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External Combustion Engine Technology (Vapor and Liquid Cycles) for Individual Soldier Power Systems

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13. ABSTRACT *(Maximum 200 words)*

Soldiers of the future will be equpped with electronic and electro-mechanical hardware to enhance their capabilities that will require individual power. There are several potential methods of supplying these power requirements. This report is an initial study of advanced vapor-cycle and liquid-cycle engine systems directed at this application. Results of the study indicated that miniature external combustion vapor-cycle engines in the 1- to 10-horsepower class, employing modern materials and design practices, have the potential to provide a compact, high-efficiency, quiet, and multi-fuel (especially military diesel fuel) power source to meet individual soldier and special operations requirements. A major appendix presents a thermodynamic analysis of the performance of a high-compression uniflow steam engine and the associated Rankine vapor cycle.

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1. Background

1.1 Soldier Enhancement Program

The U.S. Army has recently initiated a program to investigate technologies that could enhance the performance of the individual soldier of the future. Possible materiel enhancements include electronic devices such as individual voice and data communications, navigation and display devices, night-vision and enhanced hearing systems, and weapon ranging and sighting devices; individual climatic conditioning for heat and cold extremes; directed-energy weapons; and systems to enhance mobility and strength, such as individual equipment carriers or exoskeletal devices.

1.2 Power System Requirements

All such hardware items will require power for their operation. The electronics items alone might be powered by advanced batteries. But the climatic conditioning, potential directed-energy weapons, and mobility- and strength-enhancement devices, alone or in combination with the electronics items, will certainly require engine-driven, or at least nonbattery-type power supplies. I have given an estimate of such power requirements in table 1. The specific values are subject to change, as the result of detailed evaluations being done elsewhere. Note that the gross power levels required fall into two classes: an ≈1-hp class (assuming power for all electronics items plus climatic conditioning plus accessory and conversion losses) and a 5- to 10-hp class for exoskeleton or other mobility-enhancement devices.

1.3 Status of Miniature Engines

No engines in these classes are now available in the Army [1] or industry that are suitable for the individual soldier to "wear on his back" on the battlefield. Such engines should preferably provide relatively low power at a low weight and size, with low vibration, thermal, and noise signatures, meeting stringent ruggedness and operational duration requirements, and preferably using the militarily available diesel fuels. Such a combination of features suggests that the solution is to develop special power systems for this kind of soldier application. Appendix A is a complete listing and discussion of the individual requirements, or evaluation factors, for such power systems.

Table 1. Individual soldier power—potential uses

Item	Estimated power		Estimated duty cycle	Estimated energy (per day)		Diesel fuel equivalent (pints at 25%*)	Fuel weight and volume	
	(W)	(hp)	(% day)	(W-hr)	(btu)		(lb)	(in.^{3})
1. Voice & data communication	20	0.03	20	96	328	0.07	0.07	2.0
2. Soldier computer	20	0.03	100	480	1,638	0.37	0.34	10.0
3. Individual navigation	20	0.03	100	480	1,638	0.37	0.34	10.0
4. Night vision (IR) & counter-surveillance	30	0.04	50	360	1,229	0.28	0.26	7.5
5. Enhanced hearing	10	0.01	30	72	246	0.06	0.05	1.5
6. Helmet display	20	0.03	100	480	1,638	0.37	0.34	10.0
7. Weapon ranging, sighting, & control	60	0.08	30	432	1,474	0.33	0.31	9.0
8. Microclimatic conditioning	200	0.27	70	3,360	11,468	2.58	2.41	69.7
9. Individual directed-energy weapon	400	0.54	30	2,880	9,829	2.21	2.07	59.7
10. Mobility-enhancement device	4000	5.4	30	28,800	98,294	22.1	20.7	597
11. Battery recharge	N/A	N/A	As required	Included in basic power		—	—	—
Total of items 1–8	380	0.51	—	5,760	19,659	4.41	4.14	119.5

*Pints/day at 25% efficiency; assume fuel at 19,000 btu/lb, 7.5 lb/gal.

1.4 Army Engine Program

To address this power system deficiency, the Army's Natick Research, Development, and Engineering (RD&E) Center commissioned the Army's Belvoir RD&E Center (BRDEC) to conduct a "front-end" study of potential power systems that might be suitable for the future soldier's needs. This report is a part of that study. A limited level of effort and a short schedule prevented extensive analytical evaluation, so the result is somewhat general and subjective in nature. More time to develop and consult additional reference materials should prove valuable during follow-on investigations.

1.5 Author's Relation to Project

During 1960 the author worked as a mechanical engineer for the U.S. Army Transportation Corps, conducting investigations in the area of "light steam vehicular power." During this period, I developed an appreciation of the potential advantages of vapor- and liquid-cycle external combustion engines, including their potential for compactness, quiet and efficient operation, and ability to burn any fuel. Some of the information used in this report has been taken from reference material gleaned from the light-steam-power project files at that time.

2. Introduction

2.1 Types of Power Systems

Many types of power systems can be considered for individual soldier applications. One possible classification scheme is given in figure 1. (One type not shown is the direct energy-conversion systems—thermoelectric, thermionic, and magnetohydrodynamic.) This report focuses on the continuous external combustion engine technologies where the basic energy source is a petroleum-based liquid fuel.

2.2 Internal Combustion Engines

Small internal combustion engines are available now from the model plane hobby trade. They are inexpensive and they deliver attractive gross horsepower-to-weight and size ratios on the order of 1.5 hp/lb and 3 hp/in.^{3} displacement (see fig. 2 and table 2). However, they have minimal fuel, cooling, and other accessory systems, they run on special fuel (alcohol and nitromethane with oil added), they run at very high speed (10,000 to 30,000 rpm), they are somewhat difficult to silence, and they are not designed for long-duration dependable operation. BRDEC is investigating the necessary modification and possible adoption of this engine technology, in addition to "weed-whacker" type of two-cycle gasoline engine technology, for individual soldier power uses. Both durable (long-life) and disposable (short-life) internal combustion engine systems are being considered.

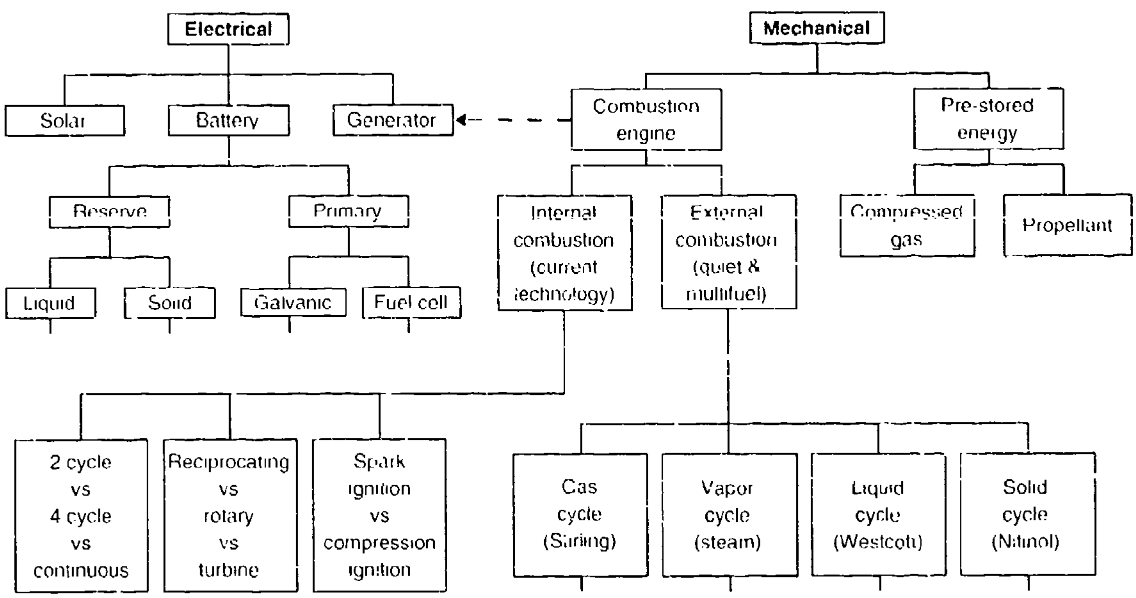


Figure 1. Possible classification of some power systems.

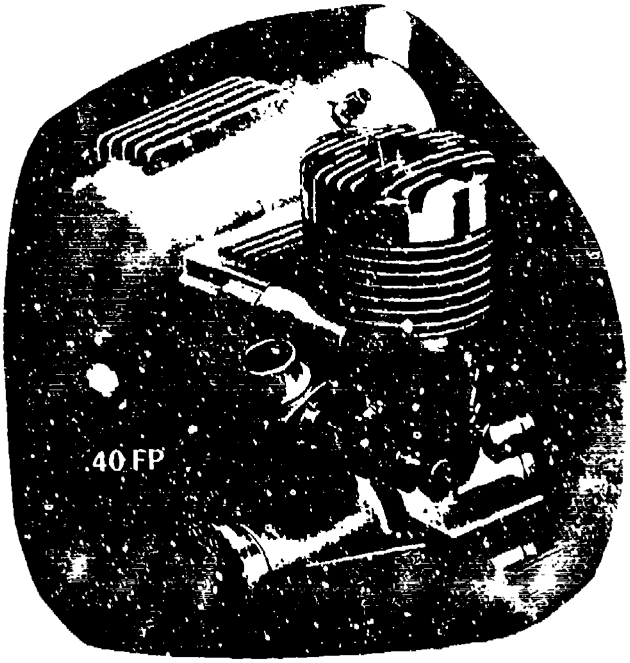


Figure 2. Hobby-type internal combustion engine (Tower Hobbies Catalog, 1991).

Table 2. Typical hobby-grade internal combustion engine performance (Tower Hobbies Catalog, 1991)

Designation	Size (in.^{3})	Weight (oz)	hp	Speed (rpm)	hp/in.^{3}	hp/lb
OS-.21EX-R-ABC	0.21	10.20	1.95	30,000	9.3	3.1
OS-.40FP-ABC	0.40	8.60	1.00	15,000	2.5	1.9
OS-.46SF-ABC-P	0.46	11.30	1.50	15,000	3.3	2.1
OS-.61SF-ABC-P	0.61	20.44	2.00	16,000	3.3	1.6
OS-.91FSR	0.91	25.30	2.50	16,000	2.7	1.6
OS-1.08FSR	1.08	26.50	3.00	16,000	2.8	1.8
OS-FT-120-II, 4C-Twin	1.20	38.60	1.70	10,000	1.4	0.7
OS-120 Surpass SP, 4C	1.20	36.36	2.50	10,000	2.1	1.1

2.3 External Combustion Engines

In contrast to the internal combustion engine, where fuel is burned directly in the working fluid (air), external combustion engines use a working fluid or substance such as water, steam, or air that is heated in one location and then delivers its thermal energy elsewhere, similar to a home heating system. External combustion engine design and development, in the form of the steam engine, preceded development of the internal combustion engine by over 100 years. Its continued development for mobile applications was slowed and essentially halted after the second World War, by the successful development of the internal combustion engine and its supporting infrastructure, the development of convenient electricity from large central power plants, and by the inability of independent (and often intractable) inventors to secure and sustain the financial support necessary to advance the technology [2]. External combustion engines can be contrasted with internal combustion engines as shown in table 3.

Table 3. Contrast of external combustion power systems with internal combustion power systems

Advantages	Disadvantages
Essentially continuous combustion (rather than high frequency intermittent combustion)	Not in common use: "new" technology; developmental status
Very low air pollution in terms of nitrogen oxides, carbon monoxide, and unburned hydrocarbon emissions [3]	Combustion heat must be transferred through a barrier to the working substance
Engine alone operates in thermally insulated environment (compared to water- or air-cooled environment of internal combustion engine)	Requires steam generator (burner and boiler) separate from engine
Relatively low-temperature engine operation, 500–1500°F (compared to fuel combustion temperature in internal combustion engine)	Past practice has been generally inefficient (except for certain designs)
Relatively low-pressure operation (compared to fuel combustion and detonation pressures in internal combustion engine)	Past practice has been somewhat heavy and bulky
Wide range of power available from a single engine size (depends on capacity of heater for working substance)	Proper lubrication is often difficult
Engines develop maximum torque at stall and can be run in reverse to produce braking force (gives transmissionless operation for vehicular applications)	Controls and auxiliary items tend to be more extensive than for internal combustion engine
Pay-as-you-go operation (engine stops when power demand stops) improves economy	Generally necessary to have "closed cycle" operation (recycle working fluid); this tends to create sealing problems
Ability to burn any fuel (particularly diesel fuel)	
Generally quiet operation	

2.4 Why Revive External Combustion Technology?

Because the Army is now faced with a new requirement that will be especially difficult to meet, and one that is positioned in a specialized military field, it seems appropriate to reconsider the special advantages of external combustion technology with an open mind. The advantages of the external combustion engine, particularly its multifuel capability, quiet operation, and low pollution output [3], are attractive enough in view of the individual soldier application (and other applications such as robotic sensors/weapons and special forces communications) that the Army should focus some research effort on seriously exploring its potential.

2.5 How to Revive External Combustion Technology

The potential disadvantages of external combustion systems listed in table 3 can be overcome by modern materials and design practices, such as the use of advanced ceramic and metallic materials, solid lubricants and special coatings, high-performance seals from the spacecraft, nuclear, and water-jet cutting industries, and generally higher operating pressures, temperatures, and speeds than were used in early practice. The fact that the engines are small and intended for service in combination with a generator and battery (and possibly electric or hydraulic drive systems), which can be used to level the electric load and tend toward constant-speed operation, may also prove to be an advantage for this application.

3. External Combustion Engine Types

There are four basic types of external combustion engines, classified according to the working substance used: gas, vapor, liquid, and solid. Another type, not having a shaft output and not considered here, uses an electrically charged aerosol as the working medium [4].

3.1 Gas as Working Substance

The Stirling external combustion power system, named for the Scottish inventor of its high-efficiency thermodynamic cycle, is the primary example that uses gas as a working substance. Gas is attractive as a working fluid in that it is relatively easily handled (within the engine) and does not have appreciable low-temperature storage and starting problems. Another advantage is that it is relatively easy to make the power system operate in any attitude. Potential disadvantages of using a gas as a working fluid are that relatively large quantities need to be transported, and heat transfer is relatively less efficient than for the other types of working fluids. Thus, the engine system's pumping and heat transfer components tend to be somewhat bulky. Also, gases such as helium and hydrogen are desired for improved performance, and their use makes leak-proof seals mandatory. Natick RD&E Center is currently exploring miniature free-piston Stirling engine technology for individual soldier applications under a contract with Mechanical Technology, Inc., of Latham, NY. Natick will report on this type of external combustion technology, so it is not discussed further in this report.

3.2 Solid as Working Substance

At the other end of the working substance spectrum are engines that use a solid as the heat-transfer medium. The primary example is engines made using Nitinol memory metal wires or strips. Many different configurations have been explored [5–9]. They can operate on the reversible thermodynamic cycles, including the Ericsson cycle (constant-pressure heat exchanges) or the Stirling cycle (constant-volume heat exchanges). Two example engine concepts are shown in figure 3. Advantages of a solid working substance are the absence of fluids needing to be sealed, and simplicity of construction. Disadvantages include slow-speed operation (which translates to low power density) and relatively low thermal efficiency: about 10 percent at best, especially at the limited material strains needed to preserve durability. Because of the significant disadvantages of using a solid working substance with respect to the requirements for an individual soldier power system, this type of engine is not recommended for further investigation, and is not discussed further.

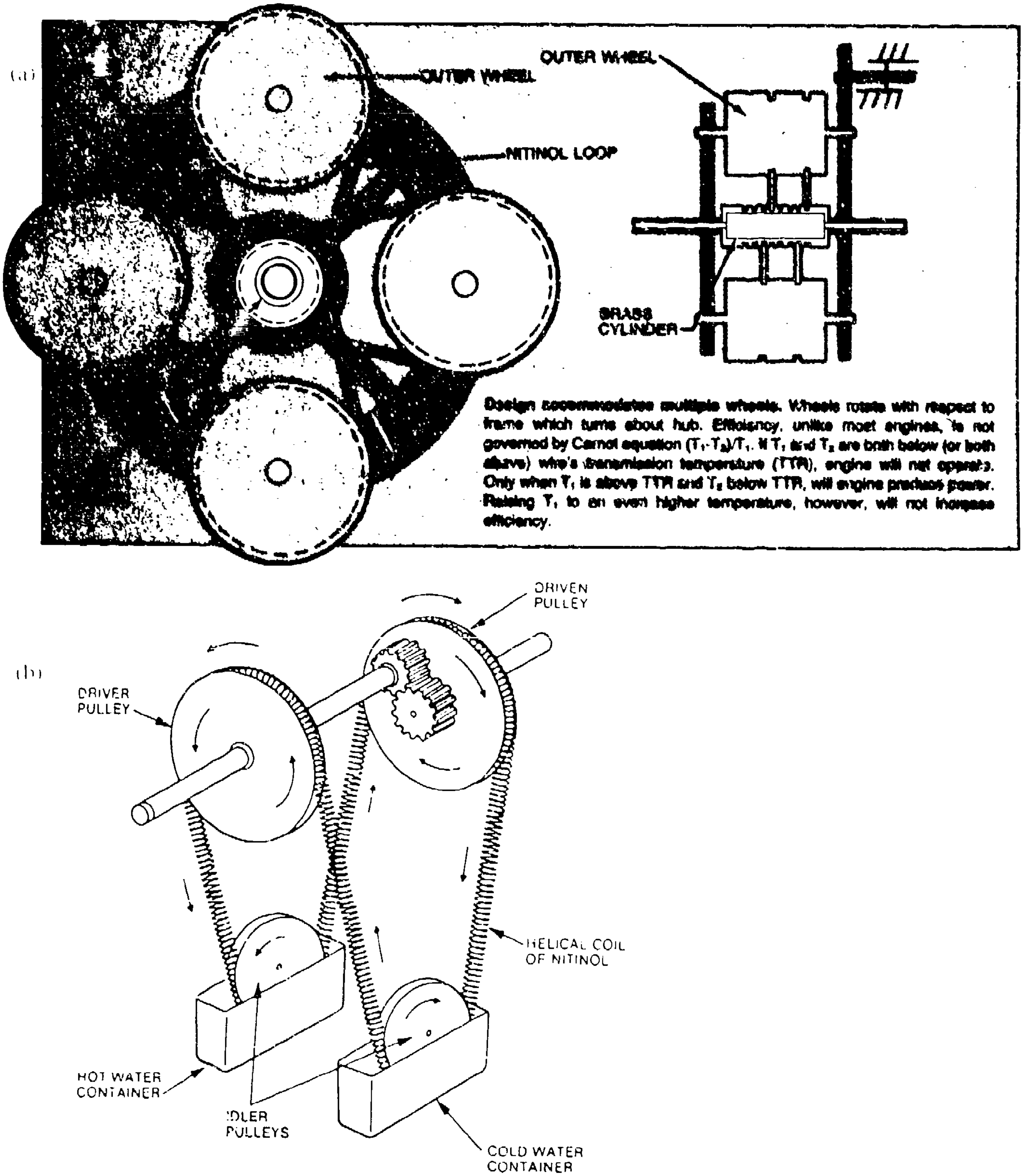


Figure 3. Example external combustion engines having a solid material as working substance: (a) schematic of Fredrick Wang's Nitinol wire prototype 20-W engine [6] and (b) schematic of David Johnson's Nitinol wire engine.

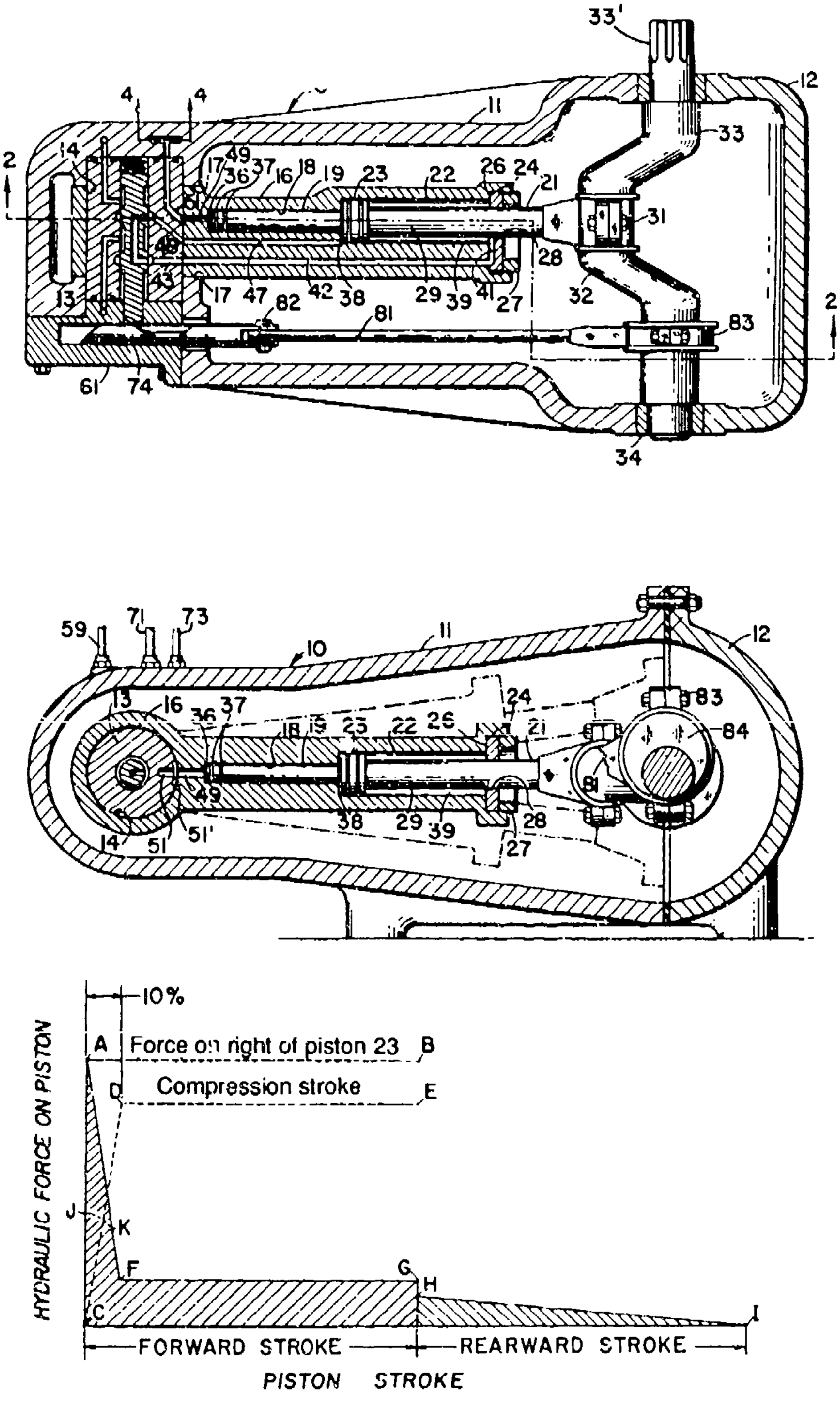
3.3 Vapor as Working Substance

When vapor is considered as the working substance, the primary example is the steam engine. (Some engines have also used fluorocarbon-based and other working fluids.) Vapor-cycle engine systems generally operate by converting the working substance between its liquid and gaseous form. This allows for more compact pumping and heat-exchanger components than when a gas is used as the working substance. Water vapor particularly is readily available, highly characterized, and generally tractable, as demonstrated by its long and successful history of application. Disadvantages of steam as a working substance include poor lubricity, problems accommodating engine operation at below-freezing temperatures, and the need to manage the heats of vaporization and liquefaction when the working substance is constantly converted back and forth between the vapor and liquid states.

3.4 Liquid as Working Substance

Very few engines have been demonstrated that use liquid as the working substance. An engine patented by W. B. Westcott, Jr. [10], and demonstrated by the Cleveland Pneumatic Tool Co. around 1958 to 1960 (fig. 4) is the major example I have found. It is based on compressing, heating, and then expanding a liquid substance such as acetone or an oil. The compressibility of liquids has long been used to advantage in springs for aircraft landing gear and punch press operations [11]. The advantages of a liquid-cycle based engine are compact size because of the high working pressures involved (10 to 20 hp/in.^{3} of engine displacement), no change of state of the working substance, efficient heat transfer, and suitable lubricity (for certain liquids). The major disadvantage is the high working pressure (10,000 to 30,000 psi) and associated high-performance seals needed to achieve adequate compression/expansion ratios.

Figure 4. Liquid-cycle heat engine [10].



4. Steam Engine Technology

4.1 Types/Classification

Hundreds of different configurations of steam engines have been built over the last 200+ years [12]. These include reciprocating, rotary, and turbine (steady-flow) units. Reciprocating engine designs include single-cylinder, in-line, vee, opposed-piston, and free-piston arrangements operating on "two-stroke" cycles. Steam is expanded in a single cylinder or multiple (compound) cylinders, using pistons that are single-acting or double-acting (pressure applied to both forward and reverse stroke). Compound units may reheat the steam between cylinders. Steam has been admitted through slide, rotary, and poppet-type valves, and exhaust can use the uniflow (exhaust at bottom of stroke) or the counter-flow (exhaust at top of stroke) principle. Exhaust can be discarded ("open-cycle" as in railroad steam engines) or recovered in a condenser ("closed-cycle"). Exhaust conditions can be condensing (mixture of water and vapor) or noncondensing (saturated or superheated steam only).

Condensers include vacuum units to allow additional expansion or energy extraction, atmospheric pressure units, and units with regeneration to transfer waste heat to the boiler feed-water. Boilers to generate the steam have been of the fire-tube type (as in railroad and the Stanley steam car plants) and water-tube type (as in central generating stations and more modern vehicular steam engines). Boiler circulation can be by natural convection or forced by pumping, as in the once-through "flash" boilers used in vehicular light-steam power plants and some domestic hot-water heaters.

4.2 Status of Light Steam Power

Except for central power-generating stations and power plants for some large ships, steam or other vapor-cycle engines are rarely used today. The potential advantages of steam power for modern commercial vehicular applications are far outweighed by the disadvantages of attempted competition against the deeply entrenched and highly advanced internal combustion vehicular engine technology. It makes sense to consider vapor-cycle technology only for special applications, such as power for space platforms and new military systems. The individual soldier power systems are just the type of application where a void exists in engine technology, and an open mind can seek fair evaluation of all appropriate alternatives.

A review of past mobile steam power practice reveals that steam lost out to diesel and electric propulsion for railroads because steam locomotive power failed to modernize in a timely manner. The fire-tube boilers employed up to the very end of the era could not operate at the high pressures and temperatures required for efficient thermal performance. The Doble and White steam automobiles and the Bessler Corporation's steam-powered aircraft did use forced-circulation water-tube boilers, and they advanced mobile steam propulsion to a fairly high level. The Doble car was noteworthy for its relatively highly developed automatic control system, and Bessler was noted for his efficient and lightweight boiler designs [13]. However, these companies also did not modernize to the extent necessary to compete. They failed to employ the highly efficient uniflow [14,15] engine design practice (see the excerpt from Barnard et al [12], app B), similar to the U.S. automakers' recent failure to employ, in a timely manner, the multivalve per cylinder internal combustion engine technology advanced by Japanese automakers. Ayres and McKenna have written an excellent book [16] covering all varieties of vapor-cycle engines and other types of potential power systems for vehicular applications.

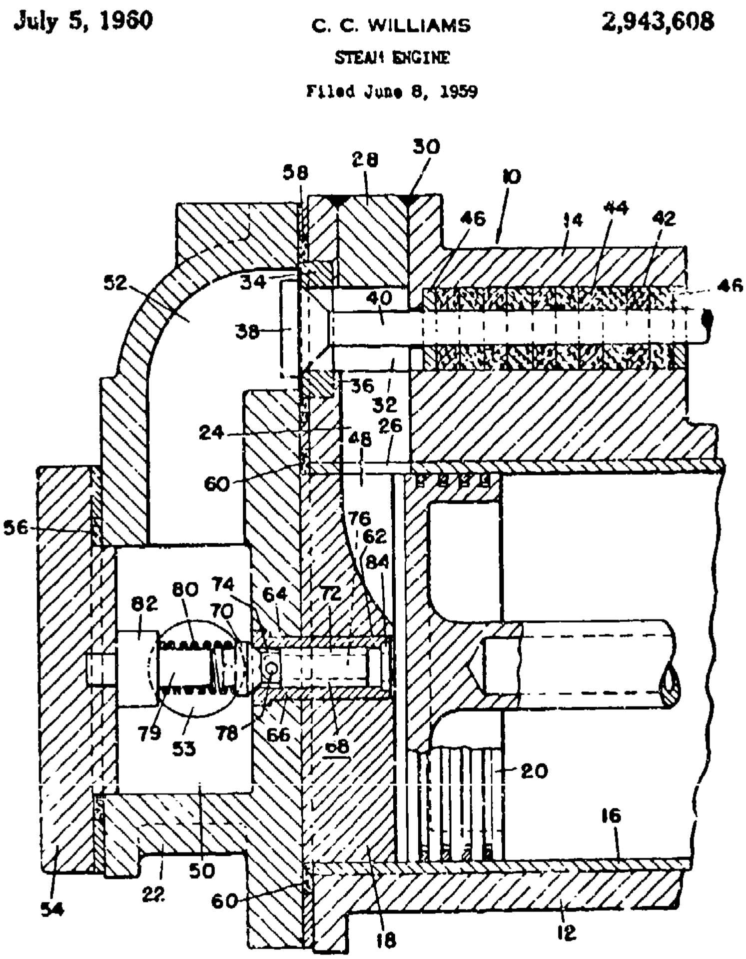
Perhaps the closest anyone came to being competitive in modern vehicular steam engine technology was the engine developed by the Williams Engine Co. of Hatfield, PA, between 1932 and 1970 [17,18]. The Williams engine employed high-pressure (1000 psi) and high-temperature (1000°F) steam, generated in a monotube flash boiler, to drive a high-compression, uniflow, single-expansion, noncondensing, multicylinder, poppet-valve engine at high rpm (see excerpt from Wise [19], app B). The uniflow engine pressure-volume cycle is similar to the air-standard diesel thermodynamic cycle, wherein residual exhaust steam in the cylinder is isentropically compressed to boiler pressure and high temperature, thereby providing ideal conditions for preserving (and perhaps increasing) the energy in the incoming steam. A pressure relief valve built into the cylinder head of the Williams engine (fig. 5) is used to regulate peak pressures, and direct oil injection in the cylinder wall is used to provide proper lubrication under the high steam temperature conditions.

A four-cylinder 56-in.^{3} Williams engine, used to power a city bus through the hills of Pennsylvania, reportedly gave better performance than the original internal combustion power plant. Various lab tests of Williams engines have supposedly given thermal efficiencies in the range from 30 to 38 percent.

4.3 Performance Potential and Resolution of Problems

The Williams high-performance engine technology is suggested as the starting point for the design of miniature advanced vapor-cycle power plants for the Army's individual soldier power applications. It is expected that high overall system performance can be sustained by (1) employing high-speed

Figure 5. Williams engine pressure relief valve arrangement [17].



operation of a small engine; (2) using modern ceramic and metallic materials, modern solid lubricants and antiwear coatings, and modern high-temperature-insulation materials; and (3) employing advanced heat exchanger and burner practice, along with modern electronic control systems, to reduce the size and weight of boiler and condenser components. The recent development of graphite pistons by Mercedes Benz [20], to improve sealing and lubrication and to reduce friction in the internal combustion engine, is an example of modern technology that is expected to be of significant benefit to advanced vapor-cycle engines.

Miniature steam turbine engine technology might also be considered for evaluation against the requirements for the individual soldier power system. This report does not pursue this option, since the very-high-speed operation and the large quantities of steam traditionally required by this type of engine would probably be insurmountable disadvantages for a miniature power system. However, there may be special steady-flow engine designs, unknown to this author, that could overcome these disadvantages.

4.4 Comparison with Evaluation Factors for Individual Power Systems

Table 4 gives a preliminary evaluation of potential vapor-cycle engine technology against the individual soldier power system requirements or evaluation factors from appendix A. The vapor-cycle power system is expected to meet or exceed all requirements, though it will probably be slightly more expensive than an internal combustion engine solution, and special design and operating practice will have to be used to meet the all-attitude and cold-temperature requirements.

Table 4. Vapor-cycle engine preliminary evaluation

Factor	Pros	Cons
Cost	Development costs expected to be little different than for any new miniature high-performance durable engine system. Maintenance costs may be slightly less because of demonstrated long life of steam engines.	Production costs expected to be somewhat higher than for internal combustion engine because of cost of steam generator, special control system, and special materials. Overall, estimate 15–20% premium for this technology over internal combustion engine.
Weight	Weight of this high-performance engine system is expected to be competitive with other technologies of equivalent low noise and diesel fuel performance. Estimated system weight (including fuel) for 24-hr operation at 166 W avg. power is ≈20 lb.	Steel will be basic construction material to get durability and performance at high temperatures and pressures. Not as lightweight as air-cooled aluminum model airplane engines.
Signature: noise, electromagnetic, infrared, visual	Noise is expected to be small (for equivalent suppression) because external combustion systems are inherently quiet (as are home heating systems). Electromagnetic signature is expected to be slight in that there is no high-voltage continuous ignition system required after startup. High-efficiency combustion will give exceptionally clean exhaust.	Infrared signature is expected to be equivalent to other engine systems in that the overall thermal efficiency is similar. Peak engine temperatures are relatively low, but combustion temperature is similar. Signature will be higher than for battery-only system.
Safety	Air pollutants from external combustion systems are the lowest achievable. Pressure of working fluid is not dangerous because of exceptionally small volume and packaging shielding.	Fuel-combustion temperature and exhaust are similar to any engine system. Hot surfaces will have shields and guards.
Vibration, gyroscopic forces	Opposed-piston engine design is inherently balanced to high degree. All components are very small and speeds are moderate for engine of this size.	Vibration and gyro forces should be negligible, though more than for a battery-only system.
Attitude	Expected to be amenable to operation at any attitude, as is normal for aircraft engines.	Design for operation in any attitude is expected to be difficult.
Shelf life	High grade materials should permit indefinite storage time without fluids.	Fuel, oil, and water will be required to be installed upon issue.
Integrated logistics support	Special fuel is not required; diesel is used. System will be designed for a long life as traditionally demonstrated for steam power systems. Low cost and small size will permit easy return and exchange of defective units.	Routine maintenance will be needed as for any engine system. Detailed operation/maintenance manual should permit any engine mechanic to service system.
Reliability/ availability/ maintainability	Maintenance manual can be stored in soldier's computer. Steam engines are traditionally rugged and operate for long periods without maintenance.	Pilot light will be required to protect against freezing temperatures, but all components except condenser are thermally insulated.
Size	A high-performance engine system is expected to fit in a shoe-box size volume of ~0.3 ft^3. Packaging will provide ease of handling and use.	System will probably not be as small as a disposable model airplane engine.
Starting/restarting	Starting will be automatic at the press of a button. Restarting will be automatic, just as it is for home heating systems.	Proper operating temperature for full load should be reached in about 5–10 seconds.
Efficiency	This is an efficient engine system; expected thermal performance is similar to that of a high-performance diesel engine. Nominal fuel consumption is estimated at less than 0.03 gal/hr.	High temperatures and pressures (1000°F, 1200 psi) are required for efficient operation.

5. Vapor-Cycle Engine Approach for Individual Soldier Requirement

5.1 System Block Diagrams

Block diagrams for a steam plant for individual soldier power use are shown in figure 6. It is assumed that a motor-generator would be used for engine starting/restarting and electrical power production. It also serves as the engine flywheel. The system battery would be large enough to meet all system power requirements, not only for engine starting, but also for all operations, including driving the refrigeration compressor, during periods of up to a half hour, whenever "silent" and low-thermal-signature conditions must be met. It also serves as the major load-leveling device to accommodate needed power surges and other transient requirements. The refrigeration compressor for cooling can be driven directly by the engine to save an additional electric motor and improve efficiency. A modular arrangement has been devised, wherein the compressor or other auxiliary items can be attached externally, as required by the mission.

The blower is assumed to fulfill the air-handling requirements for the condenser as well as the burner. The control system box includes all sensors and actuators that would be required for fully automatic control of the various pressure, temperature, and flow-rate functions. A feed-water preheater component (probably part of the condenser) is assumed to provide recovery of available exhaust heat from the engine. An alternative approach to waste heat recovery (which is not considered further in this report) is to employ a miniature exhaust-driven steam turbine to power some of the engine accessories.

5.2 Mission Profile

Preliminary requirements for individual soldier mission power, obtained from BRDEC, are shown graphically in figure 7 for a nominal 300-W system having a 25-percent overload capability. (100- and 700-W systems are also suggested, but the 100-W requirement can probably best be handled by batteries, and the 700-W requirement would be a simple extension of the 300-W system.) The average power and cumulative energy requirements for 6-, 12-, 18-, and 24-hour mission scenarios are also shown, again assuming the nominal 300-W system.

Figure 6. Vapor-cycle power system block diagrams.

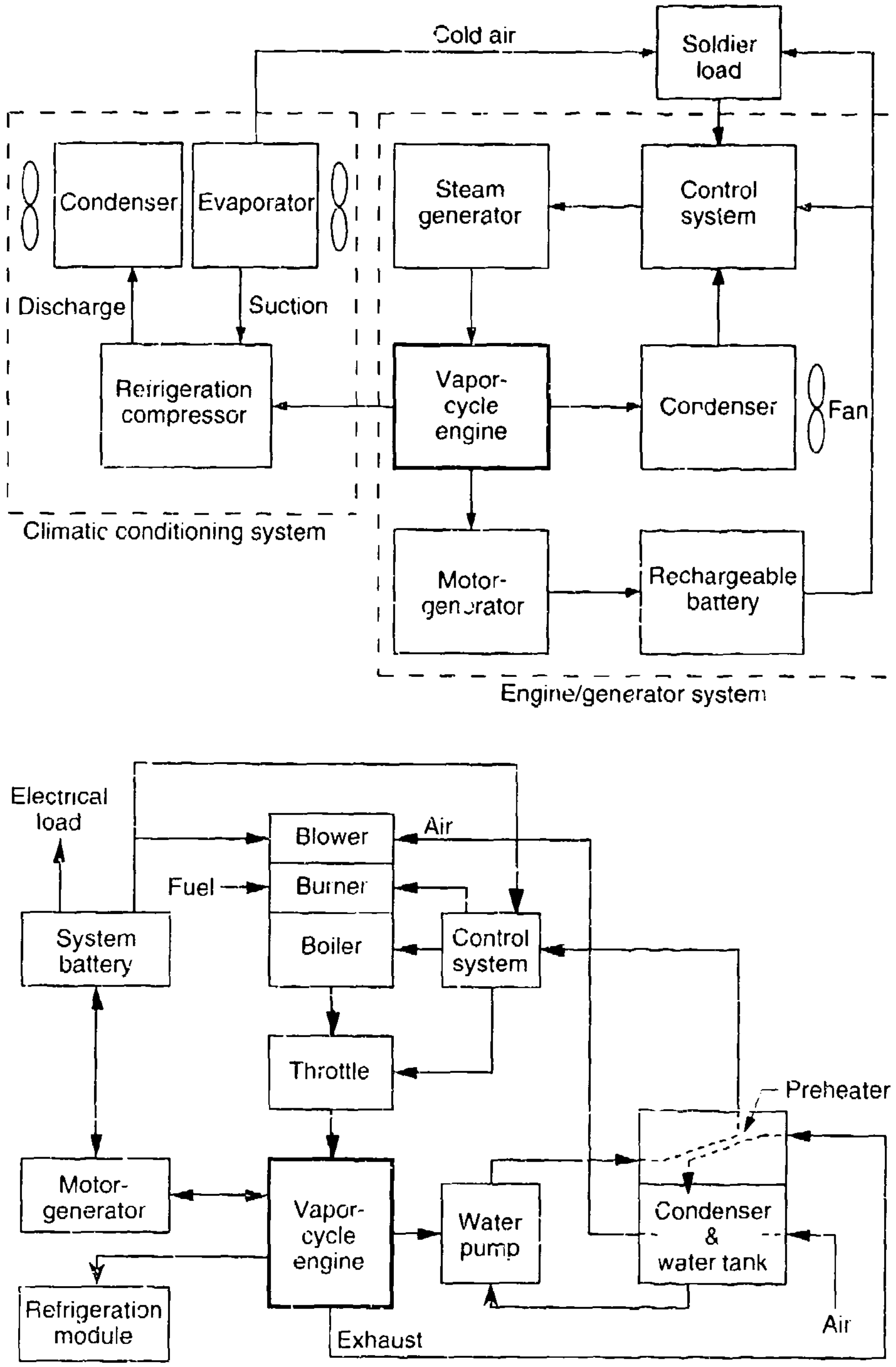
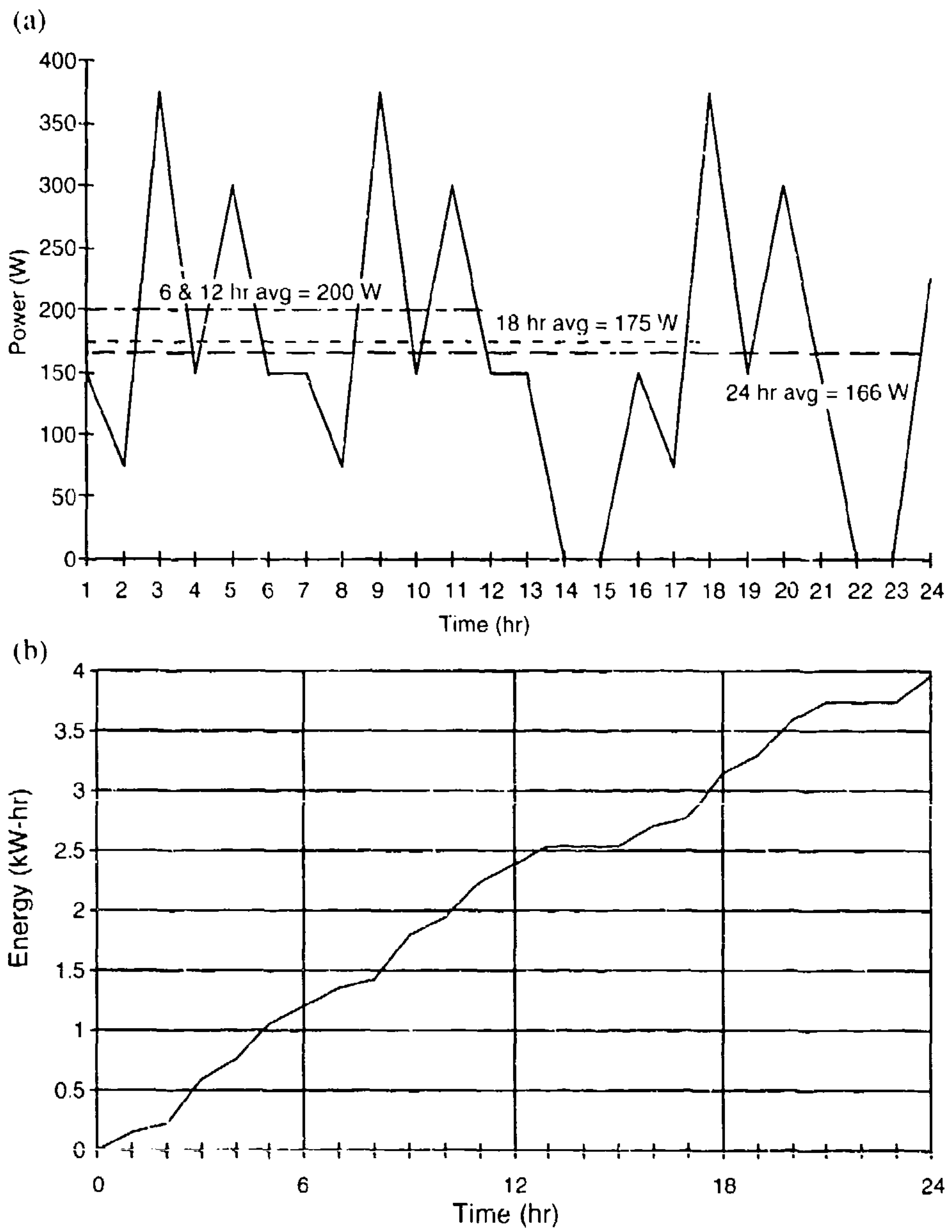


Figure 7. Individual soldier mission—assumed power profile.



5.3 Engine Sizing and Configuration

Appendix C (sect. C-1) shows how engine size can be determined based on assuming a peak output capacity of 375 W, a motor-generator and refrigeration compressor efficiency of 70 percent, an engine speed of 9000 rpm, an engine accessory requirement of 100 W, and an engine mechanical efficiency of 85 percent. These values result in an engine indicated power of 1.0 hp. Compared to efficiency assumptions discussed later (see sect. 5.8), this power level is expected to be conservative by about 25 percent. Volumetric sizing of the engine depends on the indicated mean effective pressure (IMEP) during the engine cycle. If an IMEP of 100 psi is assumed (based on typical practice), an engine swept volume of 0.44 $in.^3$ is obtained.

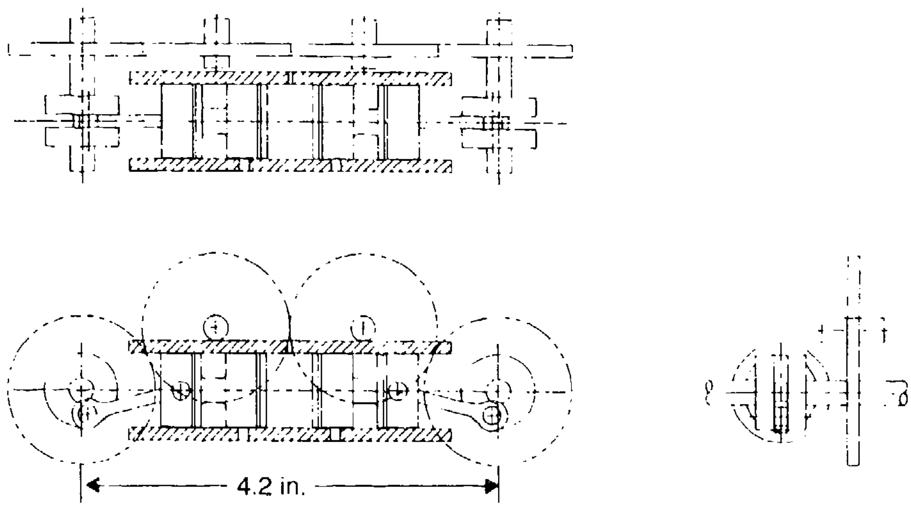
A two-cylinder opposed-piston engine configuration, having a common steam-admission area, is proposed to obtain good control over the high compression ratio desired (≈30–35 to 1), to consolidate the high-temperature portions of the engine for best thermal management, and to provide optimum balance of reciprocating forces. The result is an engine bore of 0.75 in. and a stroke for each piston of 0.5 in. Based on conventional internal combustion engine practice, an engine of this size should not have any problem attaining the 9000-rpm speed requirement.

A full-scale layout of the suggested engine configuration is shown in figure 8. The "cross-head" piston configuration allows for the high-temperature ceramic or graphite piston crown to be separated from the low-temperature section attached to the lubricated connecting rod. The two pistons are synchronized by means of four spur-gears, although a high-performance belt or chain might be considered for this function during detailed design. All gears are shown as the same size, but they can be made different sizes if needed to drive auxiliary loads at higher or lower optimum speeds. The central engine component is about 1.0 in. in diameter and 5 in. long. Its size will increase when the valves, inlet and exhaust manifolds, and certain accessory items are added.

5.4 Steam Generator and Fuel Tank Sizing

Preliminary sizing of the steam generator (app C, sect. C-2) is based on an assumed engine thermal efficiency of 26.5 percent, a boiler efficiency of 87 percent, and an excess boiler capacity of 50 percent. Normally, detailed thermal calculations would be done to determine the boiler size, but a

Figure 8. Basic vapor-cycle engine configuration.



preliminary estimate is obtained from performance parameters reported by the Bessler Corporation [13], a company known from the 1930's for their good monotube boiler designs. Bessler reported demonstrated heat rates of 1.25, 2.0, and 3.0 million btu/hr per cubic foot of boiler volume. Assuming a value of 2,000,000 btu/hr/ft^3, a volume of 14 in.^{3} is obtained. Further doubling the volume to account for the small-size plant yields a steam generator size of 3 in. in diameter and 4 in. long. Peak steam rate is estimated at 9.6 lb/hr, which allows for vapor-cycle thermal efficiency to be as low as 17.7 percent.

I arrived at the preliminary sizing for the fuel tank (app C, sect. C-2) by assuming 633 btu/in.^{3} of diesel fuel, and an average daily power rate of 166 W. These assumptions yield a tank size of approximately 186 in.^{3} ($6 \times 8 \times 3.9$ in.), holding 0.8 gallons.

5.5 Other Components

Table 5 lists the various components assumed for a general engine system arrangement; table 6 gives a summary of various factors affecting the fuel supply, and table 7 summarizes performance of the complete power system. Assumed sizes and weights for the components are also given. Some of these values, especially the weights of the heat exchangers, may not be very accurate. It is recognized that the heat exchanger components (primarily the condenser) are normally a major factor in determining the weight and volume of a steam-powered system. Design calculations need to be done to obtain better preliminary sizes for these components. A wet weight of approximately 21.5 lb is estimated for the power system, when we assume enough fuel for a 4-kW-hr day. An updated estimate of the fuel required is about 0.65 gallon, so the capacity of the fuel tank was made to be about 0.7 gallon. The motor-generator size is an estimate based on HDL's hand-cranked generator developments for the U.S. Special Forces and other information on this subject obtained from BRDEC. The rechargeable system battery, used for starting, load leveling, and a half-hour "run-silent" mode, is conservatively sized at 2 lb and 21 in.^{3} for an ≈200-W unit having a capacity of about 100 W-hr.

5.6 General System Arrangement

A layout of the general arrangement of the preliminary power system is shown in figure 9. It assumes a box of $6.25 \times 8.25 \times 10$ in. (0.3 ft^3) to contain all the system components, including an "L"-shaped fuel tank for 24 hours of operation, and an allowance of 105 in.^{3} for a removable refrigeration module. The engine-generator and vapor-cycle system components are packaged in a volume of 0.14 ft^3. Obviously many different arrangements

Table 5. Estimated vapor-cycle engine component sizes for general system arrangement

Item	Diameter (in.)	Length (in.)	Height (in.)	Width (in.)	Volume ($in.^3$)	Weight (lb)
Motor-generator housing	—	7.00	7.00	5.00	245.0*	0.3
Fuel tank	—	13.25	10.00	1.25	165.6*	0.3
System battery	—	7.00	1.00	3.00	21.0	2.0
Control circuits	—	7.00	7.00	0.50	24.5	0.4
Preheater	—	7.00	1.00	1.25	8.8	0.7
Condenser	—	7.00	5.00	1.00	35.0	0.8
Water tank	—	7.00	1.00	10.25	71.8	0.1
Air filter	—	7.00	5.00	0.25	8.8	0.1
Controller	—	1.50	1.00	1.00	1.5	0.5
Engine	—	6.50	1.75	2.00	22.8	3.5
Cooling module	—	7.00	3.00	5.00	105.0*	TBD
Motor-generator	2.00	2.50	—	—	7.9	2.0
Steam generator	3.00	4.00	—	—	28.3	3.0
Cooling fan	2.50	1.50	—	—	7.4	0.3
Water pump	2.00	1.00	—	—	3.1	0.4
Fuel pump	1.00	1.00	—	—	0.8	0.2
Throttle	0.75	1.00	—	—	0.4	0.2
Miscellaneous parts	—	—	—	—	—	1.4
Water	—	—	—	—	—	0.3
Fuel ≈4 kW-hr	—	—	—	—	151.2	5.0
					Total of * values: 0.3 ft^3	
						Total: 21.5

Table 6. Basic values and conversion factors for fuel system

Parameter	Unit	Value
Volume conversion	$in.^3$/gal	231
Heat/electricity conversion	btu/kW-hr	3,413
Fuel heating value	btu/lb	19,000
Fuel weight/gal	lb/gal	7.7
Specific volume	$in.^3$/lb	30.0
Specific weight	lb/$in.^3$	0.033
Heat per volume	btu/$in.^3$	633
Heat per gallon	btu/gal	146,300
Electrical energy/volume	W-hr/$in.^3$	186
Electrical energy/weight	W-hr/lb	5,567

Table 7. Power system performance summary

Parameter	Unit	Value
Average power required	W	166
Overall thermal efficiency	%	14.2
Heat input required	btu/kW-hr	24,035
Average heat rate	btu/hr	3,990
Fuel required	gal/hr	0.0273
Fuel weight rate	lb/hr	0.21
Fuel volume rate	$in.^3$/hr	6.30

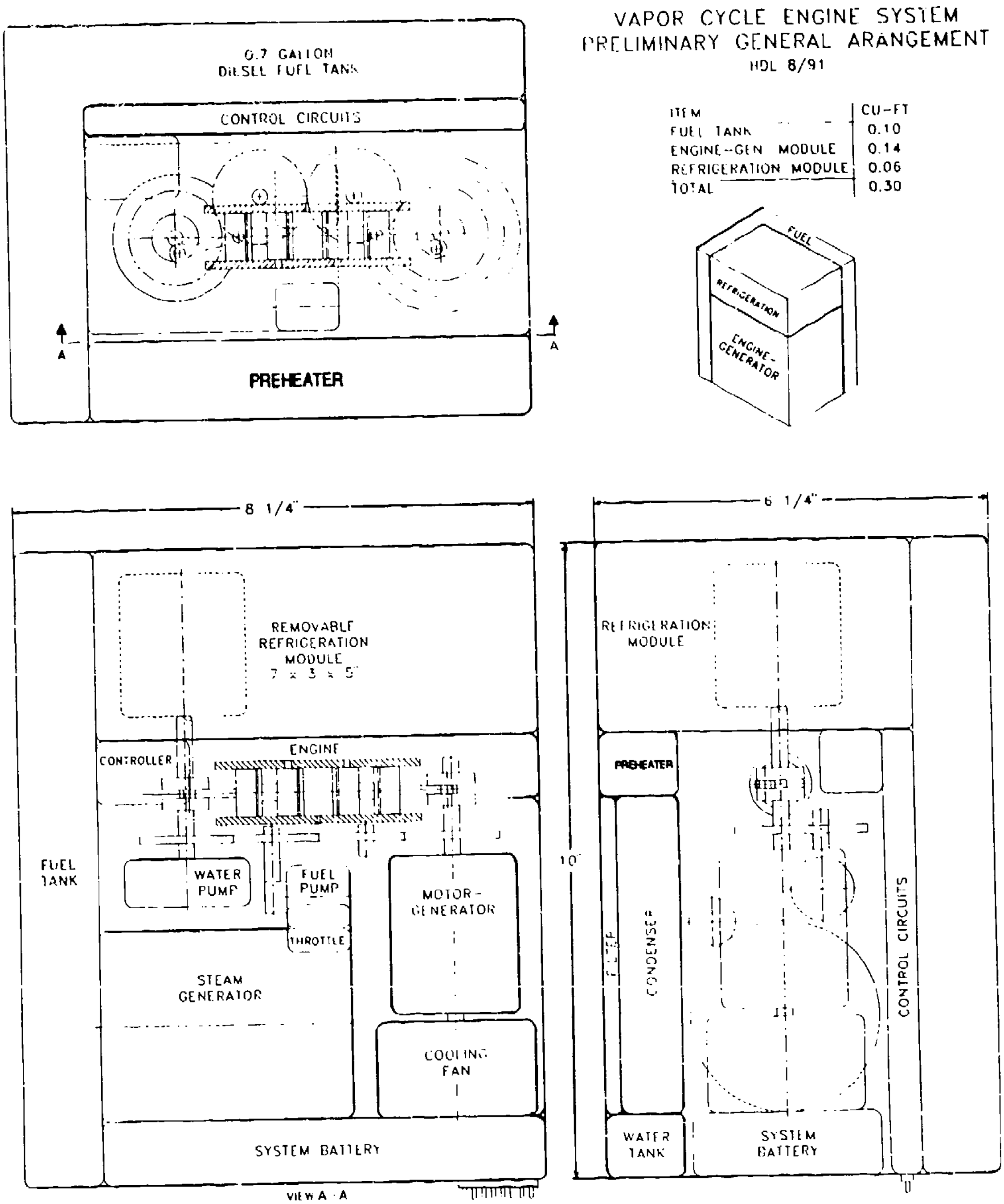


Figure 9. Preliminary general arrangement for vapor-cycle power system.

can be pursued with the goal of optimum packaging. However, more detailed design of the components and their interrelationships needs to be done before an optimum packaging design effort could be fruitful.

5.7 Thermodynamic Analysis

A thermodynamic analysis for the vapor-cycle power system, using a steam engine as an expander, is given in appendix D. Beginning with specified feed steam conditions of 1200 psia and 1000°F, an assumed exhaust pressure of 16 psia, and an engine compression ratio of 28 : 1, the amount of steam supplied to the engine is varied until the desired IMEP, such as 100 psi, is obtained (fig. 10). The resulting portion of the piston stroke during which feed steam is admitted (referred to by the term *cutoff*), the amount of feed steam, and the overall efficiency of the ideal Rankine vapor cycle are subsequently determined.

The analysis was done for several cases of interest, including a half-power case, a double-power case, conditions of higher than normal exhaust back-pressure, and an alternative volumetric compression ratio of 33 : 1. Selected results from the analysis are presented in table 8. The 100-psi IMEP case has a cutoff of 2.5 percent, 6.8 lb/indicated-hp-hr of steam consumption, 11,900 btu/kW-hr energy consumption, and an overall thermal efficiency of 28.4 percent. Additional analysis was done to estimate inefficiencies in the engine cycle, the condenser, feed-water heater, boiler piping, and engine inlet valve. The result was that the thermodynamic efficiency of the actual vapor cycle was 3 to 4 percent below the ideal value. The ideal efficiency exceeded 30 percent at the higher compression ratio of 33 : 1.

Figure 10. Basic engine cycle analysis.

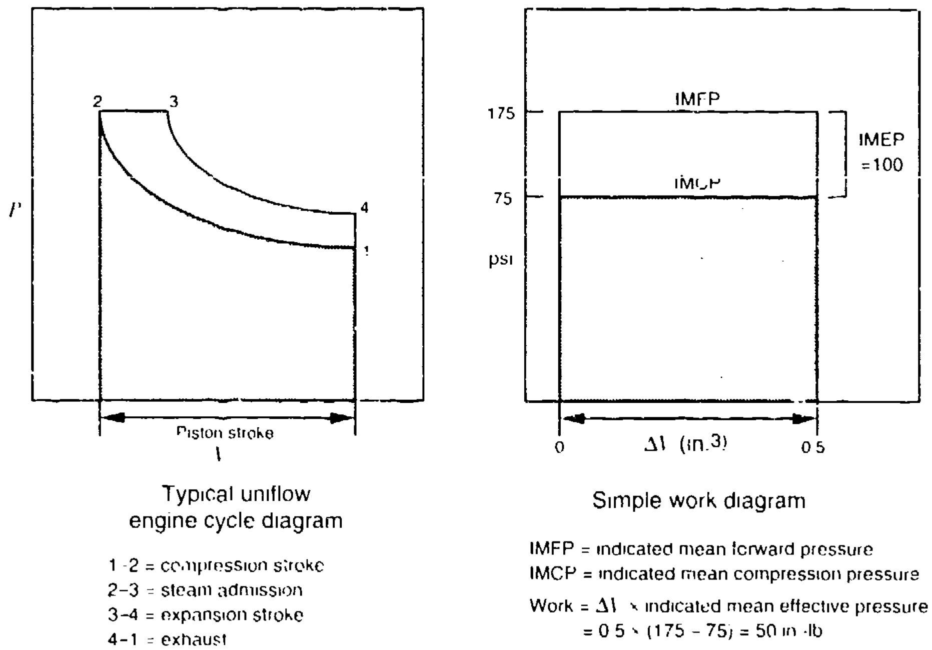


Table 8. Selected results from vapor-cycle system analysis

Case	Units	Half power	Nominal power	Twice power	High-pressure exhaust	Higher compression
IMEP	psi	50.0	100.2	200.1	100.1	99.4
Compression ratio	to 1	28	28	28	28	33
Exhaust pressure	psia	16	16	16	19	16
Entropy	—	1.5742	1.6047	1.6183	1.6263	1.6287
Exhaust quality	%	87.7	89.9	90.8	92.2	91.5
Pressure 2	psia	875.6	926.3	949.3	1169.5	1193.1
Temp 2	°F	775.8	859.7	898.8	984.6	996.7
Feed weight	lb	0.535	1.057	2.207	0.898	1.017
Cutoff	%	0.74	2.52	6.19	3.18	3.15
Cutoff ratio	to 1	1.20	1.68	2.67	1.86	2.01
Temp 3	°F	869.5	940.6	973.4	992.8	998.6
Expansion ratio	to 1	23.34	16.66	10.49	15.06	16.44
Pressure 4	psia	25.97	36.24	60.14	39.35	35.47
Gross work	btu	193.73	397.76	802.52	347.14	404.14
Specific work	btu/lb	361.97	376.22	363.70	386.42	397.52
Net work	btu/lb	358.30	372.55	360.04	382.74	393.85
Water rate	lb/ihp-hr	7.03	6.76	6.99	6.58	6.40
Heat rate	btu/ihp-hr	9214.8	8865.7	9170.8	8572.5	8390.7
Cycle efficiency	%	27.33	28.42	27.46	29.39	30.04

5.8 System Performance

Based on these results, a final overall indicated thermal efficiency for a vapor-cycle system is estimated to be 26.5 percent, assuming an engine operating with the proper compression ratio (probably 33–35 to 1) to optimize efficiency. Another loss to be considered in the overall power-generation system is related to the thermal efficiency of the steam generator, which is estimated at 87 percent. This value was selected on the basis that some modern hot-water heaters and home heating boilers do much better than this.

A further loss is the difference between the indicated engine power used in the analysis to this point, and the engine "brake" power, which accounts for power used to overcome internal friction and related losses, and to power various engine accessories, such as the control system and the condenser-cooling-fan/burner blower. These losses combined are estimated to be about 18 percent of the indicated engine output (compared to the separate but more conservative values of 15 percent for engine efficiency and 100 W for accessory power, which were used to size the 1-ihp engine per app C, sect. C-1). When all the above losses are considered, only about 19 percent of the energy available from the fuel appears as input to drive the load.

A final loss mechanism is the inefficiency in the electric generator and refrigeration system. This is estimated at 25 percent of the power input to these elements. Thus, the overall soldier power plant thermal efficiency is estimated to be

$$E = 100 \times 0.87 \times 0.265 \times 0.82 \times 0.75 = 14.2\% \ ,$$

as illustrated graphically in figure 11. This translates to a required heating value input of

$$\frac{3413 \text{ btu/kW-hr}}{0.142} = 24{,}035 \text{ btu/kW-hr}$$

of actual work delivered to the soldier. For fuel at 19,000 btu/lb and 7.7 lb/gal, the fuel rate for an average power output of 166 W is 0.21 lb/hr and 0.027 gal/hr, as summarized in table 7.

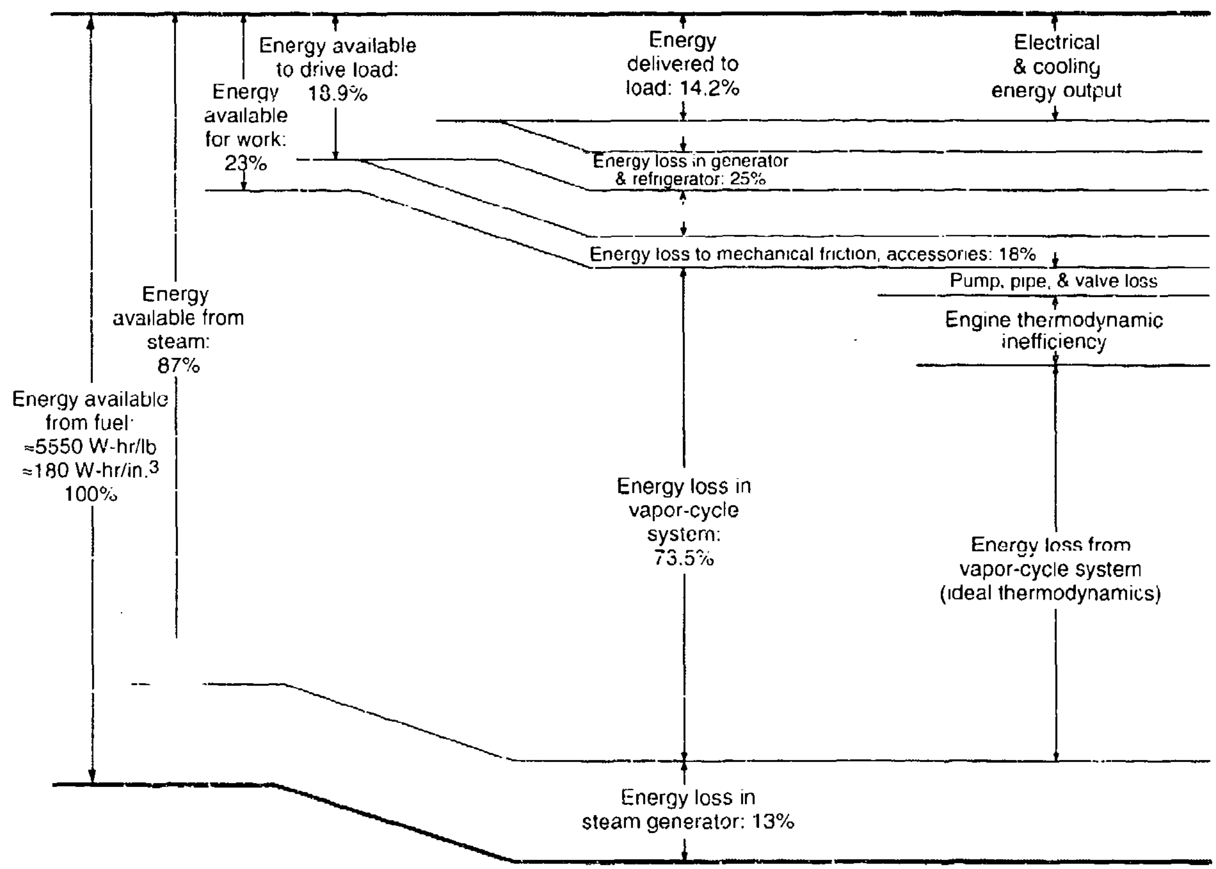


Figure 11. Energy stream for vapor-cycle powered system.

6. Liquid-Cycle Engine Technology

6.1 Summary of Available Information

The status and viability of liquid-cycle engine technology is undetermined at the present time. The limited scope of this study did not permit extensive literature review or design analysis. A report by the Cleveland Pneumatic Tool Co. [21] (and associated patent [10]) is the major item obtained, and it is not available in the public literature. However, the principle of a liquid-cycle engine may be useful in future research efforts, so the information from Cleveland Pneumatic [21] is presented here to stimulate analysis and further evaluation of the concept. Two other liquid-cycle engine patents by Westcott [22,23] relate to this technology, as well as two patents by Malone [24,25] and a recent extension of the Malone concept by White et al [26].

The Cleveland Pneumatic report [21], written in 1960, was a "preliminary" report/proposal [27]. It contains only a few short paragraphs and some illustrations. The relevant passages of this report are as follows:

> Cleveland Pneumatic has been actively engaged in the design and manufacture of liquid springs over a period of the last ten years. These springs work on the compressibility of liquids at pressures up to 50,000 psi. They are currently used on the USAF F-104 in the landing gear shock absorption system and the Polaris shock mitigation system.
>
> The development work on liquid springs led to the study of the possible use of the compressibility and thermal expansion of liquids to convert heat to work. A liquid thermal engine has the potential of delivering large horsepower from a relatively small engine. It has the capability of extracting heat energy and converting it to useful work when relatively low temperature differentials exist.
>
> Over the past two and a half years, feasibility studies have been completed and verified by Dr. R. E. Bolz, head of the Department of Mechanical Engineering at Case Institute of Technology. Dr. Bolz's analysis of one of the proposed fluid cycles, which was completed two years ago, is included as an appendix of this report. The engine performance indicated by Dr. Bolz's analysis has been considerably improved by development work over the past two years.

Figure 12 is a composite of two figures taken from the report [21]. It shows a test setup for compressing, heating, expanding, and measuring the temperature and volume of a fluid, and the resulting pressure-volume diagram from a test conducted on "Cellulube 150" fluid. Figure 13 is a "heat balance" diagram for a liquid thermal regenerative thermodynamic cycle from the report [21], and some simple calculations for efficiency (34.6 percent) and specific engine size (19.8 hp/in.^{3}) condensed from those in the report.

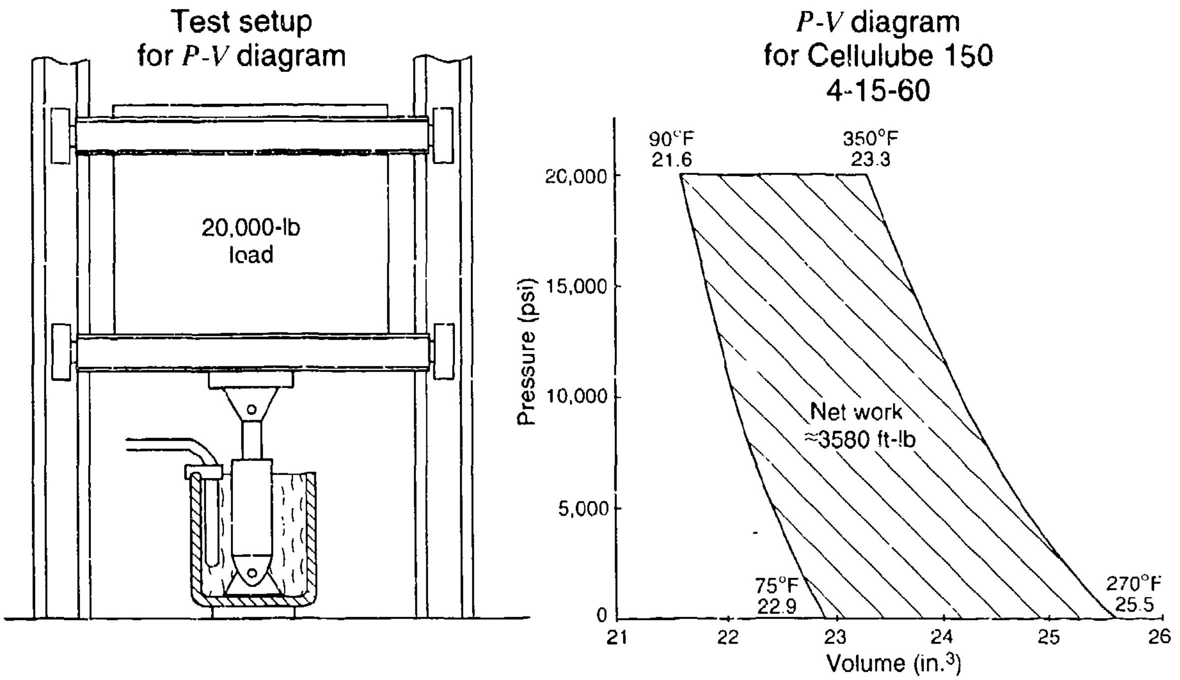
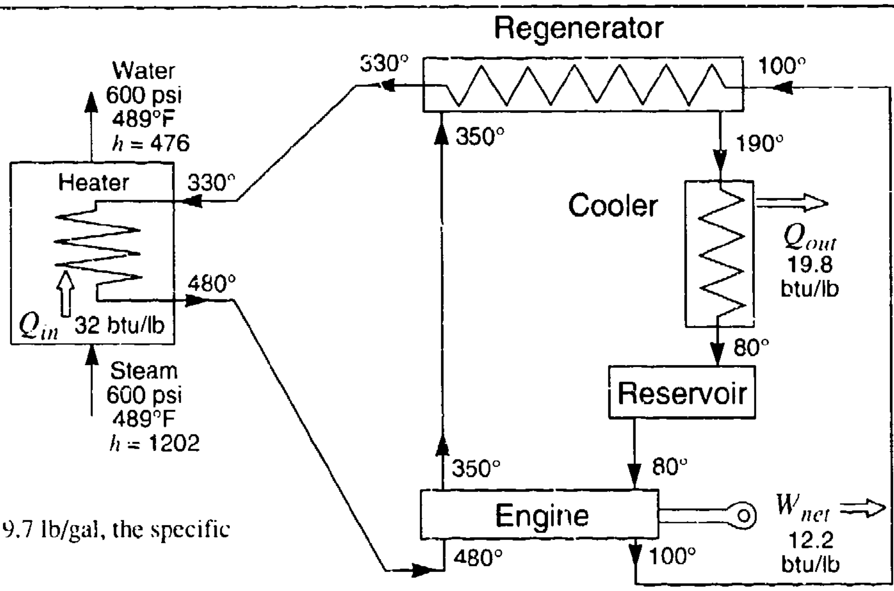


Figure 12. Illustrations from Cleveland Pneumatic Tool Co. paper [21] on liquid thermal engine.

The values in figure 13 appear to be speculation on the part of the author of the Cleveland Pneumatic report [21]. The efficiency is about twice as high as the values derived by Bolz, whose analysis is reproduced in appendix B (sect. B-3). Also, the value of the heat input, Q_{in} (32 btu/lb), when divided by the 150°F temperature differential shown for the heater, results in a specific heat value for the fluid that is about half as large as expected for a petroleum-based fluid. This small specific heat, when used in the simple analysis of figure 13, would yield an efficiency about twice as large, and an engine size about half as small, as would an expected value of specific heat. Furthermore, the indicated efficiency of 12.2/32 = 38.1 percent seems too close to the theoretical Carnot efficiency of 42.5 percent for the specified temperature difference (480 – 80 = 400°F) to be realistic.

Bolz's analysis (app B-3) refers to "a separate enclosure" and "the original/referenced report" in support of part of his analysis. These items were not available. He also uses values for the properties of acetone taken from an article by P. W. Bridgeman [28]. Six graphs of these acetone data are included with the report [21], but they are not reproduced here.

Representatives from Western Gear Corporation, a contractor for the Army's Transportation Research Command in 1960–61, visited Cleveland Pneumatic Tool Company at that time and observed an experimental liquid-cycle engine in operation. A recent conversation [27] with one of the mechanical



For fluid at 9.7 lb/gal, the specific volume is

$$\frac{231 \text{ in.}^3\text{/gal}}{9.7 \text{ lb/gal}} = 23.8 \text{ in.}^3\text{/lb} \ .$$

The flow rate for the specified 12.2-btu/lb output is then

$$\frac{2545 \text{ btu/hp-hr}}{12.2 \text{ btu/lb}} = 208.6 \text{ lb/ihp-hr} \ ,$$

$$208.6 \text{ lb/ihp-hr} \times 23.8 \text{ in.}^3 = 4964.8 \text{ in.}^3\text{/ihp-hr} \ .$$

Assuming an engine speed of 1800 rpm, the indicated power per in.^{3} is

$$\frac{1800 \text{ rpm} \times 60 \text{ miles/hr}}{4964.8 \text{ in.}^3\text{/ihp-hr}} = 21.7 \text{ ihp/in.}^3 \ .$$

Assuming that 1.1 indicated horsepower yields 1.0 brake horsepower, the brake horsepower per in.^{3} is

$$\frac{21.7 \text{ ihp}}{1.1 \text{ ihp/bhp}} = 19.8 \text{ bhp/in.}^3 \ .$$

Then the system thermal efficiency is

$$\frac{\text{work out}}{\text{heat in}} = \frac{12.2/1.1}{32.0} = 34.6\% \ ,$$

and the Carnot efficiency is

$$1 - \frac{80 + 460}{480 + 460} = 42.5\% \ .$$

Figure 13. Heat balance and calculations for regenerative liquid thermal engine from Cleveland Pneumatic Tool Co. report [21].

designers on the liquid thermal engine project at Cleveland Pneumatic revealed that they did not have problems with the high-pressure seals on their experimental engines. However, they did have problems with inefficient heat-exchanger performance that limited engine speed and reduced thermal efficiency to much less than 10 percent.

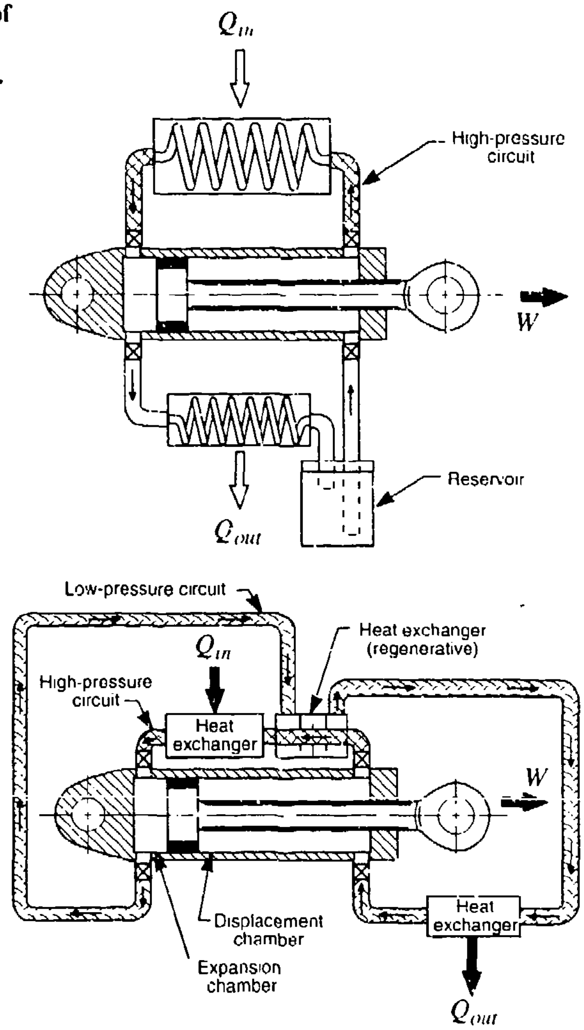
6.2 Engine Description

The cycle of the liquid thermal engine consists of introducing a cooled liquid at near atmospheric pressure into a cylinder and adiabatically compressing the fluid with a piston to a high pressure (approximately 30,000 psi). The act of compression increases the fluid's temperature. The fluid is delivered to a high-pressure heat exchanger, in which the fluid temperature is further raised at constant pressure. The heated fluid is introduced to a cylinder where it is adiabatically expanded against a piston to the original low pressure, and in doing so performs useful work. Part of this work consists of compressing the cooled liquid as mentioned initially, and the excess energy is used as a prime mover. At the end of the cycle, the expanded fluid is exhausted, cooled, and transferred to the compression chamber, and the cycle repeats. The operating cycle of the engine is two-stroke. The up-stroke exhausts the expansion cylinder and permits or causes the compressing cylinder to be filled with cooled liquid. The down-stroke, caused by the expanding liquid, compresses the cooled liquid and also delivers the engine output.

As shown in figure 14, a composite of the simple and the regenerative engine schematics from the Cleveland report [21], and also in figure 4, the cylinder/piston combination is of the conventional double-acting type. The cylinder volume at the piston rod end is used for compression, and the larger volume at the other end of the cylinder is used for expansion. This balances the force of compression so that only net work is delivered to the crankshaft. The expansible fluid is delivered to the cylinder, by means of suitable valves, when the piston is near the top of its stroke. After a specific amount of fluid is admitted, the valve closes and the compressed fluid expands to force the piston to the end of the stroke. Because energy in the form of heat is added to the fluid in the heat exchanger, the expanding fluid exerts a greater thrust on the piston than required to compress the fluid below the piston. This extra thrust is used to perform useful work because the piston rod is coupled to a crankshaft and flywheel system. Flow of the fluid through the various stages of the cycle is accomplished by valves timed from the crankshaft rotation.

Acetone was chosen for Cleveland Pneumatic's initial engine design because of its relatively high coefficient of thermal expansion. However, other liquids, such as silicon oil, may be more desirable in that they could be nonflammable, noncorrosive, and self-lubricating, could reduce leakage, and could provide improved thermodynamic properties. Another possibility would be to introduce a small amount of an inert gas such as helium into the working fluid, so as to favorably affect compressibility and expansion.

Figure 14. Schematics of liquid thermal external combustion engine [21].



6.3 Advantages and Disadvantages

Advantages of a liquid-cycle thermodynamic power system could be as follows:

- Relatively low working temperatures, of about 400 to 800°F
- Relatively efficient heat exchange due to liquid-to-metal interfaces
- Quiet operation, owing to steady-state condition of the working fluid
- Small engine size due to the high mean effective pressure
- Relatively slow speed operation, of about 1000 to 3000 rpm
- Simple starting and operation

Disadvantages of the liquid-cycle system would be expected to relate to the following:

- Efficiency may be low because of the low peak-temperature level, although the extreme pressure conditions may offset this effect to some extent.
- High operating pressure will require special sealing techniques and possibly result in high seal friction forces.
- At least one high-pressure heat exchanger will be required, and this will add weight to the power system.
- Throttling of fluid through high-pressure valves may result in high energy losses and unacceptable wear of seating surfaces.
- Relatively low coefficients of expansion and small compressibility of the fluid could result in some critical relationships between respective volumes within the engine and heat exchanger elements, and may require rather close control over some of the temperature conditions.

7. Liquid-Cycle Engine Approach for Individual Soldier Requirement

The liquid-cycle external combustion system is similar to the vapor-cycle system in terms of engine sizing, types of components, etc. Assuming the lower limit of Bolz's analysis value of about 10 hp/in.^{3} at an engine speed of 1000 rpm, an engine of 1 hp (indicated) would have a cylinder volume of only 0.1 in.^{3} at this speed. A more realistic speed for an engine of this size would be 3000 rpm. The result is a displacement of 0.033 in.^{3}, which translates to a bore and stroke for a single cylinder of about 0.25 in. diameter by 0.67 in. long.

Although heavy-walled structures are needed to contain the very high pressures involved, the small scale of the components is an advantage, in that small size results in higher strength-to-weight ratios. Thus, the weight penalty normally associated with high-pressure systems may not be as significant for a system sized to produce a net output of only 0.5 hp.

The general arrangement for a liquid-cycle power plant would probably be similar to that for the vapor-cycle plant shown in figure 9. The names of the components would be modified appropriately, the engine configuration would change to be similar to that shown in figure 4, and a regenerative heat exchanger might be added. The heat source would probably need to be designed to achieve close control over the fluid temperature, and a hydraulic accumulator may need to be added. A modern synthetic fluid, perhaps with some entrained gas, would probably be used as the working substance. The overall weight of the power system is expected to be close to that of a comparable vapor-cycle system, whereas the size of the system might be smaller.

Designing a liquid-cycle engine and power system is expected to require significant ingenuity. There is relatively little standard practice on which to base the design. However, there is a good body of literature on high-pressure technology (see Tsiklis [29], for example), and much progress has been made in the last two decades in high-pressure water-jet cutting technology [30], where the pressures used are similarly high. The task should begin with a study to select and characterize the best fluid or fluid system to use in the design. This should be followed by a rigorous analysis of the engine and the entire liquid thermodynamic cycle, including realistic energy-loss mechanisms for each of the components. If the concept survives to this point, then detailed design of a research engine should be done, followed by construction and testing of an experimental model.

8. Conclusions

Table 9 is a preliminary comparison of the vapor-cycle system with presumed Stirling and internal combustion power systems for the individual soldier. It shows that the vapor-cycle system is essentially competitive with these alternative technologies, and it could be a better choice depending on the specific mission. At the current stage of development of these concepts, comparative estimates have so much probability of error that such detailed conclusions are unwarranted, other than to point out that additional research needs to be done on all three candidates.

The vapor- and liquid-cycle engine technologies potentially offer a distinct combination of advantages for soldier system power. These are quiet and efficient operation, diesel fuel compatibility, compact size, a broad power range, and long-duration mission capability. However, these advantages probably come at a slight penalty in cost and weight compared to an internal combustion engine system. Another consideration is the fact that these technologies have to be developed much further than does the internal combustion technology, which already has a partially established industrial base.

The vapor-cycle engine technology is especially viable and of reasonably low development risk. Selection of the proper baseline design and the application of modern component technology is expected to eliminate any problems or disadvantages residual in the prior art. Liquid-cycle engine technology is not nearly as advanced as vapor-cycle engine technology, so a larger investment in research will be required for its potential to be fully explored.

Table 9. A comparison of internal combustion, Stirling, and vapor-cycle power plants

Evaluation factor	Weight[a]	Internal combustion engine[b]		Stirling		Vapor cycle[c]	
		Raw[d]	Weighted	Raw[e]	Weighted	Raw	Weighted
Cost	18	70	1260	100	1800	110	1980
Weight	14	85	1190	100	1400	115	1610
Signature[f]	9	115	1035	100	900	100	900
Size	9	70	630	100	900	80	720
Safety	4	105	420	100	400	105	420
Forces	4	115	460	100	400	100	400
Attitude	5	110	550	100	500	120	600
Shelf life	7	100	700	100	700	100	700
ILS & fuel	8	135	1080	100	800	100	800
RAM	7	130	910	100	700	110	770
Starting	5	110	550	100	500	100	500
Production base	6	70	420	100	600	100	600
Power range	4	80	320	100	400	70	280
Totals	100	1295	**9525**	1300	**10,000**	1310	**10,280**

[a]*Factor weights are subjective, as are raw scores.*
[b]*Internal combustion engine is modified single-cylinder model airplane type, running on special fuel.*
[c]*Vapor cycle is opposed-piston reciprocating engine and shaft-driven generator.*
[d]*On raw score, lower is better, higher is worse.*
[e]*Baseline is a perceived Stirling system which is a hermetically sealed free piston design with internal linear generator.*
[f]*Bold factors are strengths of vapor-cycle system.*

Comments

Examples: for unweighted values, cost of internal combustion (IC) engine is expected to be about 30% lower than Sterling and 40% lower than the vapor-cycle (VC) engine; the VC engine is expected to be about 20% smaller than the Sterling and 10% larger than the internal combustion; production base for the IC engine is anticipated to be about 30% better than for the external combustion systems, but fuel and ILS problems are expected to be about 35% greater; etc.

Conclusion: All three systems are within 1% on the total of the raw scores, but the IC engine comes out about 7.5% ahead of the VC system when weights appropriate for the baseline scenario of the soldier system are applied.

9. Recommendations

Because the Army is faced with a need that is new and especially difficult to meet in a specialized military field, it seems appropriate to reconsider the special advantages of external combustion technology with an open mind. The particular advantages of multifuel capability and quiet operation are attractive enough in view of the individual soldier application that the Army should focus some research effort on seriously exploring its potential. It is recommended that the Army explore both vapor- and liquid-cycle technologies, by analysis and laboratory experiment, to the extent necessary to validate their capabilities and to establish their viability for applications that may require their unique combination of characteristics.

The following topics are recommended for university research projects to explore and advance the technology of high-performance vapor- and liquid-cycle external combustion engine power systems for individual soldier applications of the future. Each of the topics is visualized as a modular task at the level of effort of a graduate student thesis.

- Refine a mathematical model of a high-performance uniflow steam engine, including detailed energy-loss mechanisms, and determine the best design characteristics for the target application. (See recommendations in app D.)
- Refine a mathematical model of a liquid-cycle engine, including a study of optimum working fluids, and determine the best design characteristics for the target system.
- Design a miniature once-through "boiler" for heating the working substance at optimum efficiency and power density.
- Investigate miniature high-performance burner technology and design an optimum burner system for use with a miniature power system boiler.
- Based on the most advanced heat-transfer component technology, design miniature high-performance air-cooled heat exchangers/condensers tailored for application to vapor- and liquid-cycle power systems.
- Analyze the performance of Rankine vapor- and liquid-cycle power systems based on the detailed design analysis done for their expander, heat-exchanger, boiler/burner, piping, valving, pumping, and insulation components.
- Design an optimized control system for fully automatic and safe operation of a miniature vapor-cycle power plant.

- Investigate optimum materials (metals, ceramics, thermal insulation, lubricants) and seal technology for application to durable, high-pressure, high-temperature, steam- and liquid-cycle power system components.

- Investigate design configurations for the most practical method of achieving closed-cycle operation (with no loss of working fluid) and any-attitude operation of vapor- and liquid-cycle power plants.

- Demonstrate miniature steam engine technology by designing, building, and testing an experimental engine of 1 to 2 indicated-hp capacity, including appropriate steam generator, condenser, etc. components, that would be suitable to power a model airplane. (This topic would require a level of effort equivalent to three or four of the other tasks.)

- Demonstrate miniature liquid-cycle engine technology by designing, building, and testing a laboratory experimental engine of 1 to 2 indicated-hp capacity. (This topic would require a level of effort equivalent to three or four of the other tasks.)

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Appendix A. Evaluation Factors for Individual Power Systems*

**The material in this appendix has been adapted from a similar document prepared by the Belvoir Research, Development, and Engineering Center (BRDEC).*

Contents

A-1. Introduction

The selection of potential technologies for use in individual power systems for the future soldier must take into account all the various factors that have been used to evaluate military power systems over the past several decades. Though the factors used here are generally the same as those used previously, their relative importance is tied to the individual soldier application. The military application and the fact that the system will be worn or carried by the individual soldier makes these power supplies considerably different from those used for most commercial purposes. The following factors are considered to be important for the early stage of development:

- Cost
- Weight
- Signature
- Safety
- Vibration/gyroscopic forces
- Attitude effects
- Shelf life
- Integrated logistic support (ILS)
- Reliability/availability/maintainability
- Size
- Starting/restarting
- Efficiency

These factors are not truly independent, since there is a large degree of interaction among them. They are meant to guide the selection of designs and technologies away from those that are obviously not appropriate for the individual soldier application. They should not be considered to preclude the investigation of promising power systems, just because some of the requirements cannot be met today. However, when a particular power approach is deficient, any effort to correct the deficiency should be considered from the aspect of cost and realism.

Before discussing each of the factors, it is helpful to discuss the military environment to help point out the extreme difficulty of meeting all the possible requirements. The military environment is taken to mean any environment where the soldier will use the power system. It covers the temperature extremes found worldwide and altitudes from below sea level to thousands of feet above sea level. One consideration is that the density of air (and the oxygen content) can vary by a factor of two from sea level and –65°F, to 8000 ft and 95°F. Other military environment concerns are in the areas of blowing dust/sand, salt fog/spray, and chemical/biological agents. These environmental conditions must be considered when applying several of the evaluation factors to various power supply candidates.

A-2. Cost

The cost factor is perhaps the least definable yet most important factor in the long term. Although high cost may be acceptable when enhanced performance is attained, the expected decline in future Army budgets makes cost a prime consideration. Life-cycle cost is related to many of the other factors delineated below. Considering the technological barriers, it may be a matter of whether it can be done, rather than how much it will cost. If a particular technology has prohibitive costs at this time, a manufacturing methods and technologies program could be instituted. If material costs are prohibitive, research programs aimed at specific items will be needed. The cost will be considerably affected by the quantities, which cannot be known at this time.

A-3. Weight

The weight of the power system must be minimized. The weight of all the various pieces of equipment that the soldier must carry on the battlefield is already a substantial burden. The addition of various new pieces of equipment will not relieve the need for much of the current inventory, particularly in the non–power-consuming area. The weight factor is influenced by power requirements (peak, continuous, and average) and by the time between resupply/refueling. Weight should be considered from the perspective of total mission weight, which includes unit weight, the weight of fuel, and the weight of any other expendables required for missions of various lengths.

A-4. Signature

Several categories of signatures may be affected by a power source and need to be evaluated.

Noise signature. The noise generated by the power source can be further broken out into noise that is a health hazard, noise that will interfere with communications, and noise that will render the soldier detectable by enemy forces. General policy states that personnel should be provided an acoustic environment that will not cause personnel injury, cause fatigue, or in any other way degrade overall effectiveness. The first two categories of noise are covered by various human-engineering documents. The third category, detectability, is more difficult to specify, since it depends on environmental conditions, frequency spectrum, and the capabilities of the opposing forces. The future threat may be equipped with amplified, frequency-selective enhanced-hearing devices, so the power system noise should not contain bands or characteristics that are significantly different from the natural background.

Electromagnetic signature. This category involves the same three issues as the noise discussed above. A health hazard associated with the power source is not considered to be probable but does need to be considered. Electromagnetic signals can interfere with communications signals either sent or received by the soldier. The soldier will also be vulnerable to detection if electromagnetic emissions are present.

Infrared signature. One of the most difficult signatures to suppress for a power source is its infrared (IR) signature. No power source is 100-percent efficient, so that some waste heat is given off. Present IR detection devices are sophisticated enough to detect, at close range, small objects that are only a few degrees above or below ambient conditions. Future opposing forces could have quite formidable IR detection capabilities, so the power system will need a high degree of suppression in this area. Generally the IR signature should be patterned to simulate background environmental conditions.

Visual signature. Detection by visual means is a threat that is continual and requires no special equipment, although the enemy may use vision enhancement to aid detection some of the time. This threat can be defeated by typical camouflage methods; however, designs should avoid obvious shortfalls such as shiny surfaces or sharply contrasted packaging.

A-5. Safety

The safety and wellbeing of the individual soldier is one of the primary driving forces of the soldier modernization program. Providing a power source that is inherently dangerous to the soldier cannot be justified. The soldier will need to be protected from any detrimental effects of the power source, such as high temperatures, toxic exhausts, dangerous chemicals/fuels, electric shock, or fragmentation/explosion of the system or its components.

A-6. Vibration/Gyroscopic Forces

The power source may produce vibration or gyroscopic forces that affect the soldier. MIL-STD-1472 does not specifically cover the vibrations caused by such equipment, but it does recognize that vibrations may impair human performance and could decrease effectiveness. Gyroscopic forces should be limited so that the soldier retains full freedom of movement without exerting additional effort.

A-7. Attitude

The soldier may have to run, dodge, jump, crawl, and do other motions that will drastically change the attitude of the power source and/or its fuel supply. The adverse effects of changes in attitude must be considered in designing the power system.

A-8. Shelf Life

The units must be storable in a nonoperating mode when not required. Any special equipment or facilities needed for storage should also be considered.

A-9. Integrated Logistic Support (ILS)

The current trend to reduce the number of personnel in the armed forces will require each soldier to have greater capabilities and be more effective. Since support personnel make up most of the force, the power system will have to require minimum support. In other words, it would be foolish to provide a tenfold increase in the capabilities of the soldier, if it meant an associated tenfold increase in required support personnel. Consideration must be given to the requirements for training, spare parts, manuals, special tools, and the other elements of ILS. A key factor in the ILS area will be the fuel used; the use of any special fuel should be coordinated within the logistic supply community as early as possible, to avoid proceeding with a technical solution that will not be logistically supportable.

A-10. Reliability/Availability/Maintainability

The power source must be reliable, easily maintained, and available when required. These factors are interrelated and are also closely tied to the ILS aspects of the system. For high reliability, the system should be simple, rugged, and capable of operation in all environmental conditions. Simplicity of design and operation will also improve maintainability. It would be desirable to incorporate maintenance procedures in the soldier's personal computer, but since the unit will not be operable during most maintenance, this may be impractical. On extended missions, parts such as filters may need to be cleaned in place, or spares may need to be carried to support missions between resupply periods. There are no current required values for the necessary reliability and availability. However, similar systems in command and control technology have extremely high values for operational availability that are generally met through redundancy. The aspects of environment discussed earlier will have a large impact on this factor.

A-11. Size

The size of the unit, which is closely tied to its weight, risks inhibiting the soldier during the performance of his tasks or increasing his target size. Another consideration is the carrying requirement. The system should be designed to provide maximum ease of handling and should provide a package that distributes the weight so that the center of gravity is near the spinal axis.

A-12. Starting/Restarting

The method and time required to start and restart the system is an important consideration. The method should be simple and quick. The user will need to determine if startup time is critical, since some systems will take more time to be brought up to the operating temperature needed to support the full load. The effect of numerous start/stop cycles should also be considered.

A-13. Efficiency

Efficiency is closely tied to weight, since a less efficient system will require more fuel (weight) for a given mission. This relationship is not necessarily linear, since the type of fuel used may be different. A high-efficiency system could require more pounds of fuel if the fuel is low in energy density.

Appendix B. Excerpts from Literature on Earlier Technology

Contents

B-1. Uniflow Steam Engine

The following extract is reproduced from W. Barnard, F. Ellenwood, and C. Hirshfeld, *Heat Power Engineering*, John Wiley & Sons, third edition (1926), part I, pp 388--390.

2.3.2. Uniflow Engine. (a) This type of engine was invented[12] in 1885 by T. J. Todd of England but remained undeveloped until 1908 when Professor Stumpf, of Charlottenburg University, succeeded in making it a highly perfected prime mover.[13] Since that time others have modified it to suit special conditions so that it is now widely used for a variety of purposes.

(b) The *unusual features* of the engine may be noticed by referring to Fig. 209. The piston is of the box type having a length equal to about 90 per cent of the stroke; and when near the end of its movement it uncovers the exhaust ports located around the middle of the cylinder, thus taking the place of the exhaust valve. By having the exhaust all pass out through these central ports, the admission ends of the cylinder and the clearance surfaces are not cooled by the outflowing steam, as they are in the usual "counter-flow" type of engine. Furthermore, by reducing the time during which exhaust occurs, the amount of cylinder condensation is still further decreased. The exhaust period may be kept very short without causing any appreciable increase in the back pressure because the area of the exhaust ports is relatively large. Another means of preventing (or reducing) cylinder condensation on these engines is the use of the steam jackets on the heads and part of the walls of these cylinders. Also the clearance is kept extremely small, the compression ratio is very large, and highly superheated steam is commonly used. All these factors combine to minimize the cylinder condensation or to eliminate it altogether, even though the engine be run condensing with a high ratio of expansion.

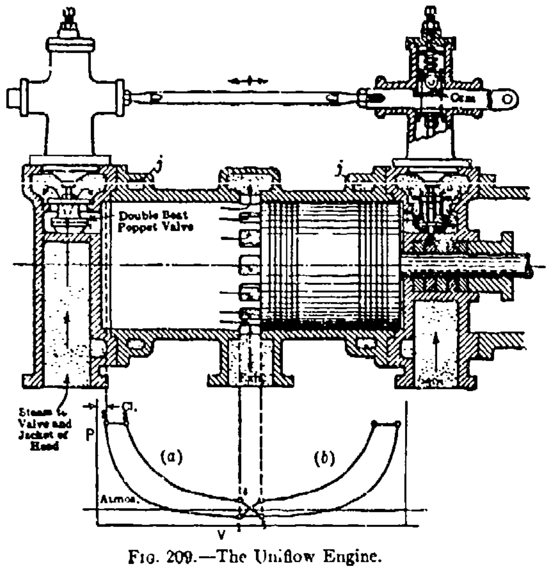


Fig. 209.—The Uniflow Engine.

[12]See "The Uniflow Engine," by F. B. Perry, Proceedings of Inst. Mechanical Engineers (British), July, 1920, p. 731

[13]See Professor Stumpf's book, "The Uniflow Steam Engine."

(c) The admission valves shown in Fig. 209 are the double-beat poppet type, but other kinds are also often adopted. The poppet type allows highly superheated steam to be used and the double beat feature permits a smaller lift to be employed for a given area of opening. The piston functions as the exhaust valve, as already noted.

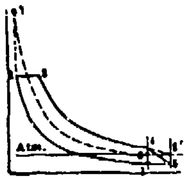


Fig. 210.

(d) The *indicator diagrams* from each end of the cylinder of a condensing engine will be similar to those shown by (*a*) and (*b*) in Fig. 209. The compression must necessarily begin very early because it is controlled by the closing of the exhaust ports by the piston.

If the engine is run non-condensing with the large compression ratio that is used with the condensing type, the compression pressure is likely to go much higher than the admission pressure, as shown by the dotted line 6-7 in Fig. 210. This would be very undesirable, and consequently the uniflow type of engine is built in a variety of ways to take care of this feature in case it becomes necessary to run the engine non-condensing. Thus, extra clearance is often introduced by means of a special valve that opens automatically when the compression pressure exceeds the admission pressure. Sometimes some of the compression steam is by-passed to the other side of the piston, and in certain engines small auxiliary exhaust valves are employed.

(e) The *sizes* of uniflow engines that are in use vary widely. There are large numbers of them under 500 horsepower, and many above 1000. Recently a 30,000 horsepower uniflow was built.

(f) The chief *advantages* of the uniflow engine are:

(1) The *thermal efficiency* is high because the cylinder condensation is small, and because the mechanical efficiency is large even though a high ratio of expansion is used.

(2) The *indicated steam rate* at three-quarter load is only slightly more than at full load; and at 25 per cent overload, and also at half load, it is only about 5 per cent greater.

(3) The engine itself, its floor space and foundation are much *less costly* than for the compound engine of the same economy.

(4) It is *simpler* to operate and less expensive to maintain than is the compound of the same economy.

B-2. Williams Steam Engine

The following extract is reproduced with permission from C. Wise, *Steam is Back*, Machine Design Magazine (29 August 1968), p 22.

STEAM IS BACK

Williams: Exceeding Theoretical Limits?

Using methods which are by no means obvious, the Williams Engine Co. has achieved extraordinarily low water-rates (and high efficiencies) in small, reciprocating, single-expansion steam engines. The thermodynamic explanation given by the inventors is almost certainly incorrect, and no one seems to have offered a satisfactory explanation.

In the Williams engines about 2/3 of the steam remains in the cylinder after the exhaust phase; on the return stroke it is compressed and reheated. Since the compression ratio (26.6:1) is greater than the expansion ratio (18.6:1) the ultimate temperature of 1,492 F is higher than the feed steam temperature of 1,000 F. Work done on the recompressed steam is recovered when the hotter compressed steam raises the temperature of the feed steam, resulting in more efficient utilization of its heat content.

It can be shown that in an ideal Rankine engine that this 'split' cycle neither increases nor decreases the overall enthalpy efficiency of the energy-conversion process. But in practice it appears that the Williams engines do, in fact, perform better than the ideal Rankine cycle.

It has been determined that for each pound of compressed steam in the cylinder, 0.56 lb of feed steam is added before (10%) cutoff. For these conditions, the theoretical Rankine thermal efficiency should be 25.8%. However, a shop test on a 56-cu in., 4-cyl Williams engine showed the test result given in the table on Page 26. Although this remarkable thermodynamic performance cannot easily be explained, it seems to be an empirical fact worthy of attention.

The present-day Williams engines—which would be similar in gross detail to other modern reciprocating engines—are single-acting, uniflow models with four vertical cylinders arranged in-line over a crankshaft. Inlet and exhaust valves are operated by connecting rods controlled by cams on the crankshaft.

There are several power ranges, corresponding to several degrees of cutoff, which the operator can choose at will (like changing gears). For startup, a 20-25% cutoff is needed, while for sustained high-speed driving a 10% cutoff is sufficient. Maximum torque is obtained with a 70% cutoff.

The engine can be reversed by changing the phasing of the cylinders, which is accomplished by engaging an appropriate cam. This permits dynamic braking. The engine operates on an open cycle, but with a relatively small condenser (35 lb for a 300-lb engine); water consumption is only about 1 gal per hr at variable speed operation, or 10 gal for 500 mi. The monotube steam generator, which burns diesel fuel, No. 1 fuel oil, or kerosene, weighs approximately 250 lb. It can produce steam enough to move the car in 20 sec and develops a full head of steam in less than 1 min.

Apart from its remarkable efficiency, the Williams engine incorporates several engineering innovations. One is a valve, built into the engine, which automatically matches engine compression to feed-steam pressure. Without such a mechanism a uniflow steam engine cannot operate smoothly under rapidly variable load conditions—the engine tends to "buck" or stall. This problem particularly vexed Doble, who reportedly attributed his company's failure to not having solved it.

Williams has offered to sell complete steam-powered cars, fitted with 105-cu in. engines on a Chevelle chassis, for roughly $7,000 for the engine or $10,000 for the complete vehicle

TEST RESULTS ON WILLIAMS ENGINE

Steam pressure, gauge (psi)	1000
Load on Clyton Dynamometer (lb)	66.25
Engine speed (rpm)	2500
Feed system temperature, measured by calibrated thermocouples with Leeds & Northrup porentiometer (F)	981.2
Exhaust pressure (psi)	14.7
Exhaust temperature (F)	300
Weight of exhaust steam collected in 2-pass Ross condenser (lb/hr)	203
Power, indicated (hp)	36.23
Power, dynamometer (bhp)	31.5
Water rate, actual (lb/bhp/hr)	6.44
Thermal efficiency, calculated (%)	38.8

Data compiled by Dr. Robert U. Ayres.

B-3. Bolz's Analysis of Liquid Thermal Engine Performance

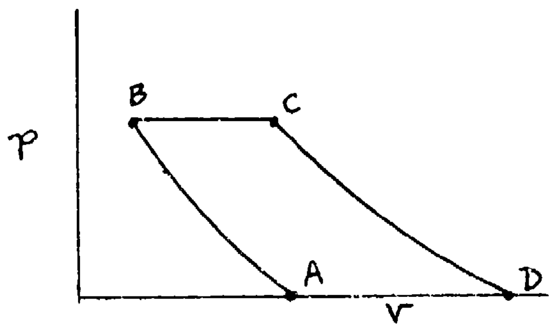
The following extract is reproduced from a paper (*Liquid Thermal Engine*, Internal Report No. 1258, 11 May 1960), obtained from the Cleveland Pneumatic Tool Company of Cleveland, OH, in 1960. The analysis makes reference to data on acetone obtained from P. W. Bridgeman.[1]

[1] *P. W. Bridgeman.* ***Thermodynamic Properties of Twelve Liquids Between 20° & 80°, and up to 12,000 KGM/sq-cm.*** *Proc. Amer. Acad.* ***49*** *(1913), 1–114.*

SUPPLEMENT TO LIQUID THERMAL ENGINE PERFORMANCE ANALYSIS

Analysis of Constant Pressure Cycle

The operation of this engine devised by Cleveland Pneumatic is described in a separate enclosure along with the preliminary mechanical design of a practicable engine. The engine follows the Brayton cycle given in Figure 1.



In this cycle liquid is adiabatically compressed to a predetermined pressure p_B and by means of a check valve delivered to a heat exchanger system. Hot fluid is simultaneously delivered to an expansion cylinder at pressure $p_C = p_B$ and, after the delivery valve closes, is expanded adiabatically back to the original atmospheric pressure $p_D = p_A$.

Analysis of this cycle can be easily performed using one of the equations of the original report which this discussion supplements.

Accordingly:

$$\text{Net Work} = -\int_C^D p\,dV + p_B(V_C - V_B) - \int_A^B p\,dV$$

Or from equation (6) page (7) of the reference report

$$W_{NET} = -\int_{C-B}^{D} kV \left(\frac{C_v}{C_p}\right) p\,dp - \int_A^B kV \left(\frac{C_v}{C_p}\right) p\,dp + p_B (V_C - V_B)$$

$$= k_{C-D} V_{C-D} \left(\frac{C_v}{C_p}\right) \left(\frac{p_B^2 - p_D^2}{2}\right) - k_{A-B} V_{A-B} \left(\frac{C_v}{C_p}\right) \left(\frac{p_B^2 - p_A^2}{2}\right)$$

$$+ p_B (V_C - V_B) \qquad (1)$$

The heat added is

$$Q_{IN} = C_p \, \Delta T_{B-C} \qquad (2)$$

and the efficiency is, of course, the ratio of Eq. (1) to Eq. (2)

$$\eta = \frac{\text{Eq. 1}}{\text{Eq. 2}} = \frac{\text{Net Work}}{\text{Heat Input}} \qquad (3)$$

I) Numerical Calculations of Cycle Using Acetone - Bridgeman's Data

Conditions of Problem

1) Take $p_A = 0$
2) Take $p_B = 3000$ kg/cm² $\cong$ 43,000 psi
3) Take $T_A = 0°C$
4) Assume C_p/C_v invariant
5) Take $V_A = 1$ cm³
6) Take $T_C - T_B = 125°C$
7) The unit of mass upon which Bridgeman's data are based is that mass which occupies 1 cm³ volume at $p = p_A$, $T = 0°C$

Data

1) C_p at $p = p_{ATM} = 19.5$ kg-cm/°k for the quantity of mass in (7) above
2) Take C_p at $p = 3000$ kg/cm² = 16 kg-cm/°k
3) $k_{A-B} = 4.2 \times 10^{-6}$
4) $k_{C-D} = 7 \times 10^{-6}$ (extrapolated conservative value for high temp)
5) $V_B = .894$
6) $V_{A-B} \approx .947 = .95$
7) $V_C = V_B + \beta_{B-C}\,\Delta T = V_B + 5.5 \times 10^{-4} \times 125 = .963$
8) $V_D = V_A + \beta_{A-D}\,\Delta T = V_A + 1.6 \times 10^{-3}\,(90) = 1.14$
9) $V_{C-D} = 1.05$
10) $C_v/C_p = .86$

Then for these operating conditions

$$W_{NET} = (7 \times 1.05 - 4.2 \times .95)\ .86 \times 10^{-6} \times \frac{3000^2}{2} + 3000(.963 - .894)$$

$$= (63.4 - 34.4)\ 4.5 + 207 = 130 + 207 = 337\ \frac{\text{kgcm}}{\text{cc}}$$

For an engine operating at 1000 rpm (two-stroke cycle) we get:

$$HP = \frac{337 \times 2.22 \times 2.54^3}{12} \times \frac{1000}{33000} = \boxed{12 \text{ HP per in}^3} \text{ Piston Displacement}$$

$$\text{Efficiency} = \frac{337}{16(125)} = \boxed{17\%}$$

II) As a Second Example Assume $\Delta T = 200^{\circ}C$

Conditions of Problem

All other conditions are the same as in (I).

Data

1) C_p during heat input stroke take as 16 kg-cm/°k
2) $k_{A-B} = 4.2 \times 10^{-6}$
3) $k_{C-D} = 8 \times 10^{-6}$ (conservative extrapolation)
4) $V_B = .894$
5) $V_{A-B} = .95$
6) $V_C = V_B + 5.5 \times 10^{-4} \times 200 = 1.00$
7) $V_D = V_A + 1.6 \times 10^{-3} \times 150 = 1.24$
8) $V_{C-D} = 1.12$
9) $C_v/C_p = .86$

$$\text{Net Work} = (8 \times 1.12 - 4.2 \times .95)\ .86 \times 10^{-6} \times \frac{3000^2}{2} + 3000(1.00 - .894)$$

$$= 193 + 318 = 511 \text{ kg-cm}$$

$$= \boxed{18.5 \text{ HP}} \text{ at 1000 rpm engine speed}$$

$$\text{Efficiency} = \frac{511}{16 \times 200} = \boxed{16\%} \text{ thermal efficiency}$$

These efficiences are without regeneration. It is felt that with good regeneration this efficiency may be increased by 50% and theoretical values of 25% obtained. Again, I feel that frictional losses in the cylinder and heat exchangers will not be large and with good valving a reasonable portion of the theoretical efficiency may be obtained--perhaps 15% to 17% for the cycle efficiency.

No part of the engine, as designed to date by Cleveland Pneumatic, appears to present any insurmountable engineering problems. Ingenuity and experimental experience are necessary to minimize engine and heat transfer sizes and weights and the ability to cope with the high pressures needed for good efficiency is, I believe, just a matter of experience.

K. E. Bolz

Appendix C. Calculations to Determine Preliminary Size of Vapor-Cycle Engine and Power System Components

Contents

C-1. Engine Size

The nominal output power requirement is 300 W, and 25-percent excess capacity is needed for 1-hour periods. Thus,

$$\text{peak capacity} = 300 \times 1.25 = 375 \text{ W} \quad .$$

Assume an efficiency of 70 percent for the high-speed motor-generator and refrigeration compressor. Then,

$$\text{net power} = 375/0.70 = 536 \text{ W} \quad .$$

Assume that the engine operates at a speed of 9000 rpm. Average net torque equals net power/speed, so we calculate

$$\frac{536 \times 12 \times 33{,}000 \text{ (ft-lb/hp)}}{746 \text{ (W/hp)} \times 2\pi \times 9000} = 5.0 \text{ in.-lb} \quad .$$

Assume that the accessory power required for the engine pumps, fans, and controls is 100 W. Then,

$$\text{engine brake power} = 536 + 100 = 636 \text{ W} \quad ,$$

and

$$\text{engine brake torque} = 5.0 \times 636/536 = 6.0 \text{ in.-lb} \quad .$$

Assume that the engine mechanical efficiency is 85 percent. Then

$$\text{engine indicated power} = 636/0.85 = 748 \text{ W} = 1.0 \text{ hp}$$

$$= 42.5 \text{ btu/min} \quad ;$$

also,

$$\text{horsepower} = \frac{\text{IMEP} \times \text{volume} \times \text{rpm}}{12 \times 33{,}000}$$

where IMEP is indicated mean effective pressure.

Rearranging, the swept volume or engine displacement is

$$\frac{\text{hp} \times 12 \times 33{,}000}{\text{rpm} \times \text{IMEP}} = \frac{1 \times 12 \times 33{,}000}{9000 \times \text{IMEP}} = \frac{44}{\text{IMEP}} \text{ in.}^3$$

Substituting typical values of IMEP produces the following:

IMEP (psig)	Volume (in.^{3})
44	1.00
50	0.88
88	0.50
100	0.44
150	0.30

In an actual engine, the power can be varied from a nominal value by changing the IMEP using variable inlet valve timing.

Assume the nominal IMEP equal to 100 psi to give an engine displacement of 0.44 in.^{3} Also, assume a two-cylinder engine configuration of horizontally opposed pistons in a common cylinder in order to obtain good balance for smooth operation and better control of the high compression ratio ($\approx 30:1$).

This results in a displacement of $0.5 \times 0.44 = 0.22$ in.^{3} per cylinder.

Assuming a piston stroke, s, of 0.50 in., the cylinder bore is

$$(4V/\pi s)^{0.5} = (4 \times 0.22/\pi \times 0.50)^{0.5} = 0.75 \text{ in.} = 0.44 \text{ in.}^2$$

where V = volume.

C-2. Steam Generator Size

From the preliminary engine size of 1 ihp or 42.5 btu/min, and an assumed engine thermal efficiency of 26.5 percent, the input heat rate to the engine will be 42.5/0.265 = 160 btu/min.

Assume a full-load boiler efficiency of 87 percent. Then,

$$\text{net boiler heat input} = 160/0.87 = 184 \text{ btu/min} \ .$$

Assume an excess capacity for contingency and growth potential of 50 percent. Then,

$$\text{gross boiler heat input} = 184 \times 1.5 = 276 \text{ btu/min} = 16{,}600 \text{ btu/hr} \ .$$

The Bessler reference[1] gives demonstrated heat rates for modern forced-circulation flash boilers of 1.25, 2.0, and 3.0 million btu/hr/ft^3. Assume a boiler heat-rate capability of 2,000,000 btu/hr/ft^3 for this design. Then, gross boiler size is

$$16{,}600/2{,}000{,}000 = 0.0083 \text{ ft}^3 = 14 \text{ in.}^3$$

[1] *W. J. Bessler and J. L. Boyen, Design Study of a Steam Power System for a Landing Craft, under contract No. 2159(00) for the Office of Naval Research, Amphibious Branch (30 September 1957).*

Since this is a small-scale design, assume a size of double this volume, or a net boiler size of 28 in.^{3} This is equivalent to a shape of 3 in. in diameter and 4 in. long.

From Keenan and Keyes,[2] steam at 1200 psi and 1000°F has an enthalpy of 1500 btu/lb. Thus, the peak steam rate is

$$16{,}600 \times 0.87/1500 = 9.6 \text{ lb/hr} \ .$$

This allows enough steam for an engine efficiency as low as

$$\frac{42.5 \text{ btu/min/ihp} \times 60 \text{ min/hr} \times 1 \text{ ihp}}{1500 \text{ btu/lb} \times 9.6 \text{ lb/hr}} = 17.7\% \ .$$

C-3. Fuel Tank Size

Assume that diesel fuel has a heating value of 19,000 btu/lb and a specific volume of 30 in.^{3}/lb. This results in a heating value density of ≈633 btu/in.^{3}

At a peak boiler heat rate of 16,600 btu/hr, the peak fuel rate will be

$$16{,}600/633 = 26 \text{ in.}^3/\text{hr} \ .$$

The nominal fuel rate is

$$\frac{166 \text{ avg W/day}}{375 \text{ peak W/day}} \times \frac{16{,}600 \text{ btu/hr peak rate}}{1.5 \text{ contingency factor}} \times \frac{1}{633 \text{ btu/in.}^3}$$

$$= 7.74 \text{ in.}^3/\text{hr} = 186 \text{ in.}^3/\text{day} \ .$$

This results in preliminary tank sizes for one day of 6 × 12 × 2.6 or 6 × 8 × 3.9 in. At 231 in.^{3}/gallon, these example tanks would hold about 0.8 gallon.

[2] *J. H. Keenan and F. G. Keyes, Thermodynamic Properties of Steam, John Wiley & Sons (1936).*

Appendix D. Analysis of Vapor-Cycle and Steam Engine Performance

Contents

Figures

D-1. Introduction

This initial analysis* is an attempt to estimate the thermal efficiency of the vapor-cycle power system, the amount of steam required, and the appropriate cutoff or expansion ratio needed for the specified engine design. (Cutoff is the end of that portion of the piston stroke during which admission of the feed steam occurs.) The analysis is based on "indicated" conditions: that is, values measured within the engine cylinder. This gives theoretical gross power and efficiency. Net or "brake" power and efficiency are treated in section 5.8 of the main report.

The vapor-cycle system includes the steam generator (consisting of a boiler and a superheater), an expander or engine, a condenser, and a pump, as shown in figure D-1. A temperature/entropy (*T/S*) diagram for the ideal simple steady-flow Rankine thermodynamic vapor cycle is also shown.

The processes that make up the thermodynamic cycle are

1–2: Reversible adiabatic pumping process in the liquid pump

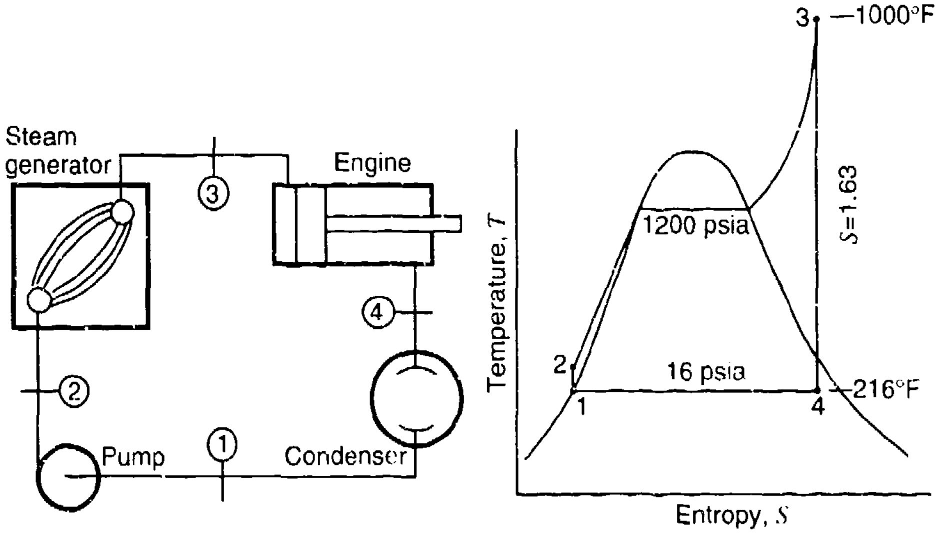
2–3: Constant-pressure transfer of heat in the boiler and superheater

3–4: Reversible adiabatic expansion of the vapor in the prime mover

4–1: Constant-pressure transfer of heat in the condenser

The usual type of expander is a steady-flow turbine-generator system. In this case, the expander is an intermittent-flow reciprocating engine operating at ≈150 Hz to expand relatively small quantities of steam (≈7 lb/hr). Under

Figure D-1. Simple steam power plant and Rankine vapor cycle.



*Recommendations for improving the analysis are made in section D-9.

steady-state conditions, the engine behaves in a fashion similar to the turbine, as far as selection and analysis of a Rankine thermodynamic vapor cycle is concerned.

The type of steam engine to be analyzed is a high-compression, uniflow, single-expansion, essentially noncondensing (see sect. D-4) system operating on high-pressure, super-heated vapor. The engine also operates in a cyclic fashion, as illustrated in figure D-2. Here the ideal pressure/volume (*P/V*) diagram is similar to that for the ideal air-standard diesel engine thermodynamic cycle, except that it is *not* a true thermodynamic vapor cycle, as shown by the vertical line on the *T/S* diagram.

Ideally the engine cycle consists of four stages:

1–2: a reversible adiabatic compression process, where the residual exhaust steam left in the cylinder is compressed to high pressure and temperature, as determined by the exhaust steam properties and the volumetric compression ratio of the engine

2–3: a constant-pressure admission of steam from the boiler* through valves in the cylinder head

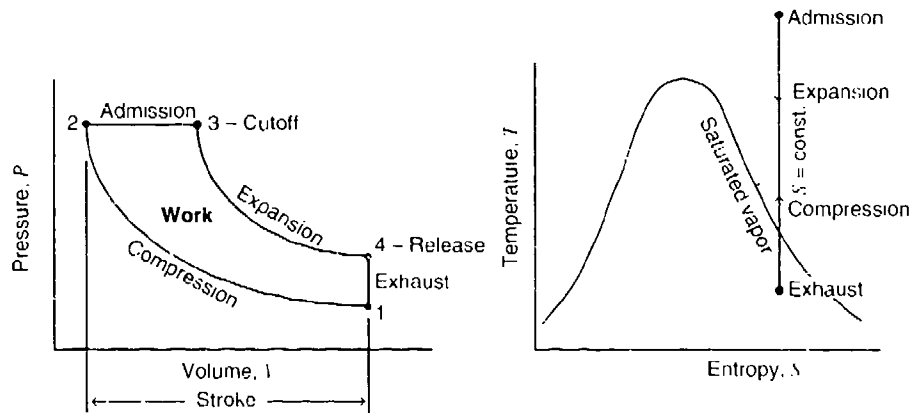
3–4: a reversible adiabatic expansion of the vapor in the cylinder after cutoff of the admission process and until release of the steam to the exhaust process

4–1: a constant-volume exhaust process, where expanded steam escapes into the condenser through ports uncovered at the bottom of the cylinder

D-2. Baseline Conditions

Before detailed analysis can begin, several parameters and elements of the engine operating cycle need to be established. These are the admission process, the exhaust process, the exhaust pressure, the feed-steam pressure

Figure D-2. Single-expansion uniflow steam engine cycle.



*The term "boiler" is used in place of the more correct term "steam-generator" in this text.

and temperature, and the engine compression ratio. First, assume that steady-state conditions prevail, and that compression and expansion occur along a common entropy line, as illustrated in figure D-2 (on the right) and the normal simple Rankine cycle for turbine-type expanders (fig. D-1). Thus, properties of the residual exhaust steam in the cylinder, as the exhaust port closes, determine the entropy value for the steady-state engine cycle. Conversely, the amount of steam admitted to mix with the compressed exhaust steam determines the properties of the residual steam at the end of the exhaust process.

D-2.1 Admission Process

Assume for this analysis that the engine compression ratio is such that the residual exhaust steam is compressed along this entropy line to a pressure condition that is lower than the feed-steam pressure from the boiler.* The admission process, where the incoming steam mixes with the compressed exhaust steam, can be handled in different ways. The method selected for this analysis is illustrated in figure D-3. It assumes that there is no throttling of the incoming steam to a lower pressure (this is treated in sect. D-7.2), and that there is no pressure drop during admission. Thus, the admission process occurs at the feed-steam pressure condition of 1200 psia.

As shown in the Mollier (h/S) diagram of figure D-3, mixing during the admission process increases the enthalpy (h) and temperature of the compressed exhaust steam, and decreases the enthalpy and temperature of the feed steam until the thermodynamic balance defining the end of admission and beginning of expansion, point 3 or cutoff, is reached. Larger cutoffs (longer admission period) allow more boiler steam to enter, thereby changing the heat balance and determining the final entropy for the expansion process. Thus, a different entropy line for operation is defined for each new value of cutoff. Each new value of cutoff also determines a new value of mean effective pressure or horsepower for the engine as shown in section D-6.

D-2.2 Exhaust Process

The exhaust process can also be handled in different ways. The method selected for this analysis was adapted from a diesel engine exhaust process [1] and is illustrated in the h/S diagram of figure D-4. As shown, the steam that escapes through the exhaust ports at release is assumed to be throttled [2] at constant enthalpy to the selected exhaust back-pressure condition. It then flows to the condenser. The steam that remains within the cylinder is assumed to expand isentropically to the exhaust back-pressure condition. It is then compressed, without further condensation (see sect. D-4), from this condition.

**Compression to pressures greater than or equal to boiler pressure is discussed in sections D-2.5, D-8, and D-9.*

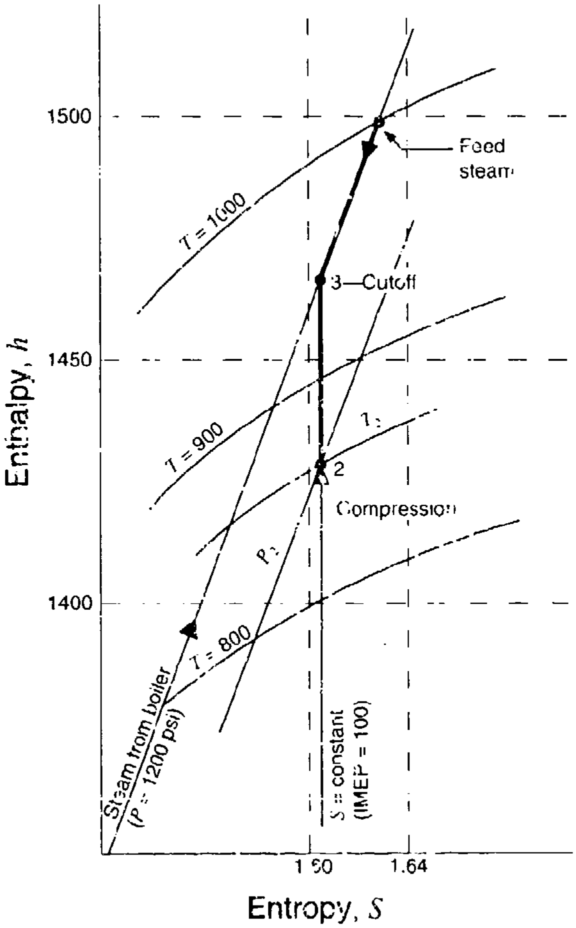


Figure D-3. Admission process for steam engine analysis.

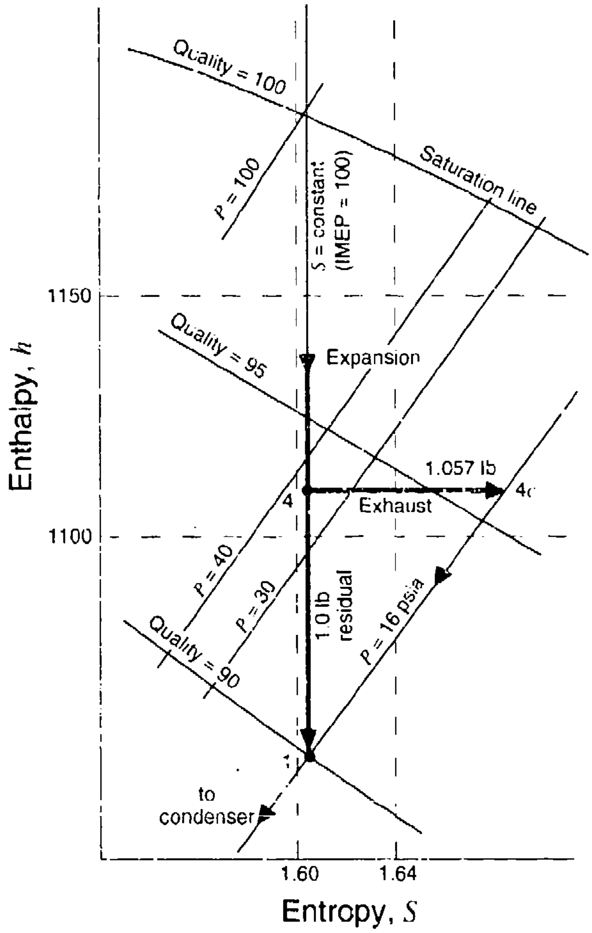


Figure D-4. Exhaust process for steam engine analysis.

D-2.3 Exhaust Pressure

Assume that the engine will operate at essentially "atmospheric" (nonvacuum) exhaust pressure conditions. Assume that there will be a slight back-pressure in the condenser of 1.3 psi and that normal atmospheric pressure is 14.7 psia. Thus, the nominal exhaust pressure for this analysis is $P_1 = 14.7 + 1.3 = 16.0$ psia.

D-2.4 Feed-Steam Conditions

Assume the boiler or feed-steam pressure to be 1200 psi and the temperature to be 1000°F for this system. The desire is to have values that are as high as practical in order to increase thermal efficiency [2]. Considerations of material strength, weight, and lubrication and sealing difficulties are involved in the selection of these conditions. The Williams engine [3] discussed in the body of the report operated on 1000 psia and 1000°F steam conditions.

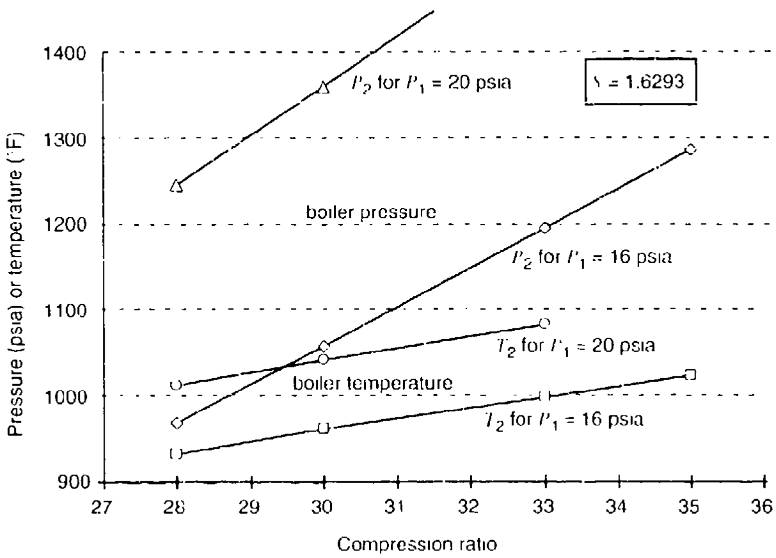
D-2.5 *Engine Compression Ratio*

The engine volumetric compression ratio is a critical design element, as shown in this analysis. The Williams engine used a 26 : 1 ratio. The desire is to always compress the residual steam back to boiler pressure because this maximizes thermal efficiency. However, boiler pressure can easily be exceeded, depending on the properties of the exhaust steam. Very high compression pressures make engine operation difficult, so special relief valves in the cylinder head are used to automatically bleed off compressed steam when the admission pressure is exceeded.

In order to simplify this preliminary analysis, I selected a compression ratio that is low enough to prevent the steam from bleeding off during compression for most anticipated exhaust conditions. In order to select an appropriate compression ratio, I evaluated several cases for adiabatic compression of steam. Since the entropy for the desired operating condition in the engine is unknown, assume that $S = 1.6239$, which corresponds to feed-steam properties of 1200 psia and 1000°F. Initial condenser pressures assumed are the nominal $P_1 = 16$ psia and a higher value, $P_1 = 20$ psia, that could represent the increased back-pressure that might result from the higher volumes of steam needed at higher power levels.

Results for volumetric compression ratios of 28, 30, 33, and 35 to 1 were calculated as shown in figure D-5. Note that compression from an initial pressure of 20 psia causes the boiler pressure and temperature conditions to be exceeded for virtually all the compression ratios evaluated. Based on these results, a compression ratio of 28 : 1 is the nominal value selected for the analysis. A ratio of 33 : 1 is also evaluated as a special case, because it produces essentially boiler conditions when the exhaust pressure is 16 psia.

Figure D-5. Results of steam compression analysis



D-3. Analysis Process

The engine system analysis process is defined as follows.

(a) Assume a value for the operating entropy line, S, that will give the desired indicated mean effective pressure (IMEP) for the engine.

(b) With this entropy and the assumed exhaust pressure, calculate (by means of interpolation from the steam tables [4]) the other steam properties (v, h, and T) at point 1, the beginning of compression. (Two of the properties are independent, and the other three are dependent at each point in the cycle.)

(c) Assume that the engine cylinder is sized to hold exactly 1 lb of steam at this specific volume, v_1, so that $V_1 = v_1$. In this analysis, capital H and V represent actual values, and lower-case h and v represent specific (relative) values.

(d) Use the compression ratio $R_c = 28:1$ to calculate $v_2 = V_2 = V_1/R_c$.

(e) Use v_2 and S to obtain the other steam properties at point 2, end of compression.

(f) Use the steam properties at points 1 and 2 and the cylinder volumes at V_1 and V_2 to determine the indicated mean effective compression pressure (IMCP) from the change in internal energy of the compressed steam, as illustrated in figure D-6.

(g) Assume that the admission valve opens at exactly top-dead-center of the piston stroke and that the compressed steam is instantly pressurized to 1200 psia in the clearance volume.

(h) Assume that the steam properties at the end of admission (cutoff) are $S_3 = S$ and $P_3 = 1200$ psia, and determine the enthalpy and specific volume of the mixture of compressed and admitted steam at point 3, cutoff.

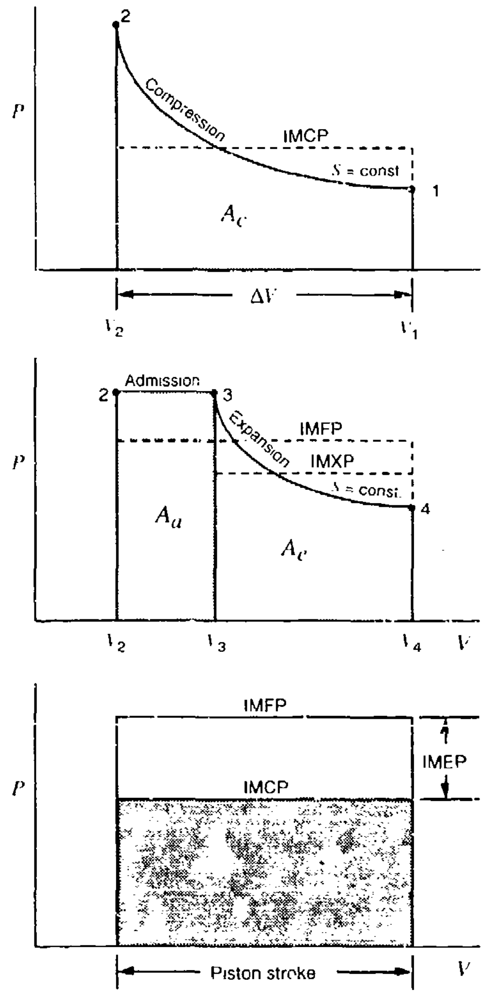
(i) Assume that the admission process is adiabatic and conduct a heat balance using the enthalpies at points 2 and 3 and the weight of the compressed steam at point 2 to determine the weight of the steam admitted.

(j) Use the total weight of steam and its specific volume at point 3 to determine the cylinder volume, V_3, at cutoff.

(k) Use V_3 and $V_4 = V_1$ to calculate the expansion ratio, R_x, and the specific volume, $v_4 = v_3 R_x$ at point 4, release.

(l) Use v_4 and $S_4 = S$ to determine the steam pressure and enthalpy (P_4 and h_4) at release.

Figure D-6. Process for calculation of mean effective pressure for steam engine.



(m) Use the steam properties at points 3 and 4 and the cylinder volumes V_3 and V_4 to determine the indicated mean effective expansion pressure (IMXP) from the change in internal energy of the expanded steam, as illustrated in figure D-6.

(n) Compute the indicated mean effective forward pressure (IMFP) using the IMXP and the area $A_a = P_3(V_3 - V_2)$, as shown in figure D-6.

(o) Compute the IMEP from IMEP = IMFP − IMCP and compare it with the desired value from step (a).

(p) Adjust the value assumed for the entropy line, S, and repeat the above process until the desired value of IMEP is obtained.

D-4. Engine Cycle Thermodynamics

An example engine cycle analysis for the conditions necessary to give IMEP = 100 psi, the nominal case of interest, is as follows. First, a value for the entropy line is selected as $S = 1.6047$ (previously determined by trial and error). With $P_1 = 16$ psia, the other steam properties at point 1 are found to be

$$P_1 = 16.0 \text{ psia},$$
$$T_1 = 216.3°\text{F},$$
$$v_1 = 22.2447 \text{ ft}^3/\text{lb},$$
$$h_1 = 1054.0 \text{ btu/lb},$$
$$S_1 = 1.6047.$$

Note that this steam is not superheated. Its quality is 89.9 percent (10.1-percent condensed liquid) in theory. In actual practice the steam is probably in a supersaturated condition wherein condensation has not had a chance to take place because of the high-speed engine operation (≈150 Hz) and the slight amount of expansion into the condensation zone.

D-4.1 Compression Analysis

To obtain the conditions at point 2, assume an adiabatic (no heat transfer) and isentropic (constant entropy) compression process. An insulating jacket on the engine and operation at such a high speed makes this a reasonable assumption. Also, assume that the cylinder is sized to hold exactly 1 lb of residual exhaust steam so that $V_1 = 22.2447$ ft^3. Applying the specified compression ratio, find $v_2 = v_1/R_c = 22.2447/28.0 = 0.7945$ ft^3/lb. The other steam properties at point 2 are found to be

$$P_2 = 926.3 \text{ psia},$$
$$T_2 = 859.7°\text{F},$$
$$v_2 = 0.7945 \text{ ft}^3/\text{lb},$$
$$h_2 = 1427.7 \text{ btu/lb},$$
$$S_2 = 1.6047.$$

Note that the desired pressure and temperature conditions of 1200 psia and 1000°F were not reached with this low compression ratio.

The area under the compression curve for this adiabatic and thermodynamically closed system represents the change in internal energy, U, of the steam caused by the compression process [5]. Internal energy is computed from the enthalpy (h) values as follows.

Since

$$h = u + pv \quad ,$$

then

$$u = h - pv \quad ,$$

and

$$A_c = \int p \, dv = \Delta U \quad .$$

So

$$\text{IMCP} = \Delta U/\Delta V = (U_2 - U_1)/(V_1 - V_2) \quad .$$

Since exactly 1 lb of residual exhaust steam is being compressed, $W_1 = W_2 = 1$, $V_1 = v_1$, $U_1 = u_1$, $H_1 = h_1$, and $V_2 = v_2$, etc. So,

$$U_1 = 1 \times (1054.0 - 16 \times 22.2447/5.404) = 988.14 \text{ btu} \quad ,$$

$$U_2 = 1 \times (1427.7 - 926.3 \times 0.7945/5.404) = 1291.51 \text{ btu} \quad ,$$

and

$$\text{IMCP} = \frac{1 \times (1291.51 - 988.14) \times 5.404}{1 \times (22.2447 - 0.7945)} = 76.43 \text{ psi} \quad ,$$

where 778.165 ft-lb/btu [6] and 144 in.^{2}/ft^2 are used to make the appropriate dimensional conversions, and 5.404 = 778.165/144.

D-4.2 Admission Analysis

To obtain the steam properties at cutoff (point 3), use the admission pressure P_3 = 1200 psia, and the constant entropy value S_3 = 1.6047 to interpolate for the other three values:

P_3 = 1200 psia,
T_3 = 940.6°F,
v_3 = 0.6491 ft^3/lb,
h_3 = 1464.44 btu/lb,
S_3 = 1.6047.

For an adiabatic admission process, the heat balance is

$$W_2 h_2 + W_f h_f = (W_2 + W_f) h_3 \quad ,$$

where W_2 is the weight of the compressed steam and W_f is the weight of the feed steam. Properties of the feed steam are

$P_f = 1200$ psia,
$T_f = 1000°F$,
$v_f = 0.6843$ ft^3/lb,
$h_f = 1499.2$ btu/lb,
$S_f = 1.6293$.

Solving for the relative weight of the feed steam,

$$\frac{W_f}{W_2} = \frac{h_3 - h_2}{h_f - h_3} = \frac{1464.44 - 1427.7}{1499.2 - 1464.44} = 1.057 \text{ lb/lb compressed steam} \quad .$$

The engine cylinder volume at cutoff is

$$V_3 = v_3(W_2 + W_f) = 0.6491(1 + 1.057) = 1.335 \text{ ft}^3 \quad ,$$

which is

$$\frac{1.335 - 0.7945}{22.2447 - 0.7945} = 2.52 \text{ percent of the piston stroke} \quad .$$

D-4.3 *Expansion Analysis*

The engine expansion ratio is

$$R_x = V_4/V_3 = V_1/V_3 = 22.2447/1.335 = 16.66 : 1 \quad .$$

The specific volume of the steam at release, point 4 in the cycle, is found by applying R_x:

$$v_4 = R_x v_3 = 16.66 \times 0.6491 = 10.8128 \text{ ft}^3\text{/lb} \quad .$$

Other properties of the steam at release are found from steam table interpolation to be

$P_4 = 36.2$ psia,
$T_4 = 261.3°F$,
$v_4 = 10.8128$ ft^3/lb,
$h_4 = 1110.4$ btu/lb,
$S_4 = 1.6047$.

The IMXP is found by the same process used for determining the IMCP as follows:

$$\text{IMXP} = A_e/\Delta V = \Delta U/\Delta V = (U_3 - U_4)/(V_4 - V_3) \quad .$$

So,

$$U_3 = W(1464.4 - 1200 \times 0.6491/5.404) = 1320.3W \text{ btu} \quad ,$$

$$U_4 = W(1110.4 - 36.2 \times 10.8128/5.404) = 1038.1W \text{ btu} \quad ,$$

and

$$\text{IMXP} = 2.057 \times \frac{(1320.3 - 1038.1)5.404}{22.2447 - 1.335} = 150.2 \text{ psi} \quad .$$

From figure D-6,

$$\begin{aligned} \text{IMFP} &= (A_a + A_c)/(V_4 - V_2) \quad , \\ &= [P_2(V_3 - V_2) + \text{IMXP}(V_4 - V_3)]/(V_4 - V_2) \quad , \\ &= \frac{1200(1.335 - 0.7945) + 150.2(22.2447 - 1.335)}{22.2447 - 0.7945} = 176.6 \text{ psi} \quad . \end{aligned}$$

Also from figure D-6, the IMEP is

$$\text{IMEP} = \text{IMFP} - \text{IMCP} = 176.6 - 76.4 = 100.2 \text{ psi} \quad ,$$

which is the desired value (since the proper value of S was preselected for this example).

D-4.4 Exhaust Analysis

The properties of the steam exhausting into the condenser at 16 psia are found by referring to figure D-4 to note that $h_{4C} = h_4$. Thus, from steam table interpolation

$P_{4C} = 16.0$ psia,
$T_{4C} = 216.32$°F,
$v_{4C} = 23.6863$ ft^3/lb,
$h_{4C} = 1110.4$ btu/lb,
$S_{4C} = 1.6881$.

D-5. Vapor-Cycle Thermodynamics

Now the thermal efficiency of the vapor-cycle power system can be determined. The equation for thermal efficiency is

$$E_C = \frac{\text{net work}}{\text{heat added}} = \frac{\text{engine work} - \text{pump work}}{H_f - H_{2VC}}$$

where H_{2VC} is the heat content of the compressed boiler feed water. Also,

$$h_{2VC} = H_L + \text{pump work} \quad ,$$

where H_L is the heat content of the condensed liquid at the condenser pressure of 16 psia. From the steam tables, h_l = 184.4 btu/lb.

D-5.1 *Pump Work*

The ideal pump work, assuming adiabatic pressurization of an incompressible liquid, is

$$\Delta h_{1-2} = \int v \, dp \quad .$$

The specific volume of the saturated liquid at 16 psia is 0.01674 ft^3/lb. Thus,

$$\text{pump work} = 0.01674(1200 - 16)/5.404 = 3.67 \text{ btu/lb} \quad .$$

So

$$h_{2VC} = 184.4 + 3.67 = 188.1 \text{ btu/lb} \quad .$$

The heat added in the boiler is

$$h_f - h_{2VC} = 1499.2 - 188.1 = 1311.1 \text{ btu/lb} \quad .$$

D-5.2 *Engine Work*

The theoretical engine work is found using the IMEP and the engine cylinder volume:

$$\begin{aligned} \text{work} &= \text{IMEP}(V_4 - V_2)/5.404 \text{ btu} \\ &= 100.2(22.2447 - 0.7945)/5.404 = 397.7 \text{ btu} \quad . \end{aligned}$$

Or, accounting for the weight of the steam used per cycle,

$$\text{specific work} = 397.7/1.057 = 376.3 \text{ btu/lb} \quad .$$

D-5.3 *Ideal Thermal Efficiency with Engine*

The theoretical thermal efficiency of the vapor cycle power system, at an engine operating condition of IMEP = 100 psi, is

$$E_C = \frac{376.3 - 3.67}{1311.1} = 28.42 \text{ percent} \quad .$$

With 2544 btu/indicated horsepower-hour, the steam rate at 100 psi IMEP is

$$2544/376.3 = 6.76 \text{ lb/ihp-hr} \quad ;$$

and the heat rate is

$$1311.1 \text{ btu/lb} \times 6.76 \text{ lb/ihp-hr} = 8{,}866 \text{ btu/ihp-hr} \quad , \quad \text{or}$$

$$8{,}866/0.7457 \text{ kW/hp} = 11{,}900 \text{ btu/ikW-hr} \quad .$$

D-5.4 *Thermal Efficiency with Turbine Expander*

The thermal efficiency of the vapor-cycle power system using the engine as an expander can be compared to the theoretical efficiency of the simple Rankine cycle that uses a steady-flow turbine-type steam expander as follows. Referring to figure D-1, the steam is expanded from the boiler feed conditions along a constant entropy line $S_f = 1.6293$ to the condenser pressure of 16 psia, where its heat content, found from the steam tables, is

$$h_{4R} = 1070.6 \text{ btu/lb} \quad .$$

Rankine thermal efficiency in this case is

$$E_R = \frac{\text{net work}}{\text{heat added}} = \frac{\text{heat added} - \text{heat rejected}}{\text{heat added}} = 1 - \frac{\text{heat rejected}}{\text{heat added}} \quad .$$

The heat rejected in the condenser is

$$h_{4R} - h_L = 1070.6 - 184.4 = 886.2 \text{ btu/lb} \quad ;$$

and the heat added in the boiler is the same 1311.1 btu/lb as before. So

$$E_R = 1 - \frac{886.2}{1311.1} = 32.41 \text{ percent} \quad ;$$

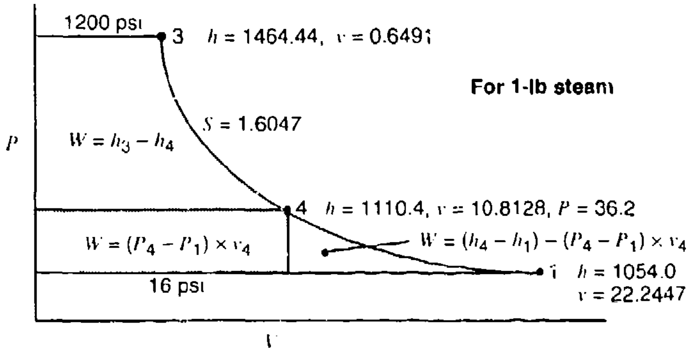
and the ratio of the engine Rankine cycle to the steady-flow Rankine cycle efficiencies is

$$28.42/32.41 = 87.7 \text{ percent} \quad .$$

D-5.5 *Incomplete Expansion Process*

The engine-type expander gives lower efficiency for the vapor-cycle power system than the turbine-type expander, because the engine does not fully expand the steam to the low 16-psia pressure level in the condenser. The reason the engine does not use full expansion is to allow it to be more compact. The piston stroke and cylinder would have to be considerably longer than required (more than two times) for a design that terminates expansion at the slightly higher pressure level of release (36.2 psia in the example above). The loss in efficiency due to the incomplete expansion operation of the engine can be estimated with the help of figure D-7, which is focused on a single pound of steam.

Figure D-7. Analysis for incomplete-expansion engine losses.



Note that the additional specific work, represented by the area of the complete expansion "triangle," is

$$(h_4 - h_1) - (P_4 - P_1)v_4/5.404$$

$$= (1110.4 - 1054.0) - (36.2 - 16)10.8128/5.404 = 16.0 \text{ btu/lb} \quad .$$

With this extra work accounted for in the full-expansion engine cycle,* the new specific engine work is

$$\frac{397.7 + 2.057 \times 16.0}{1.057} = 407.4 \text{ btu/lb} \quad ,$$

and the theoretical thermal efficiency E'_C becomes

$$E'_C = (407.4 - 3.67)/1311.1 = 30.79 \text{ percent} \quad ,$$

compared to the incomplete expansion value of 28.42 percent. The remainder of the losses, compared to the steady-flow Rankine vapor-cycle value of 32.41 percent, are likely due to the incomplete compression† and the admission mixing processes.

D-5.6 Carnot Thermal Efficiency

Another comparison of the vapor-cycle system performance is obtained from the theoretical thermal efficiency of the ideal Carnot thermodynamic cycle operating between the same temperature extremes:

$$E_{Carnot} = 1 - \frac{T_L}{T_H} = 1 - \frac{216.32 + 460}{1000 + 460} = 53.68 \text{ percent} \quad .$$

*Note that special exhaust valving would be required to expand all the steam to 16 psia and to compress only 1 lb back to near boiler conditions.

†An efficiency of 30.04 percent is obtained for the 100-psi IMEP incomplete expansion cycle when a compression ratio of 33 : 1 is used.

D-6. Comparison of Performance at Various Operating Conditions

In order to get a better picture of the steam engine operation, other cases of interest were analyzed. These include a half-power case of IMEP = 50 psia, a double-power case of IMEP = 200 psia, a case where the cutoff is 10 percent of the piston stroke (IMEP = 295 psia), a case where condenser back-pressure is increased to 19 psia, and a case where the engine compression ratio is 33 : 1. To speed the process, calculations were automated by means of a computer spreadsheet. The results are summarized in table D-1, with selected results displayed graphically in figure D-8. The arrangement of the entries in table D-1 follows the steps in the analysis in section D-3. Note the following from these results.

- The model is well-behaved over the broad range of power or cutoff conditions analyzed.

- The system efficiency peaks at an IMEP near 100 psi. The efficiency decreases for the IMEP = 50 psi case probably because only 0.535 lb of steam is being mixed with the 1.0 lb of compressed steam. The incoming steam is thereby cooled to a proportionally greater extent than in the IMEP = 100 psi case, where 1.057 lb of steam is admitted.

- The higher compression ratio of 33 : 1 results in the residual steam being compressed to essentially boiler conditions (P_2 = 1193 psia and T_2 = 997°F). As expected, this results in significantly higher efficiency operation (30.04 versus 28.42 percent).

- The higher exhaust back-pressure of 19 psia results in compression to higher pressure and temperature than when the lower pressure of 16 psia is used. This increased back-pressure also improves efficiency somewhat (29.39 versus 28.42 percent). The reason for this is probably the significantly increased energy of the compressed steam being mixed during admission.

- Note that none of the compression pressures reached the admission value of 1200 psia. The result is that a full 1 lb of compressed steam is always mixed with the admitted steam. No steam escapes through the pressure relief valve (as planned, since the model does not handle escaping steam anyway).

- Note that the size of the engine changes for each new case analyzed, ranging from 20.94 to 21.75 ft^3 for the standard cases.* This feature accommodates the "exactly 1 lb" of compressed steam requirement when the cycle operates on different entropy lines. This is a deficiency in the model that may cause the level of efficiency predicted to be too low.

*Cylinder volume for the actual engine is 0.44 in.^{3}

Table D-1. Summary of results for vapor-cycle engine and power system analysis

Parameter[a]	Units	Case 1[b]	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
IMEP*	psi	50	100	150	200	295	100	100
Entropy[c]	—	1.5742	1.6047	1.6139	1.6183	1.6224	1.6263	1.6287
Feed steam*								
P_f†	psia	1200.0	1200.0	1200.0	1200.0	1200.0	1200.0	1200.0
T_f†	°F	1000.0	1000.0	1000.0	1000.0	1000.0	1000.0	1000.0
v_f	ft³-lb	0.6843	0.6843	0.6843	0.6843	0.6843	0.6843	0.6843
H_f	btu/lb	1499.2	1499.2	1499.2	1499.2	1499.2	1499.2	1499.2
S_f	—	1.6293	1.6293	1.6293	1.6293	1.6293	1.6293	1.6293
Point 1 steam								
P_1*†	psia	16.0	16.0	16.0	16.0	16.0	19.0	16.0
T_1	°F	216.3	216.3	216.3	216.3	216.3	225.2	216.3
v_1	ft³/lb	21.7177	22.2447	22.4036	22.4797	22.5505	19.4349	22.6594
h_1	btu/lb	1033.4	1054.0	1060.2	1063.2	1065.9	1080.2	1070.2
S_1†	—	1.5742	1.6047	1.6139	1.6183	1.6224	1.6263	1.6287
Quality	%	87.7	89.9	90.5	90.8	91.1	92.2	91.5
R_{comp}*	to 1	28.00	28.00	28.00	28.00	28.00	28.00	33.00
Point 2 steam								
P_2	psia	875.6	926.3	941.9	949.3	956.7	1169.5	1193.1
T_2	°F	775.8	859.7	886.1	898.8	911.4	984.6	996.7
v_2†	ft³/lb	0.7756	0.7945	0.8001	0.8028	0.8054	0.6941	0.6866
h_2	btu/lb	1381.3	1427.7	1442.3	1449.4	1456.2	1491.2	1497.5
S_2†	—	1.5742	1.6047	1.6139	1.6183	1.6224	1.6263	1.6287
Point 3 steam								
P_3†	psia	1200.0	1200.0	1200.0	1200.0	1200.0	1200.0	1200.0
T_3	°F	869.5	940.6	962.8	973.4	983.3	992.8	998.6
v_3	ft³/lb	0.6062	0.6491	0.6622	0.6685	0.6744	0.6800	0.6834
h_3	btu/lb	1422.42	1464.44	1477.44	1483.66	1489.45	1494.96	1498.35
S_3†	—	1.5742	1.6047	1.6139	1.6183	1.6224	1.6263	1.6287
Feed weight	lb	0.535	1.057	1.615	2.207	3.408	0.898	1.017
Engine properties								
V_1	ft³	21.7177	22.2447	22.4036	22.4797	22.5505	19.4349	22.6594
V_2	ft³	0.7756	0.7945	0.8001	0.8028	0.8054	0.6941	0.6866
V_3	ft³	0.9306	1.3353	1.7316	2.1437	2.9731	1.2909	1.3783
V_4	ft³	21.7177	22.2447	22.4036	22.4797	22.5505	19.4349	22.6594
Displacement	ft³	20.94	21.45	21.60	21.68	21.75	18.74	21.97
R_{exp}	to 1	23.34	16.66	12.94	10.49	7.58	15.06	16.44
Point 4 steam								
P_4	psia	25.97	36.24	47.64	60.14	87.78	39.35	35.47
T_4	°F	242.19	261.34	277.98	292.86	330.97	266.25	260.07
v_4†	ft³/lb	14.1463	10.8128	8.5682	7.0106	5.1153	10.2377	11.2361
h_4	btu/lb	1065.34	1110.38	1137.45	1158.59	1192.00	1132.08	1126.12
S_4†	—	1.5742	1.6047	1.6139	1.6183	1.6224	1.6263	1.6287
Engine properties								
Cutoff	%	0.74	2.52	4.31	6.19	9.97	3.18	3.15
R_{cutoff}	to 1	1.20	1.68	2.16	2.67	3.69	1.86	2.01
IMCP	psi	74.0	76.4	77.3	77.7	78.1	94.9	84.3
IMXP	psi	115.9	150.2	183.5	217.0	280.8	161.9	150.7

Table D-1. Summary of results for vapor-cycle engine and power system analysis (cont'd)

Parameter[a]	Units	Case 1[b]	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
Engine properties (cont'd)								
IMFP	psi	123.9	176.6	227.3	277.8	372.5	195.0	183.7
IMEP	psi	50.0	100.2	150.1	200.1	294.3	100.1	99.4
IMEP error	psi	0.0	0.2	0.1	0.1	–0.7	0.1	–0.6
Point 4C properties								
P_{4C}†	psia	16.00	16.00	16.00	16.00	16.00	19.00	16.00
T_{4C}	°F	216.32	216.32	216.32		298.95	225.24	216.32
v_{4C}	ft^3/lb	22.5351	23.6863	24.3781	24.8644	27.9795	20.6145	24.0887
h_{4C}†	btu/lb	1065.34	1110.38	1137.45	1158.59	1192.00	1132.08	1126.12
S_{4C}	—	1.6215	1.6881	1.7282	1.7593	1.8056	1.7050	1.7114
Cycle properties								
h_{4R}	btu/lb	1070.6	1070.6	1070.6	1070.6	1070.6	1080.4	1070.6
$P_{pump\ in}$†	psia	16.0	16.0	16.0	16.0	16.0	19.0	16.0
v_{pump}	ft^3/lb	0.01674	0.01674	0.01674	0.01674	0.01674	0.01681	0.01674
h_{LC} pump	btu/lb	184.4	184.4	184.4	184.4	184.4	193.4	184.4
Pump work	btu/lb	3.6677	3.6677	3.6677	3.6677	3.6677	3.6737	3.6677
h_{2L}	btu/lb	188.09	188.09	188.09	188.09	188.09	197.09	188.09
Cycle performance								
Heat in	btu/lb	1311.11	1311.11	1311.11	1311.11	1311.11	1302.11	1311.11
Gross work out	btu	193.73	397.76	599.86	802.52	1184.29	347.14	404.14
Work out	btu/lb	361.97	376.22	371.49	363.70	347.46	386.42	397.52
Net work	btu/lb	358.30	372.55	367.82	360.04	343.79	382.74	393.85
Water rate	lb/ihp-hr	7.03	6.76	6.85	6.99	7.32	6.58	6.40
Heat rate	btu/ihp-hr	9214.8	8865.7	8978.6	9170.8	9599.7	8572.5	8390.7
Efficiency	%	27.33	28.42	28.05	27.46	26.22	29.39	30.04
Steady-flow Rankine	%	32.41	32.41	32.41	32.41	32.41	31.88	32.41
Carnot efficiency	%	53.68	53.68	53.68	53.68	53.68	53.07	53.68

[a]**Symbols:**
* = independent variable
† = independent property (see sect. D-3)
P = pressure
T = temperature
V = volume
v = specific volume
h = specific enthalpy
S = entropy
R = ratio

Subscripts:
f = feed steam
1, 2, 3, 4 = points in cycle
comp = compression
exp = expansion
C = condenser input properties
L = liquid

Acronyms:
IMEP = indicated mean effective pressure
IMCP = indicated mean compression pressure
IMXP = indicated mean expansion pressure
IMFP = indicated mean forward pressure

[b]Case 1: For IMEP = 50
Case 2: For IMEP = 100
Case 3: For IMEP = 150
Case 4: For IMEP = 200
Case 5: For cutoff of 10%
Case 6: For exhaust pressure increased to 19 psi
Case 7: For compression ratio increased to 33 : 1.

[c]Final value for entropy line; determined by iteration

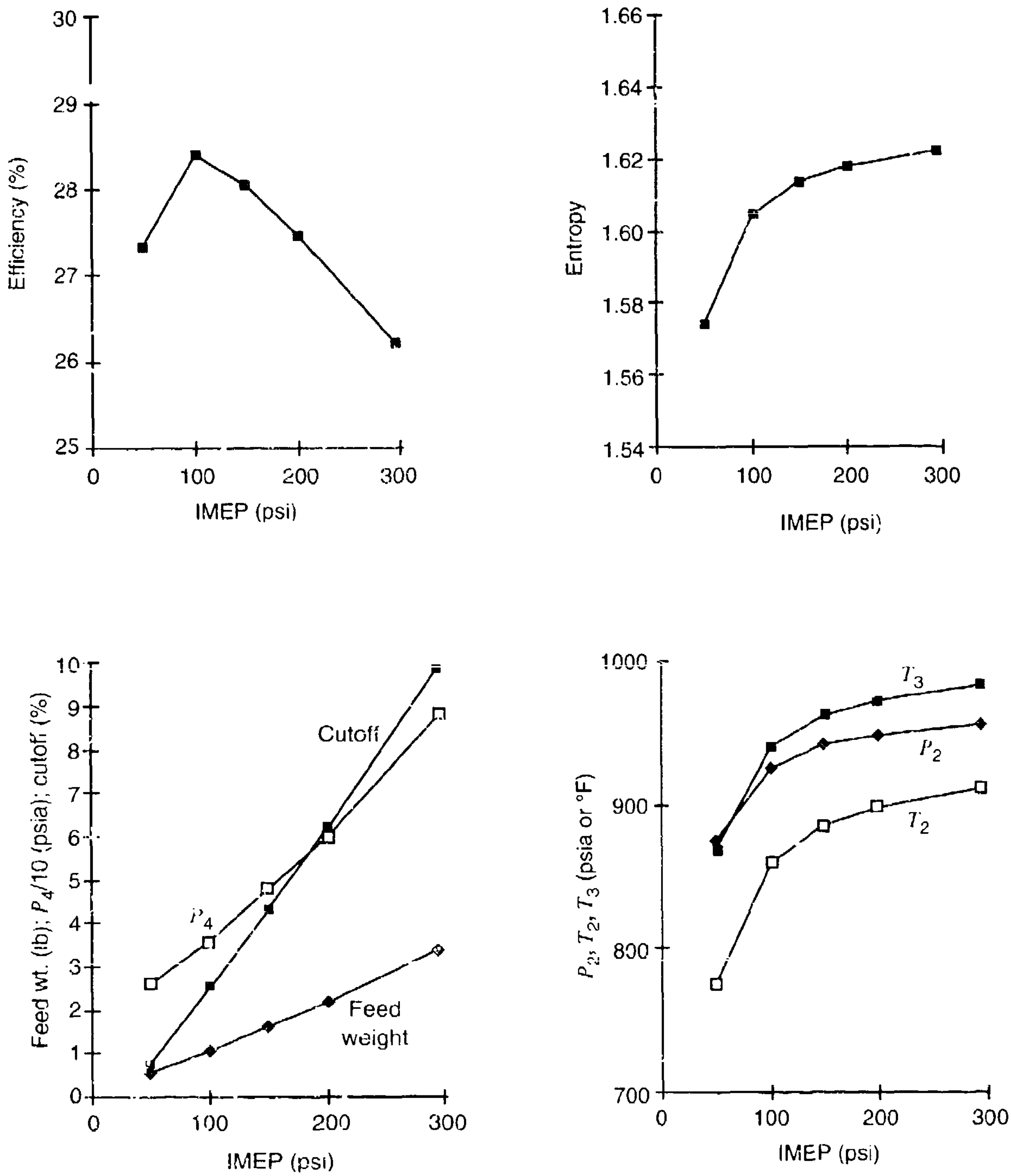


Figure D-8. Selected results from vapor-cycle power system analysis.

D-7. Losses in Thermodynamic Cycle

The thermodynamic cycle analyzed to this point does not include any losses due to inefficiencies in the various elements such as the condenser, pump, boiler pipes, and valves. For a more rigorous analysis, models and experimental data for each of these elements can be found in the literature. Only rough estimates of the losses for these elements are given here.

D-7.1 Condenser Losses

Steam exhausts into the condenser at a back-pressure of 16 psia. However, the pressure in the water tank can be assumed to be at atmospheric pressure (14.7 psia), resulting in a 1.3-psia pressure loss in the condenser. A comparison of the temperature and enthalpy for the saturated liquid at these two pressures is as follows:

P (psia)	T (°F)	h_L (btu/lb)
16.0	216.3	184.4
14.7	212.0	180.1
Δ	–4.3	–4.3

One can further assume that the liquid in the water tank cools to below 212°F, to 185°F for example, but that about 80 percent of this temperature differential is recovered by a feed water heater element that extracts waste heat from the exhaust steam. Thus, the temperature of the feed water would be 185 + 0.8(212 – 185) = 206.6°F, and the value for h_l at point 1 in the Rankine cycle would be 180.1 – (212 – 206.6) × 1 btu/lb/°F = 174.7 btu/lb, which is 9.7 btu/lb lower than the 184.3 btu/lb used for the ideal case.

D-7.2 Pipe, Pump, and Valve Losses

Losses due to the turbulence and friction in the boiler pipes and admission valve can be assumed to require increased pressure from the pump in order for the output pressure to be 1200 psia after these losses. Assuming a 50-psi loss in each element, the pump outlet pressure would have to be 1300 psia instead of the 1200 psia assumed for the ideal analysis. Also, the pump will not be 100-percent efficient because of leakage through the clearances, nonadiabatic thermodynamics, etc. Using the specific volume of saturated liquid at 206°F to help account for the revised inlet conditions, assuming that the pump is 70-percent efficient, and accounting for the pressure drops, we obtain for the pump work with losses

$$0.01667(1300 - 14.7)/(0.7 \times 5.404) = 5.664 \text{ btu/lb} \quad ,$$

compared to a value of 3.668 btu/lb for the ideal case.

The heat content of the compressed boiler feed-water (assuming that the pump inefficiency is not recovered as heat) would be

$$h_{2VC} = 174.7 + 0.7 \times 5.664 = 178.7 \text{ btu/lb} \quad .$$

Assuming that the steam admitted into the engine cylinder is throttled at constant enthalpy from the boiler output conditions, the heat content of the steam output from the boiler is 1499.2 btu/lb, and its pressure and temperature are 1250 psia and 1002.6°F.

The heat added in the boiler is 1499.2 − 178.7 = 1320.5 btu/lb, and the efficiency of the vapor-cycle power system including the above losses at IMEP = 100 psi is now

$$E_C = \frac{376.3 - 5.664}{1320.5} = 28.07 \text{ percent} \quad ,$$

compared to the 28.42 percent determined without these losses. Thus, the condenser, pump, piping, and valve losses can be expected to reduce the vapor-cycle thermal efficiency about 0.5 percent, or perhaps as much as 1 percent.

D-7.3 *Engine Losses*

We can make a rough estimate of the thermodynamic and fluid-flow losses within the engine, due to nonisentropic processes, turbulence and fluid friction, throttling pressure drops during admission and exhaust, leakage, thermal losses, etc, by assuming that the area (representing work) of the actual indicated P/V diagram would be about 10 percent less than the theoretical area of 397.7 btu (IMEP = 100 psi case), as shown in figure D-9.

Thus, specific engine work might be

$$\frac{397.7 - 39.8}{1.057} = 338.6 \text{ btu/lb} \quad .$$

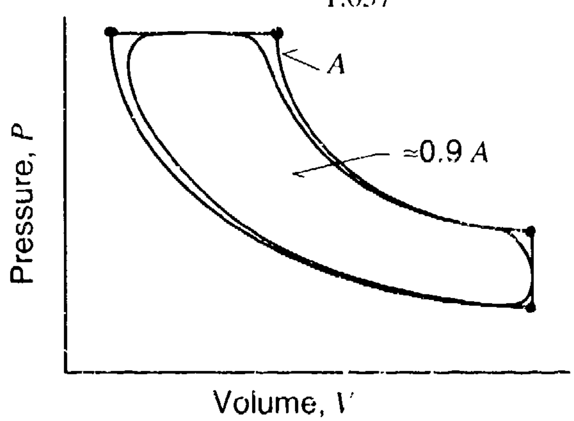


Figure D-9. Ideal engine cycle diagram and estimate for realistic diagram.

With this estimate, the thermal efficiency of the vapor cycle, including the losses discussed previously, is

$$E_C = \frac{338.6 - 5.7}{1320.5} = 25.21 \text{ percent} \quad .$$

So the overall efficiency reduction from the ideal is about 3 to 4 percent.

Other inefficiencies for the entire power plant, such as fuel-burner to water-heating efficiency, engine friction and accessory losses, and the inefficiency of the motor/generator device, are discussed in the body of the report.

D-8. Conclusions and Observations

The positive-displacement engine-type expander is expected to be much better suited to low-power applications (1 to 10 indicated hp, in the case of the soldier power system) than is the turbine-type steady-flow expander [7]. The model shows that reasonably high efficiencies can be expected from the high-compression uniflow steam engine vapor-cycle system.

It was anticipated that the high-compression processing by the engine of some of the expanded steam back to boiler output conditions (thereby bypassing the condensation and vaporization processes for a significant portion of the processed steam) would result in even higher economies than the model predicts. For example, for the ideal IMEP = 100 psi case, the change in enthalpy for the residual 1 lb of compressed exhaust steam would be $h_F - h_4$ = 1499.2 – 1110.4 = 388.0 btu/lb (work done by the engine), versus $h_f - h_L$ = 1499.2 – 184.4 = 1314.8 btu/lb (work done by the boiler) for 1 lb of exhaust steam that is first condensed to liquid. This is an apparent savings of 1314.8 – 388.8 = 926 btu/lb for part of the steam flowing only through the engine. In this model at least, the 1 lb of residual steam is merely recycled through the engine, delivering about as much work as it takes to compress it each cycle, with no apparent effect on economy.

The Williams Co. reportedly achieved thermal efficiency measurements in the lab on the order of 35 percent for one of their engines operating on 1000 psia and 980°F steam [8]. On the basis of this model, achieving such efficiencies is questionable. (However, other references [9] also indicate that higher than Rankine thermal efficiencies can be obtained by direct compression of exhaust vapor using a simple rotary compressor/expander having a very high volumetric compression ratio: 70 : 1.) By operating at a higher compression ratio such as 33 : 1, efficiency can be improved by at least 1 to 2 percent, as shown in the example case presented in figure D-8. This would put indicated efficiency, with losses, at a level of 27 to 28 percent at best. However, it may turn out that even higher efficiencies are predicted when the model refinements discussed in section D-9 are investigated.

There may be exhaust conditions, such as higher entropy values, existing in the residual exhaust steam of an actual operating engine, that result in compression temperatures that exceed boiler temperature at the 1200-psia relief valve setting. Another possibility is to set the relief valve pressure at a level higher than boiler pressure in order to insure compression temperatures higher than boiler temperature. Also, for a compression ratio that is high enough to always blow off part of the compressed steam through the relief valve (whatever the pressure setting) and back into the feed-steam manifold, so that only part of the compressed steam is used each expansion cycle, some of the economy expected from bypassing the condenser process might be realized.

Even if high compression pressures cannot improve thermal efficiency to the 30-percent level for the overall vapor cycle with losses, they do result in the highest level of engine efficiency, compared to noncompression uniflow engine systems [10]. This is because the cylinder and piston heating resulting from the high compression temperature completely eliminates any condensation or significant cooling of the incoming steam.

D-9. Recommendations

Refinements of the vapor-cycle power system and steam engine model should proceed per the following guidance.*

- Start with a fixed engine size (cylinder volume) and specify various percentages of cutoff.
- Accommodate a variable weight for the steam being compressed and for the weight remaining after blowoff to be mixed during admission.
- Increase compression ratio to 33 : 1 or 35 : 1
- Provide a pressure relief valve function in the cylinder head that vents compressed steam to the admission manifold whenever the preset pressure is exceeded in the compression process.
- Calculate IMEP produced by each specified value of cutoff and pick operating values of IMEP from the graph of these results.
- Develop a model using automated steam tables to determine properties at various points in the cycles. Evaluate performance at feed-steam pressure and temperature conditions of 1500 psi and 1200°F for comparison with baseline conditions.

*_I feel that these analyses and associated experiments would make ideal projects for engineering students majoring in the discipline of thermodynamics._

- Develop detailed loss mechanisms for each phase of the engine cycle so as to predict actual indicated effective pressure instead of ideal IMEP.

- Consider alternative admission and exhaust processes that do not necessarily support a common entropy line to define engine operation.

- Develop or adapt detailed models, based on material available in the engineering literature, for all the elements of the vapor-cycle system (boiler, piping, condenser, feed pump, valves, etc).

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