because a volume of cold water is heavier than a similar volume of mixed hot water and steam; the cold water will sink, displacing the hot water and steam upwards. Forced circulation boilers use a pump to move the working fluid.

Natural recirculation becomes more rapid as the boiler becomes taller, the weight difference between hot and cold columns increases with height, accelerating circulation. Natural circulation boilers must orient the flow paths generally upwards and, due to friction losses, work better when bends and turns are kept to a minimum.



NATURAL CIRCULATION BOILER (THORNYCROFT)

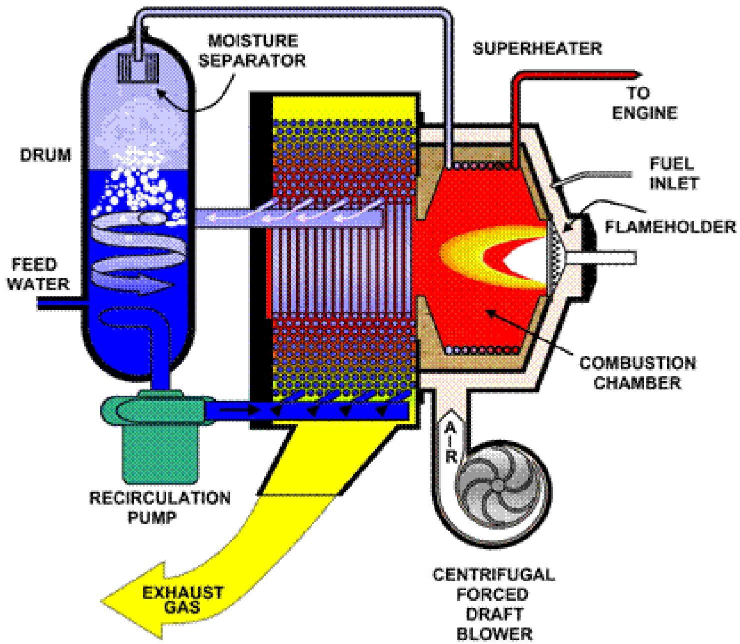
Forced flow boilers, such as once-through or forced recirculation, are usually wound into coils but readily utilize straight lengths and hairpin bends. Orientation is immaterial and need only conform to the boiler shell.

I recommend against once-through boilers, such as monotubes because, regulation of such boilers is difficult and depends on well calibrated controls, making them less suitable for use in the field. Natural circulation may prove less than ideal because boilers supplying a 75 or 100 horsepower engine are small and less likely to circulate strongly.

Forced recirculation is recommended. Positive circulation permits greater steam output and added burnout protection while the development experience is transferrable to future higher performance steam systems.

At some point, steam and water must be separated, steam being directed to the engine and water returning to the generator tubes. This separation occurs in a drum, or in drumless boilers, through the agency of specialized separators. Drumless boilers are smaller, lighter and cheaper but more difficult to control properly and lacking in reserve. The drum presents the more straight-forward option and is recommended for the following reasons:

- The pipes to and from the steam generator tubes and recirculation pump should be tangential to the drum walls so as to produce a spinning action in the steam/water mix. Centrifugal force produced by spinning improves steam and water separation; and reduces water carryover into the superheater, minimizing thermal shock and deposition of harmful deposits on the superheater wall.
- The recirculation pump suction pipe can be positioned in the drum to act as a ram intake, reducing the energy spent operating the pump.
- Hot, pressurized water in the drum readily vaporizes upon sudden pressure drops, serving as a buffer against fluctuations and assuring more stable operation.



FORCED RECIRCULATION BOILER (VAPOR CORP)

Superheaters

Elements heated by convection become hotter with increases in firing rate while those heated by radiation become cooler, by combining proper proportions of radiation and convection heated elements we can produce a superheater with better inherent temperature stability. Such units are called 'uncontrolled superheaters' to distinguish them from those with thermostatic regulation. Presumably, fluidized beds provide primarily convective superheat, causing temperatures to rise with firing rate increases.

Increased superheat temperature is desirable to promote higher efficiency but steam temperatures may potentially rise to damaging levels. Active superheat control of superheat is desirable because it permits use of higher and more efficient temperatures without swings that exceed safe limits. Three methods of control are:

- **Controlled superheat furnaces** are primarily dedicated to superheating steam and the firing rate is thermostatically controlled by the outlet temperature.
- '**Normalizers' or 'attemperators'** inject cooling water midway into the superheater path under control of a thermostat positioned at the superheater outlet. Like the de-superheater that follows, the normalizer requires a superheater that is a bit over-sized and prone to overheating, permitting temperature control by regulating the amount of cooling supplied to a point in the middle of the superheater..
- **Surface contact de-superheaters** control temperature like a car thermostat. A sensor at the superheater outlet directs a valve to split the fluid flow into two paths, directing one path to a cooling heat exchanger in the drum. Recombining the paths produces a cooler mix which passing through the remaining superheater emerges at the correct temperature. Positive feedback occurs because transferring heat to the drum vaporizes a portion of the saturated water therein, raising the pressure and causing the controls to reduce the firing rate which in turn tends to reduce the superheat.

Choosing between these options, one must note that controlling superheat temperature by throttling a furnace has proven difficult, superheated steam has no moisture content and lacks the moderating effect of latent heat, making rapid temperature swings more common. Adding water into the steam is a very effective cooling method, undoubtedly why Doble adopted the normalizer. Being so effective, the flow of cooling water is necessarily small and must be carefully regulated through a small tube or orifice. Normalizers are light and compact, but precise operation is important and clogging of the small orifice will disrupt correct functioning.

Other potential problems include risk of the coolant creating thermal shock in hot piping and the sudden cooling causing contaminants to suddenly drop out of suspension and form harmful deposits on the superheater wall.

The surface de-superheater cooling process is less effective than the normalizer because steam is a poor thermal conductor and a relatively large flow through the desuperheater is needed to produce the same temperature change. This isn't necessarily bad; the equipment can be less precise and delicate while still providing close temperature regulation. While larger and heavier than the normalizer, it is also robust and reliable. Given the service this system is meant to see, the surface de-superheater is probably a better choice.

Currently Projected Expander

As I understand it, the current contemplated design comprises a three stage compound engine composed of two 4 cylinder diesel engines connected to run in series operation with one piston in the first engine being the High Pressure stage, the following three pistons being the Medium Pressure and the second engine in its entirety comprising the Low Pressure. Admission steam pressures of 2000 psi and very low RPM operation on the order of 500 rpm is required to achieve efficient expansion while limiting output to about 100 HP.

This configuration contemplates an HP stage utilizing admission steam pressures on the order of 2,000 PSIA to achieve high efficiencies.

I have a number of reservations about achieving the level of efficiency expected:

- Steam "blow-by" past piston rings is a function of Mean Effective Pressure (MEP) and duration of steam playing against the rings (residence). Dutcher Industries compound engine used similar pressure, blow-by on the order of ten percent in the HP stage were consistently found despite attempts to control the loss; this limits compound efficiency and poses significant lubrication challenges.
- Using two diesel engines to achieve three expansion stages involves more mechanism than is traditional; the result should be a minimum of twice the typical friction losses.
- The large number of cylinders creates unfavorable surface area-to-volume ratios; heat loss through the cylinder walls appears to be on the order of 2.5 times greater than purpose built engines.
- Both the engine block and cylinder head may require cooling to control expansion and protect lubricants, the large surface areas present greater potential for cooling losses.
- Transferring steam between stages takes energy. Assuming 20 milliseconds to fill the cylinder at 500 rpm, steam velocities from 50-100 feet per second are probably reasonable, attaining this velocity requires diverting steam energy from other purposes. Single acting engines only suffer this loss once.
- Given that the boiler is less than 100% efficient, inter-stage reheating is also less than 100% efficient. We can expect losses to compound, offsetting some of the expected benefits.
- Piping between the expander and reheater increase the potential for radiative and convective losses.

Admittedly, not all of these losses are large, but the effects are cumulative. The proposed compound design leads to some other concerns:

- A four cylinder, four stroke engine divided into one HP and three MP cylinders will produce torque very unevenly; leading to greater cyclical fatigue than the crankshaft manufacturer contemplated when setting the maximum torque specifications and possibly leading to premature failure.
- Diesel engines use steel shell bearings coated with anti-friction metals and forced oil lubrication. Crankshaft rotation forms the oil into a supporting wedge upon which the shaft floats with the strength of the wedge depending upon the shaft speed. Operating at the envisioned 500 rpm places the engine at risk of "lugging", which collapses the oil wedge and promotes premature bearing failure.

- Commercial electric generators of the projected output typically operate at higher speeds than 500 rpm, mandating the use of a speed increaser which adds size, weight and cost to the unit while at the same time increasing friction and lowering overall efficiency.
- Mating two expanders directly is difficult, tight tolerances are needed to achieve correct alignment which may not have been a priority of the builder. Flexible couplings are simpler but increase the overall size as the mated units must be spaced out; couplings requires periodic inspection and maintenance, presents an increased likelihood of failure and add extra friction to the system.
- The LP stage requires a second donor engine of larger displacement, greatly raising overall size, weight, cost and complexity.
- Compound design really demands adjustable valve cutoff on each stage. The larger LP engine will require an entirely different cylinder head and valve assembly, while the variable cutoff adds components. The added hardware raises cost, complexity, increasing likelihood of component failure.
- Lubricating oil carried between stages may decompose in the resuperheaters, forming insulating carbon deposits that can cause superheater failure or break loose and damage the following valves or cylinders.

Proposed Expander

My initial conversation led me to believe the project is facing a fairly tight time-line of about 2 ½ years, which prompted the notion of converting existing diesel engines in the interest of expediency. I concur with this assessment; development ”from the ground up” of a fully tested, industrial grade system suitable for mass production is not likely in this amount of time with the resources at hand.

I firmly believe a successful power plant must be very simple and durable, to this end two approaches stand out:

1. Convert Detroit Diesel two stroke engines to uniflow operation. These were produced for a number of years and in four different numbered series: 53, 71, 92 and 149. They can be obtained as remanufactured Detroit Diesel models while some variants are still available as newly built units for military applications by MTU Detroit Diesel, Inc.
2. Purchase complete diesel gen sets, possibly marine units, and remanufacture critical components to convert them into steam power plants.

The Detroit Diesel option promises greater theoretical efficiency because the uniflow expander obviates passing cool exhaust through the cylinder head, which in turn cools incoming steam. The two stroke crankshaft layout produces very smooth torque when operated under steam. To the best of my knowledge, these engines employ pushrod operated valves actuated by a single camshaft and, in some models, contain a balance shaft; with some effort this balance shaft provides the opportunity to serve as a second cam, permitting separate and controllable admission and exhaust events.

The marine gen set option has the advantage of purchasing an entire power plant with the engine and electrical elements being fully integrated and proven; minimizing risk and effort.

Both approaches have much to commend them and the choice might come down to factors other than pure suitability as an engine. I settled on the marine gen set option because the engine, generator, frame and supporting hardware come as a tested and packaged unit; simplifying overall development. The uniflow engine itself may be a bit easier to convert to steam, so the choice really comes down to which option promises the more rapid, certain and economical outcome. While uniflow engines are typically more efficient, counterflow engines using separate exhaust and admission valves narrow the gap greatly by not pre-cooling the steam admission passage. Such counterflow engines can significantly delay the onset of compression while still achieving full recompression, and do so with less clearance volume than many uniflow designs, which increases the power output per cylinder and further negates some of the uniflow efficiency advantage.

Let me reiterate that although I chose the counterflow design based on the availability of existing donor gen sets, I am not critical of converting the "Jimmy" Detroit Diesel into a uniflow steam engine. In any case, the engine parameters for either uniflow or counterflow diesel to steam conversions are relatively similar and the following is generally applicable for either decision.

My initial conversation with you in March revealed a need for output of about 75 horsepower, a short internet search identified a packaged diesel gen set meeting this basic parameter; the Kohler Power Systems 55EFOZ (50 Hz) Marine Generator Set which is powered by the John Deere 4045TFM Marine Engine. (Note: A similar V-8 powered gen set built by the same manufacturer is available for inspection at Tom Kimmel's nursery.) I did not intend this as a specific product recommendation, other products might be considered more suitable for technical or economic reasons which we have not had time to examine; it does seem to serve as a representative sample of commercial offerings, however. A more recent conversation suggests a desired engine output of perhaps 100 horsepower; this can be accommodated by similar 6 cylinder models of the same engine family and points out that the basic features of such steam systems should be readily scaled within certain power ranges.

Model: 65EOZ (60 Hz)
55EFOZ (50 Hz)

KOHLER POWER SYSTEMS

3-Phase Diesel

ISO 9001 KOHLER POWER SYSTEMS NATIONALLY REGISTERED

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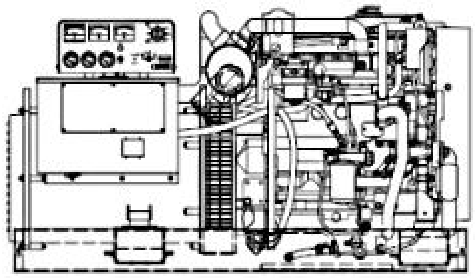


Marine Generator Set

Generator Features

- Remote control connector
- Class H insulation
- Reconnectable voltage
- 60/50 Hz field adjustable generator set
- One- or three-phase reconnectable alternator
- Voltage regulation of ±2%
- Frequency regulation of ±5%

Optional Accessories

- Sound shield
- Power takeoff (PTO)
- Electronic governor
- Paralleling governor
- Circuit breakers

Application Data

Engine

Engine Specifications	60 Hz	50 Hz
Manufacturer	John Deere	
Model	4045TFM	
Type	Inline, 4-cycle	
Number of cylinders	4	
Firing order	1-3-4-2	
Aspiration	Turbocharged	
Displacement, L (cu. in.)	4.5 (276)	
Bore and stroke, mm (in.)	106 x 127 (4.19 x 5.00)	
Compression ratio	17.0:1	
Combustion system	Direct injection	
Rated rpm	1800	1500
Maximum power at rated rpm, HP	95	76
Cylinder block material	Cast iron	
Cylinder head material	Cast iron	
Piston rings	2 compression/1 oil	
Crankshaft material	Forged steel	
Connecting rod material	Forged steel	

Lubrication

Lubricating System	60 Hz	50 Hz
Oil pan capacity with filter, L (U.S. qt.)	14.0 (14.8)	
Type	Pressure	

Operation Requirements

Air Requirements	60 Hz	50 Hz
Engine combustion air requirements, L/min. (cfm)	4502 (159)	3738 (132)
Cooling air required for generator set at 14°C (25°F) rise, m^3/min. (cfm)	N/A	N/A
Exhaust gas flow, m^3/min. (cfm)	14.2 (502)	11.9 (419)

Fuel Consumption	60 Hz	50 Hz
Diesel, Lph (gph) at % load		
100%	17.4 (4.6)	14.4 (3.8)
75%	13.2 (3.5)	10.6 (2.8)
50%	9.8 (2.6)	7.6 (2.0)
25%	5.7 (1.5)	4.2 (1.1)

Light steam power plant temperatures of 1000 to 1200 degrees F have been long proposed as a way to attain thermal efficiencies competitive with internal combustion. Unfortunately, use of such temperatures has led to engine failures from lubricant breakdown and metallurgical challenges in the boiler. A thorough program of performance and reliability testing will readily consume a minimum of six months, and possibly over a year, leaving 1 to 1 1/2 years actual development time. This constraint, and the requirement to operate in remote areas with minimal technical support, suggests initial steam temperatures of no more than 850 F with changes being based on test results. Previous steam systems have proven reliable at this temperature and efficiency should be "good enough" for our purposes. Development of "new and improved" models operating at higher conditions could be examined when the initial program has been completed.

Some years back, I wrote an Excel spreadsheet using IAPWS-97 add-ins to estimate expander behavior assuming isentropic steam properties. Using the John Deere bore, stroke and rpm I juggled variables to calculate an output of about 75 horsepower while assuming losses of about 70%. Starting with 1000 psi as a working pressure the resulting calculations revealed a solution having a clearance volume of about 2% and cutoff of about 3.6 % with recompression to near the admission pressure in order to improve efficiency. Yielding the desired power, a nice calculated water rate around 10 lbs. per horsepower-hour was had. However, the T-V diagram indicates the expansion curve intercepts the saturation line around 40% of full stroke, leading me to believe it is probably unrealistic to expect this level of performance and efficiency.

STEAM ENGINE PERFORMANCE CALCULATOR

ASSUMES OVERCOMPRESSION RELIEVES TO STEAM CHEST

DATA ENTRY BLOCK

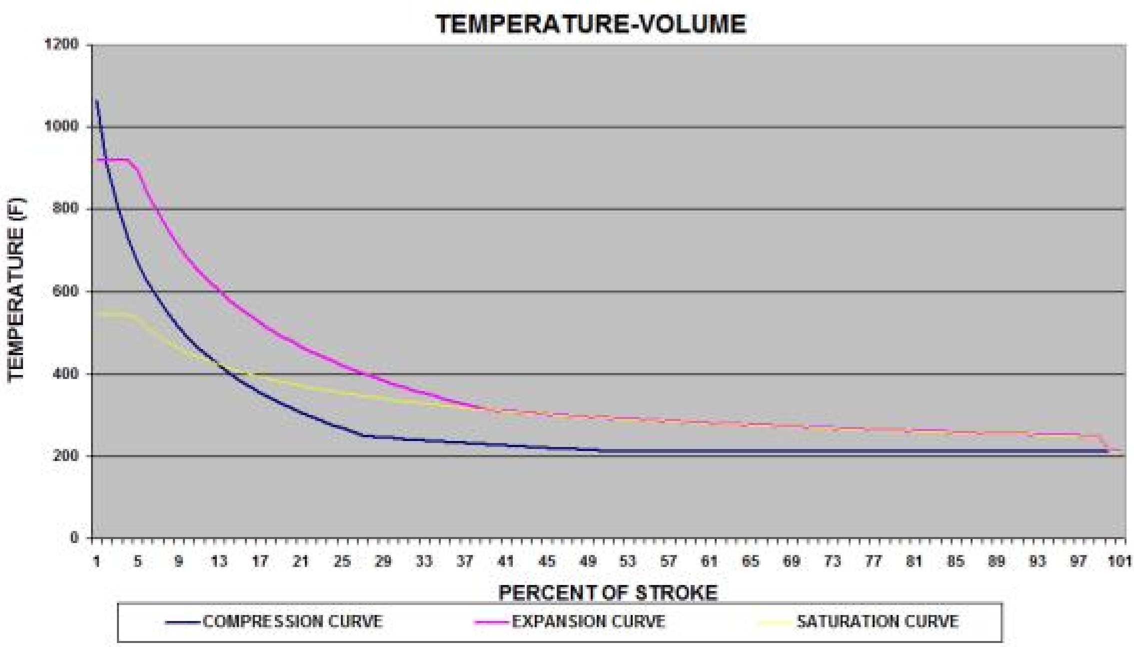
Steam Pressure (psia)	Steam Temp (F)	Exhaust	Jimmy factor
1,000	850.00	15.000	70.000
305.35 F superheat		30.54" Hg	
Bore (ins)	Stroke (ins)	Cylinders (#)	RPM
4.190	5.000	4	1500
Clearance (% of stroke)	Cutoff (% of stroke)	Release (% of stroke)	Cmprssn start (% of stroke)
2.175	3.600	98.000	50.000

Note: Check TV graph, expansion curve in saturation region.

MECHANICAL DATA			
Horsepower	76.55	Total MEP (PSI)	104.69
Expansion MEP:	151.06	Compression MEP:	46.37
Water Rate (lbs/hp-hr)	10.06	Efficiency (%)	17.82
Max piston thrust (lbs)	13,788.53	Average Torque (ft-lbs)	268.03
Water used (lbs/hr)	770.21		

EXPANSION DATA			
Expansion ratio	17.35 : 1	Release PSI	28.71
Release Temp (F)	247.79	Exhaust Temp (F)	212.99
Nominal Cutoff (%)	3.67	Stm Used (lbs/hr)	770.21

COMPRESSION DATA	
Compression ratio: 23.98	Nominal clearance (%) 2.22
Mixing temperature: 919.7	Yields superheat of: 375
Compression temp: 1063.7	Yields superheat of: 540.6
No overcompression	
Inches Admiss press reached: 0.0145	after TDC
Steam wt recycled (%): 38.47	
Peak pressure (psia): 833.849181	



Reselecting for 400 psi and a doing a bit of juggling, the computer produced an engine that delivers about the same power with 2.0% clearance and about 11.5% cutoff. The water rate is a “good enough” 11.65 pounds per horsepower-hour, similar to Doble Series E compound engines. The steam temperature at release is just approaching the saturation line, so over-expansion does not seem to be a problem. The cutoff is still short enough to be challenging, but not to the level of the 1000 psi engine. These parameters seem to make a decent starting point from which to develop an engine, an ongoing test program would include fine tuning the details.

STEAM ENGINE PERFORMANCE CALCULATOR

ASSUMES OVERCOMPRESSION RELIEVES TO STEAM CHEST

DATA ENTRY BLOCK

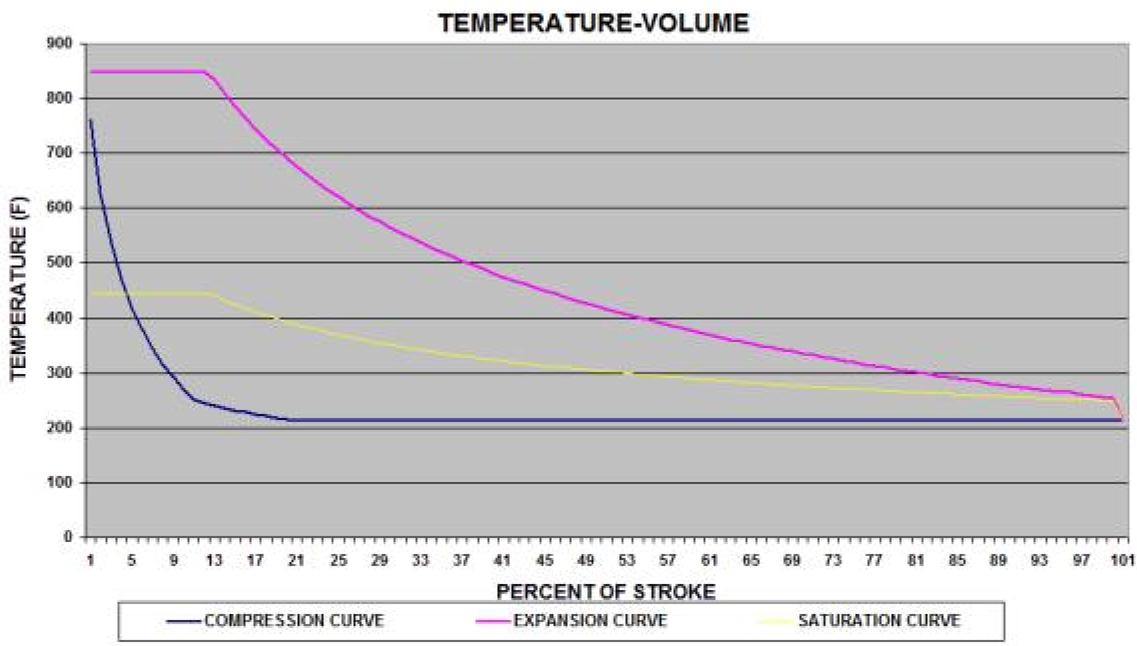
Steam Pressure (psia)	Steam Temp (F)	Exhaust	Jimmy factor
400	850.00	15.000	70.000
405.37 F superheat		30.54" Hg	
Bore (ins)	Stroke (ins)	Cylinders (#)	RPM
4.190	5.000	4	1500
Clearance (% of stroke)	Cutoff (% of stroke)	Release (% of stroke)	Cmprssn start (% of stroke)
2.000	11.500	99.000	20.000

MECHANICAL DATA			
Horsepower	77.11	Total MEP (PSI)	105.45
Expansion MEP:	128.45	Compression MEP:	23.00
Water Rate (lbs/hp-hr)	11.65	Efficiency (%)	15.13
Max piston thrust (lbs)	5,515.41	Average Torque (ft-lbs)	269.97
Water used (lbs/hr)	898.37		

EXPANSION DATA			
Expansion ratio	7.48 : 1	Release PSI	29.37
Release Temp (F)	254.24	Exhaust Temp (F)	212.99
Nominal Cutoff (%)	11.62	Stm Used (lbs/hr)	898.37

COMPRESSION DATA	
Compression ratio: 11	Nominal clearance (%) 2.02
Mixing temperature: 848.3	Yields superheat of: 403.7
Compression temp: 762.1	Yields superheat of: 342.3
No overcompression Inches Admiss press reached: 0.0184 after TDC	
Steam wt recycled (%): 13.91	
Peak pressure (psia): 307.867657	

Note: TV curve generally superheated.



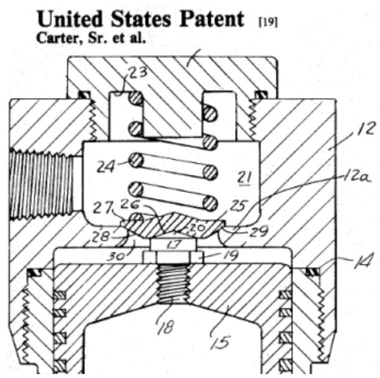
With a water rate of about 12 pounds/hp-hr., at 100 horsepower the water consumption would be 1200 pounds per hour; making allowances for performance degradation over time and auxiliary uses, a minimum steam generator capacity of 1500, and preferably 1800, pounds per hour would be needed to meet contingencies.

Suitable donor engines will probably be 'cast enbloc', the cylinders all comprising one casting; such designs also contemplate use of a single cast cylinder head. Thermal expansion of a long cylinder head can bow the engine block, leading to crankshaft and possibly even block failure. Internal combustion engines address this problem with cylinder and head cooling, a steam conversion may have to do likewise and may use the cooling jackets already provided with the donor engine. Selective use of internal insulation may reduce the amount of heat to be carried away, as would careful design of the head to minimize the surface area exposed to steam. An unexpected consequence is that the "uniflow advantage" may diminish; the cool counterflow exhaust cooling the head may reduce the need for applied cooling in comparison to uniflow.

The counterflow exhaust valve poses no particular difficulties; a stock diesel valve should be satisfactory, in both cases the cylinder pressure pushes the valve head against the seat, making for a tight seal until the valve is opened by applying downwards force on the valve stem. The steam admission valve poses a few more challenges, the steam pressure seeks to pop open a stock diesel valve rather than seal it shut. There are a few simple remedies for this situation:

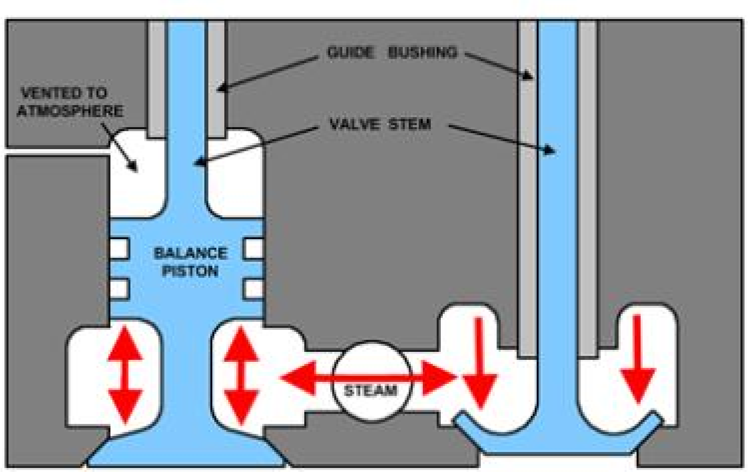
- Employ a bump valve with recompression to cancel the closing pressure.
- Use a very stout valve with an immense spring to withstand the pressure
- Add a counterbalance piston to the valve, causing the steam pressure to work in two directions and cancel out axial thrust
- Build a poppet valve that opens by lifting rather than depressing.

Bump valves contact the cylinder head under spring pressure and are lifted by a pin fitted directly to the piston top. The abrupt contact gives rise to the term "bump valve" although some detractors consider the term "bash valve" more descriptive. Bump valves potentially operate at very short cutoff, and may be tuned to reach a 'sweet spot' around some fixed rpm, a nice trait for a generator. Jay Carter used them to advantage in his steam automobile of the 1970's, and as a subject for development in future projects they show great promise. My concern is the relative lack of data regarding long term reliability. Constant failure of the valve due to the combination of high temperatures and continuous impact led Jay Carter to high strength, non-metallic valve disc materials and specially heat treated, super-alloy springs.



To the best of my knowledge, the only engine to accumulate a large number of hours using bump valves was the White Cliffs solar steam, according to Graeme Vagg they had at least one valve failure although I am unsure under what conditions. While bump valves have their passionate advocates and present a promising subject for development, I am not convinced that, at this time, the bump valve is a certain enough proposition.

The second option requires extremely stout valve stems, springs, rocker arms and cams; in this scenario the operating forces are high and potential for wear is great. We can probably safely dismiss it as an acceptable option. The next two solutions appear more competitive and are illustrated as follows:



A valve with a balance piston is shown to the left, steam pressure applied through the central passage works equally on the bottom of the piston and top of the poppet valve, creating similar opposing forces when the space above the piston is vented to atmosphere and the cylinder is exhausted. As in most engines, the valve stem is lifted shut by a spring and depressed by the cam to open. Recompressing prior to admission presses the valve upwards and a strong force must be applied to the steam to cause it to open.

The valve to the right is simply forced shut against the seat when steam is supplied. Recompressing steam in the cylinder counteracts the pressure on the valve head and reduces the needed operating force.

The counterbalance piston design operates like the exhaust valve and simplifies design of the valve train; a similar valve was used by Dutcher. The lifting valve operates more easily, is more compact and being lighter is subjected to less acceleration forces, making for increased longevity. The lifting valve presents fewer challenges to suppliers of poppet valves. For all these reasons a lifting design poppet valve is recommended. Similar valves are found in the GM SE-101 steamer and the Art Gardiner PSL engine.

Unlike compound engines, the power output of both the uniflow and simple counterflow engine can be controlled by the simple expedient of throttling the admission steam pressure; in the counterflow engine this is accompanied by an efficiency penalty as the pressure is reduced.

STEAM ENGINE PERFORMANCE CALCULATOR

ASSUMES OVERCOMPRESSION RELIEVES TO STEAM CHEST

DATA ENTRY BLOCK

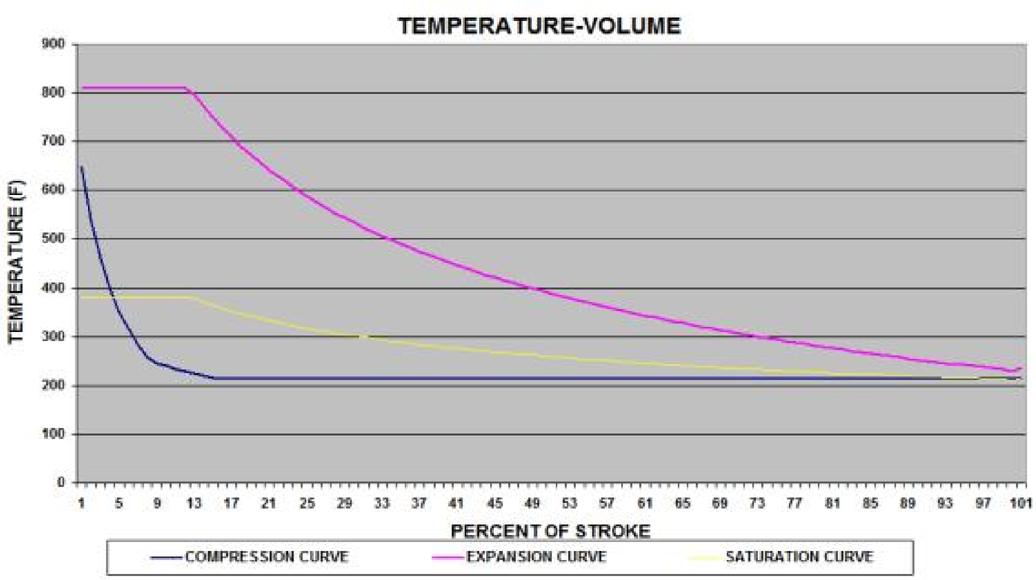
Steam Pressure (psia)	Steam Temp (F)	Exhaust	Jimmy factor
200	850.00	15.000	70.000
468.19 F superheat		30.54" Hg	
Bore (ins)	Stroke (ins)	Cylinders (#)	RPM
4.190	5.000	4	1500
Clearance (% of stroke)	Cutoff (% of stroke)	Release (% of stroke)	Cmprssn start (% of stroke)
2.000	11.500	99.000	15.000

MECHANICAL DATA			
Horsepower	32.20	Total MEP (PSI)	44.03
Expansion MEP:	64.19	Compression MEP:	20.16
Water Rate (lbs/hp-hr)	13.33	Efficiency (%)	13.15
Max piston thrust (lbs)	2,757.71	Average Torque (ft-lbs)	112.73
Water used (lbs/hr)	429.19		

EXPANSION DATA			
Expansion ratio	7.48 : 1	Release PSI	14.54
Release Temp (F)	229.49	Exhaust Temp (F)	212.99
Nominal Cutoff (%)	11.62	Stm Used (lbs/hr)	429.19

COMPRESSION DATA			
Compression ratio:	8.5	Nominal clearance (%)	2.02
Mixing temperature:	809.9	Yields superheat of:	428.1
Compression temp:	648	Yields superheat of:	266.1
Overcompression	Steam chest temp raised to:		846.4
Inches Admiss press reached:		0.008	before TDC
Steam wt recycled (%):	22.49		
Peak pressure (psia):	200		

Note: TV curve generally superheated.



Note that recompression starts at 15% of stroke rather than 20% to prevent excessive efficiency robbing over-compression; the solution appears reasonable as the expansion curve still generally avoids the saturation line. This change in compression can be achieved by either shifting the cam to another profile or by changing the exhaust cam timing through the use of an automotive camshaft phaser. Other options for dealing with over-compression range from auxiliary clearance volumes to check valves which vent over-compression to the steam chest.

If a single camshaft is employed, power modulation can be achieved solely through throttling and a check valve to vent the excess pressure either to the cylinder head although another, more advantageous, option may exist that I haven't examined thoroughly yet.

One misapprehension is that effective cutoff (the mass of steam admitted to the cylinder) is directly tied to the portion of the stroke during which the admission valve is open. We dismissed that, to some extent, by noting flow around TDC will be slight when recompression negates the differential pressure across the valve. The speed of valve movement is also important as we realize by acknowledging that the mass of admitted steam depends partly upon the passage size. Slowly opening and closing the admission valve increases the period when the valve passage chokes flow and effectively shortens cutoff. This choking can be accentuated with changes to the valve shape. The opening duration as well as speed is influential. Like all matter, steam has inertia and takes time to accelerate, as valve speed increases and the opening duration shortens we reach a point where the acceleration lag begins to interfere with steam passing through the valve. This steam acceleration is further complicated if we have recompressed the steam in the cylinder to around the admission pressure and no longer have sufficient differential pressure across the valve at TDC to promote steam flow. Testing may be necessary to determine a cam profile that is sufficiently gentle, provides the correct amount of lift and opening time while still admitting the optimum steam mass at the desired steam pressure and rpm.

Many traditional steam engines inject oil into the steam to lubricate both the valves and cylinder. Poppet valves have very little side loading and function well with a minimum of lubrication, usually a wiper is needed to remove the excess oil deposited on the valve stem by proximity to the rocker arms. However, steam cleaners are built to remove oily buildups, suggesting lubricants may function a bit differently than in an IC engine. Occasional tear downs during testing will reveal whether cylinder, piston, ring and valve wear is excessive; injecting additional oil at the admission valve stem may be necessary.

Piston ring "blow by" can lead to water in the crankcase, which demonstrably shortens bearing life; synthetic oils of high demulsibility, combined with appropriate methods of separation, will require attention and testing.

A simple means of bolting on efficiency would be to add a turbocharger to the steam engine exhaust manifold. The pressurized air produced from the expander steam exhaust can be used to feed the burner and efficiency will rise a bit as this load is no longer derived from the crankshaft.

Miscellaneous Hardware

Condenser

The feed water should be clean (distilled) and chemically treated, this is only economically feasible if the exhaust is condensed and reused.

Compressive condensation is rumored to be under consideration; an idea extensively patented and tested by Harry Percival Harvey Anderson and John McCollum in the 1920s and 30s. Compared to atmospheric condensing, the "Anderson System" reportedly improved system efficiency by about 29%. I would expect less favorable results because the Anderson-McCollum testing was done with locomotives which are less efficient than the engine I am proposing, which leaves less scope for improvement.