Comparative Thermodynamic Performance of Some Rankine/Brayton Cycle Configurations for a Low-Temperature Energy Application

F. L. Lansing
DSN Engineering Section

Various configurations combining solar-Rankine and fuel-Brayton cycles were analyzed in order to find the arrangement which has the highest thermal efficiency and the smallest fuel share. A numerical example is given to evaluate both the thermodynamic performance and the economic feasibility of each configuration. The solar-assisted regenerative Rankine cycle was found to be leading the candidates from both points of energy utilization and fuel conservation.

I. Introduction

In an effort to increase the thermal efficiency of power cycles driven by solar energy, the concept of a regenerative solar-assisted Rankine cycle was recently introduced. The main feature of this concept is that the heat addition in the cycle is made along a constant pressure close to atmospheric, by two heat reservoirs. Water is used as the working medium. The first heat reservoir is essentially a solar collector/boiler which supplies sufficient energy to convert liquid water to dry saturated steam, i.e., supplies a little more than the latent heat of vaporization. The evaporation temperatures are consistent with the capability of low concentration ratio collectors and range from 90°C to 160°C. The second heat reservoir is a high-temperature source of energy such as that obtained from fossil fuel combustion. This will superheat the steam, before expansion in the turbine, to a very high temperature comparable to that used in conventional power plants. The fraction of total heat input in the cycle provided by the second heat reservoir ranges from 20 to 25% as reported in Refs. 1 and 2. Since the working pressures of the cycle, the evaporation and condensation pressures, are close to each other in this case, the condition of steam leaving the turbine is still at a high superheat temperature and can be further exploited by adding regenerators to the cycle. Regenerators are favored when a thermal potential difference exists between turbine exhaust and feed water temperatures since they improve the cycle efficiency without affecting the net work output.

The regenerative solar-assisted Rankine cycle has a much higher thermal efficiency (about double) than its comparable simple 100% Solar Rankine. It will be compared in performance with other dual cycles for the selection of the optimum cycle configuration.

II. Fuel Utilization Versus Fuel Conservation

Several inquiries were raised during the early stages of study about whether or not the solar-assisted cycle is superior from the viewpoints of fossil fuel conservation, "best" fuel utilization or economic feasibility. The inquiry about fuel conservation was made since the thermodynamic cycle requires that a fraction of its total energy input (20-25%) be from fossil fuel. If the cost of fuel keeps on rising with current inflation rates, the selection of a fuel-powered system, based on efficiency superiority alone, might be outweighed by less efficient systems that have lesser or no dependence on fossil fuel.

On the other hand, if the concept of partial assistance by fossil fuel in a cycle was accepted, then the next inquiry would be to look into how "well" the fuel was utilized. Figure 1 shows two different schemes in which both solar energy and fossil fuel are used via power cycles. Both schemes take in the same quantities of solar or fuel energy. The first scheme is composed of two physically separate cycles: a 100% fuel-powered, high-temperature Brayton cycle and a 100% solar-powered low-temperature Rankine cycle. The second scheme is the regenerative solar-assisted Rankine cycle. The choice of the Brayton (Joule) cycle for coupling with the Rankine one was based on the fact that the Brayton cycle is composed of two isobars (constant pressure processes) and two isentropics (constant entropy processes) which makes it convenient in matching the border lines of the two cycles. The analysis is seeking the answer to which of these two schemes is more efficient, i.e., conserving energy by producing the same output work from less input heat.

It should be borne in mind that purchasing electric power directly from a utility company is equivalent in principle to utilizing the fossil fuel via a Brayton cycle, and, if combined with a solar-thermal plant, will be treated as a combined scheme in which both solar and fossil fuel participated by unequal shares in the production of the net work as in Fig. 1a.

III. Main Assumptions

In the next comparative analysis between different configurations of the Brayton/Rankine cycles, the main assumptions and idealizations are made as follows:

(1) The working fluid is water.

(2) Operation will be on clear days only, when the sun is available and able to provide the required quantities of heat to run the system efficiently. No allowance for partial shading due to clouds or any unexpected side effects which hinders a full load operation of a 100% solar-powered Rankine cycle.

(3) All processes are reversible. The sources of irreversibilities in compression and expansion processes are assumed to change the thermal efficiency in each configuration with almost an even hand. This means that the descending order of thermal efficiency of the different configurations under study will be assumed to be unchanged when irreversibilities due to friction or turbulence do occur.

(4) All heat exchangers, whether they are recuperators or regenerators, are adiabatic and ideal, i.e., have a 100% effectiveness.

(5) Both Brayton and Rankine cycles are operating at the same pressure limits in all configurations. The low pressure limit is usually determined by the cooling medium temperature. For example, a sink temperature ranging from 20°C (86°F) to 50°C (122°F) will correspond to a pressure range from 0.23 to 1.24 N/cm^2, respectively. On the other hand, the high pressure limit is determined by the ability of the solar collector to evaporate water into steam. An evaporation temperature ranging from 90°C to 160°C will correspond to a saturation water pressure ranging from 7 to 60.78 N/cm^2, respectively.

(6) Both cycles are chosen to have one common border line (or one common process) in the property diagram. This border line is the isentropic (reversible adiabatic) expansion in the Rankine cycle turbine, and it also represents the isentropic compression in the Brayton cycle compressor. The two mechanical components, the turbine and the compressor, have to be physically separate to identify the sequence of events in each cycle. However, because of their equal but opposite action along the border line, their combined effect, from a thermodynamics viewpoint, has zero energy exchange with the surroundings, i.e., a null effect and could be later dropped for simplicity.

IV. Cycle Configurations

Four configurations, using various combinations of the Rankine cycle and the Brayton cycle, were adopted for this study:

(1) A simple Rankine cycle and a simple Brayton cycle (two separate cycles).

(2) A simple Rankine cycle and a regenerative Brayton cycle (two separate cycles).

(3) A regenerative Rankine cycle and a regenerative Brayton cycle (two separate cycles).

(4) A combined regenerative Rankine and Brayton cycle (one cycle). This is called the "solar-assisted Rankine cycle."

Each of these configurations is explained in detail below.

A. Configuration 1

This configuration consists of two separate cycles: a simple Rankine cycle and a simple Brayton cycle. The flow diagram is shown in Fig. 2, and the cycles are presented on the temperature-entropy (t-S) diagram for water in Fig. 3. The simple Rankine cycle is started by extracting the condensate from the condenser well (state 1), and pumping it to the boiler (state 2). The heat added to the Rankine cycle is by means of a set of solar collectors for all three stages: sensible heating from state 2 to 3, evaporation from state 3 to 4, and a small amount of superheat from state 4 to 5. The small superheat portion is needed only to provide adequate comparison with other configurations on an equal basis and to avoid the presence of any wet steam in the matched Brayton cycle. State 5 is chosen such that, if followed by an isentropic expansion in the turbine, it will end up with state 6 as a dry and saturated condition. In the Brayton cycle, the heat is added entirely by combustion of any kind of fossil fuel. The cycle is working between the same solar-boiler pressure P_B and the Rankine cycle condenser pressure P_c. Superheated (or at least dry saturated) steam is the condition of the working fluid throughout the cycle. The obedience of superheated steam to ideal gas relations at low pressures and high temperatures makes the Brayton cycle in this case very close to an ideal-gas Brayton cycle. Dry and saturated steam (state a) is compressed isentropically in a compressor to state b. The fuel heat addition is followed to superheat to state c at constant pressure. An expansion in the turbine from state c to d and a heat rejection from state d to a will complete the cycle. It is important to note that the position restrictions made on state a and on the process a→b are artificial and do not necessarily represent all possible cycle positionings with respect to each other as indicated in assumption (6).

The total heat input to configuration 1 is determined by the area i-1-2-3-4-5-c-d-k-i in Fig. 3 which is the sum of the heat added by the sun (area i-1-2-3-4-5-j-i) and the heat added by the fuel (area j-b-c-d-k-j). The heat rejected is the sum of two parts: the part rejected from the Rankine cycle condenser (area i-1-6-j-i) and that rejected from the Brayton cycle cooler (area j-a-d-k-j). The net work done is the loop area shaded 1-2-3-4-5-c-d-a-1.

Figure 4 shows another way of doing configuration 1 with less mechanical components. The Rankine-turbine/Brayton-compressor connection is deleted, thus leaving the configuration identical to a simple, extra superheated Rankine cycle with two heat exchangers for heat addition and two heat exchangers for heat rejection. The configuration presented in Fig. 4 is totally equivalent, from a thermodynamics viewpoint, to that presented in Fig. 2.

B. Configuration 2

This configuration consists of two separate cycles: a simple Rankine cycle, as in configuration 1, and a regenerative Brayton cycle to improve the simple Brayton cycle of configuration 1. The flow diagram is sketched in Fig. 5 and the cycles are presented on t-S diagram for water in Fig. 6. The regenerator is working under a temperature potential difference $(t_d - t_b)$. In Fig. 6 the areas under the lines $\overline{be}$ and $\overline{df}$ are equal and according to assumption (4), the temperatures t_e and t_f are equal to t_d and t_b, respectively.

With the common process between the two cycles; the process from state 5 to 6 or from state a to b, the net work output in configuration 2 is *unchanged* compared to configuration 1 and is equal to the loop area 1-2-3-4-5-c-d-a-1. On the other hand, the total heat added in configuration 2 is reduced by the hatched area under line $\overline{be}$ as compared to configuration 1 due to the regeneration action. The result is a higher thermal efficiency for configuration 2 compared to configuration 1, provided that the temperature potential difference $(t_d - t_b)$ is positive.

C. Configuration 3

This configuration is one step forward in improving the efficiency of configuration 2 and is illustrated in Figs. 7 and 8. First, the regenerator in the Brayton cycle of configuration 2, given the name regenerator 1 in Fig. 8, is kept intact. Second, the superheated steam leaving the Brayton turbine at state f is still at a high-temperature potential that offers some energy savings if coupled with the Rankine cycle. The following two extra generators are used in the Rankine cycle:

(1) A vapor-vapor regenerator, given the name regenerator 2 in Fig. 8, in which the superheat energy needed from state 4 to 5 is provided by cooling from state f to g in Fig. 7. According to assumption (4), the areas in Fig. 7 under lines $\overline{fg}$ and $\overline{45}$ are equal and the

temperatures t_f and t_4 are equal to t_5 and t_g, respectively.

(2) A vapor-liquid regenerator, given the name regenerator 3 in Fig. 8, in which the sensible heat needed from state 2 to 3 is provided by cooling from state g to a. According to assumption (4), the areas in Fig. 7 under lines $\overline{ga}$ and $\overline{23}$ are equal and the temperatures t_g and t_1 ($\simeq t_2$) are equal to t_3 and t_a, respectively.

In this configuration, the net work output is still the same as in configurations 1 and 2, and is equal to the closed loop area 1-2-3-4-5-c-d-a-1. The heat addition in configuration 3, on the other hand, is different from both configurations 1 and 2. For the Rankine cycle, the heat is added only from state 3 to 4, i.e., the latent heat of vaporization part, and for the Brayton cycle, the heat is added only from state e to c. The sum of heat added to both cycles in this configuration is less than the heat added in configuration 2. The difference in heat addition is represented in Fig. 7 by the area under line $\overline{fa}$. The conclusion that can be made accordingly is that the thermal efficiency of configuration 3 is higher than that for configuration 2 which in turn is higher than the efficiency of configuration 1. The superiority of configuration 3 over configuration 1 is established by the fact that the heat rejection process from state d to a in the cooler is replaced entirely by three regenerators: (1) a part from d to f for regeneration from b to e in regenerator 1, (2) a part from f to g for regeneration from 4 to 5 in regenerator 2, and (3) a part from g to a for regeneration from 2 to 3 in regenerator 3.

D. Configuration 4

This configuration is identical from a thermodynamics viewpoint to configuration 3 but has less components to operate with. The configuration is sometimes called a "solar-assisted regenerative Rankine cycle" since it is not 100% powered by the sun. The Rankine-turbine/Brayton-compressor connection of configuration 3 is omitted and regenerators 1 and 2 are replaced by one regenerator doring their function exactly. The flow diagram for configuration 4 is presented in Fig. 9, and the thermodynamic cycle on the *t-S* diagram is identical to that illustrated in Fig. 7 except that the common process a→b or 5→6 is omitted. This configuration possesses the same high thermal efficiency of configuration 3 and is, therefore, superior to configurations 1 and 2.

V. Numerical Comparison of the Four Configurations

To calculate the efficiency and energy savings in the different configurations and the order of efficiency improvement in each case, a numerical comparison is set based on the following conditions:

(1) Maximum Brayton cycle temperature (t_c) 600°C.

(2) Condenser temperature 50°C.

(3) Condenser pressure 1.24 N/cm^2.

(4) Solar boiler evaporation temperature 100°C.

(5) Evaporation pressure 10.13 N/cm^2.

(6) Compression and expansion processes are isentropic.

(7) Heat exchangers are 100% effective.

(8) Neglect the pump work from state 1 to 2.

The numerical results of the various configurations are listed in Table 1. Also, a plot of the efficiency of each configuration against the percentage of the fuel share in each is illustrated in Fig. 10. The following remarks can be made from Table 1 and Fig. 10:

(1) The simple 100% solar-powered Rankine cycle gave a thermal efficiency of 14.43% as compared to 37.87% for the simple 100% fuel-powered Brayton cycle. This is mainly due to the large difference in heat source temperature of the solar-Rankine cycle (taken at 100°C in the example) compared with the fuel-Brayton cycle (taken at 600°C in the example). The lowest sink temperature in both cycles was the same at 50°C. On the other hand, the specific steam consumption (the amount of steam in kg needed to produce 1-kWh output at the turbine shaft) is 8.96 kg/kWh for the simple Rankine cycle versus 13.44 kg/kWh for the simple Brayton cycle. Small specific steam consumption means small size plant for the same power output and low maintenance cost.

(2) Configuration 1, which is a straightforward combination of the two simple Rankine/simple Brayton cycles, yielded a thermal efficiency of 19.19% with a partial assistance of the fuel in the heat input of 20.26%. The specific steam consumption dropped to 5.37 kg/kWh$_e$, which favors the combination of the Rankine/Brayton cycles for plant compactness.

(3) Configuration 2, which is a combination of the simple Rankine and the regenerative Brayton, showed a slight improvement of the thermal efficiency (19.40% versus 19.19% for configuration 1). The percentage improve-

ment in efficiency can be greater if the temperature difference for regeneration $[(t_d - t_b)$ in Fig. 6] was made larger at different working conditions. Note that the specific steam consumption was the same in configurations 1 through 3.

(4) The efficiency superiority of configuration 3 over the other configurations can be analyzed in conjunction with Fig. 1 as follows:

If two separate cycles, as shown in Fig. 1, were constructed such that they take in the same quantities of solar and fuel heat as those of configuration 3, the solar heat (77.1 units) is added to a 100% solar-powered Rankine cycle whose efficiency is 14.43% (case 1 in Table 1), and the fuel heat (22.9 units) is added to a 100% fuel-powered Brayton whose efficiency is 37.86% (case 2 in Table 1), the resulting net work output would be $\{(0.1443 \times 77.1) + (0.3786 \times 22.9)\}$, i.e., 19.80 units only. The latter is less than 22.90 units for configuration 3 with the same energy shares. The result is that configuration 3 (or identically configuration 4) will rank second to the 100% fuel-powered Brayton in energy conservation only. However, from the point of "fossil fuel" conservation, the answer is not definite at this stage.

VI. The "Economic Feasibility Ratio"

In order to establish the economic feasibility of each of the above configurations, a rough estimate is needed of the cost of heat energy input and the mechanical energy output. If X\$ is the cost of one energy unit (kWh_t) from a solar collection system, Y\$ is the cost of the same unit from fuel combustion, and Z\$ is the cost of electricity unit (as purchased from a utility company), the cost of input and output energy can be listed as in Table 2. An "Economic Feasibility Ratio" (EFR) can then be introduced as an indication of the size of payback period of the money invested in such energy utilization plants. The EFR is here defined as the cost of input heat energy to the cost of output mechanical energy. Higher EFR values means higher installation and operation cost than the cost savings (for a consumer) or the profit (for a utility company) of the mechanical energy output. The ratio (EFR) will reach unity if the costs were integrated over the payback period of the invested money. However, a nonintegrated value (or instantaneous value) of the EFR leads to a small payback period and a direction towards economic feasibility.

A rough estimate of the cost/kWh_t of solar heat could be found from the loan payment to cover the solar equipment divided by the heat collected during the payment period, which will be taken as the life period of the equipment. For example, to yield a 100°C boiling temperature, at good collector efficiency, a low concentration ratio collector that costs approximately \$150/m^2 (1977 price) is needed. The collector is capable of collecting a daily average of about 4 kWh_t/m^2 for a full year in a city like Los Angeles. If approximately three times the collector cost is needed for the whole installation (to include pipework, storage tanks, land cost, etc.) with a life period of 15 years, and an interest on the borrowed money of 8%, then the cost/kWh_t will be:

$$X = \frac{1.752 \times 150 \times 3}{4 \times 365 \times 15} = 3.6¢/kWh_t$$

which will be rounded off to 4¢/kWh_t. The cost of gaseous fossil fuels alone, such as natural gas, propane etc., can be taken as 1.2¢/kWh_t and adding the cost of the equipment that goes with it, the cost Y may be taken as 2¢/kWh_t. The cost of purchased electricity is taken roughly as 5¢/kWh_e. In Table 2, the EFR is calculated for each configuration with these estimated costs. The 100% fuel-powered Brayton cycle has the least payback period, followed by the solar-assisted regenerative Rankine (case 5, Table 2), and the last in the list of economic feasibility is the simple 100% Rankine (case 1, Table 2). On the other hand, from the fuel conservation point of view, the 100% solar-powered with 0% fuel share is leading the list. The three Rankine/Brayton configurations follow with close boundaries of fuel share (19.4→22.9%), and tailing the list is the 100% fuel-powered Brayton cycle. The final selection of the "best" configuration combining both fuel conservation and energy utilization points, depends of course on the rate of escalation of X, Y, and Z.

VII. Summary

Various configurations combining solar-Rankine and fuel-Brayton cycles were studied in order to find the arrangement which has the highest thermal efficiency and the smallest fuel share. The solar-assisted regenerative Rankine cycle was leading the candidate configurations in this respect.

A simple criterion was defined for comparing the economic feasibility of each configuration, and a simple numerical example was given. Again, the solar-assisted regenerative cycle was found leading the other configurations in having a short payback period.

References

1. Lansing, F. L., "Computer Modelling of a Regenerative Solar-Assisted Rankine Power Cycle," *The Deep Space Network Progress Report 42-37,* pp. 152-168, Jet Propulsion Laboratory, Pasadena, Calif., Feb. 15, 1977.

2. Curran, H. M., and Miller, M., "Evaluation of Solar-Assisted Rankine Cycle Concept for the Cooling of Buildings," *Intersociety Energy Conversion Engineering Conference Record*, pp. 1391-1398, 1975.

Table 1. Numerical comparison of various Rankine/Brayton configurations

Case	Configuration description	Turbine work, Wh/kg	Compression work, Wh/kg	Net work output, Wh/kg	Heat Added, Wh/kg	Thermal efficiency, %	Specific steam consumption, kg/kWh	Fuel share in total heat input, %	Regeneration present
1	Simple Rankine 100% solar-powered (1-2-3-4-5-6-1 in Fig. 3)	111.6	0.02	111.6	773.2 (Solar)	14.43	8.96	0	No
2	Simple Brayton 100% fuel-powered (a-b-c-d-a in Fig. 3)	186.0	111.6	74.4	196.5 (Fuel)	37.86	13.44	100	No
3	Configuration 1 Combined simple Rankine/simple Brayton (Figs. 2, 3, & 4)	297.7	111.6	186.1	969.7 (773.2 Solar + 196.5 Fuel)	19.19	5.37	20.26	No
4	Configuration 2 Combined simple Rankine/regenerative Brayton (Figs. 5 & 6)	297.7	111.6	186.1	959.3 (773.2 Solar + 186.1 Fuel)	19.40	5.37	19.40	Yes
5	Configuration 3 Combined regenerative Rankine/ regenerative Brayton (Figs. 7, 8, & 9)	297.7	111.6	186.1	812.7 (626.6 Solar + 186.1 Fuel)	22.90	5.37	22.90	Yes

Table 2. Economic feasibility of various Rankine/Brayton Configurations

Case	Cost of energy added (1)	Cost of energy produced (2)	EFR* $\frac{(1)}{(2)}$
1	$100\,X$	$14.43\,Z$	5.54
2	$100\,Y$	$37.86\,Z$	1.06
3	$79.74\,X + 20.26\,Y$	$19.19\,Z$	3.75
4	$80.60\,X + 19.4\,Y$	$19.40\,Z$	3.72
5	$77.1\,X + 22.9\,Y$	$22.90\,Z$	3.09

*Calculated for $X = 4¢$, $Y = 2¢$ and $Z = 5¢$ per kWh.

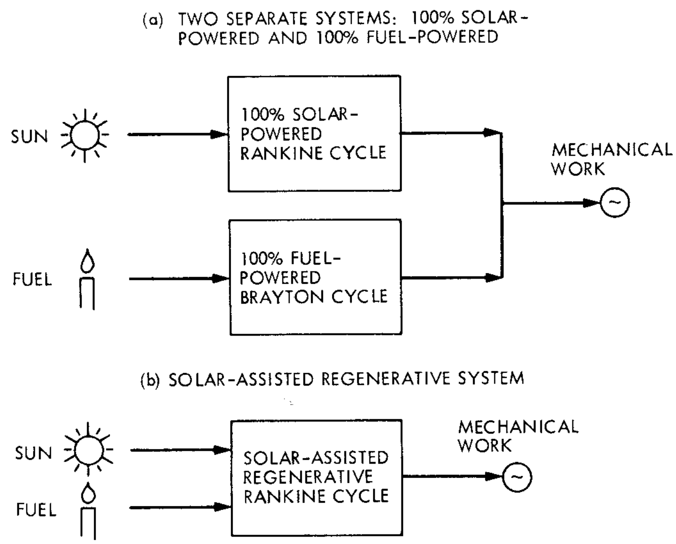

Fig. 1. Comparison between two systems

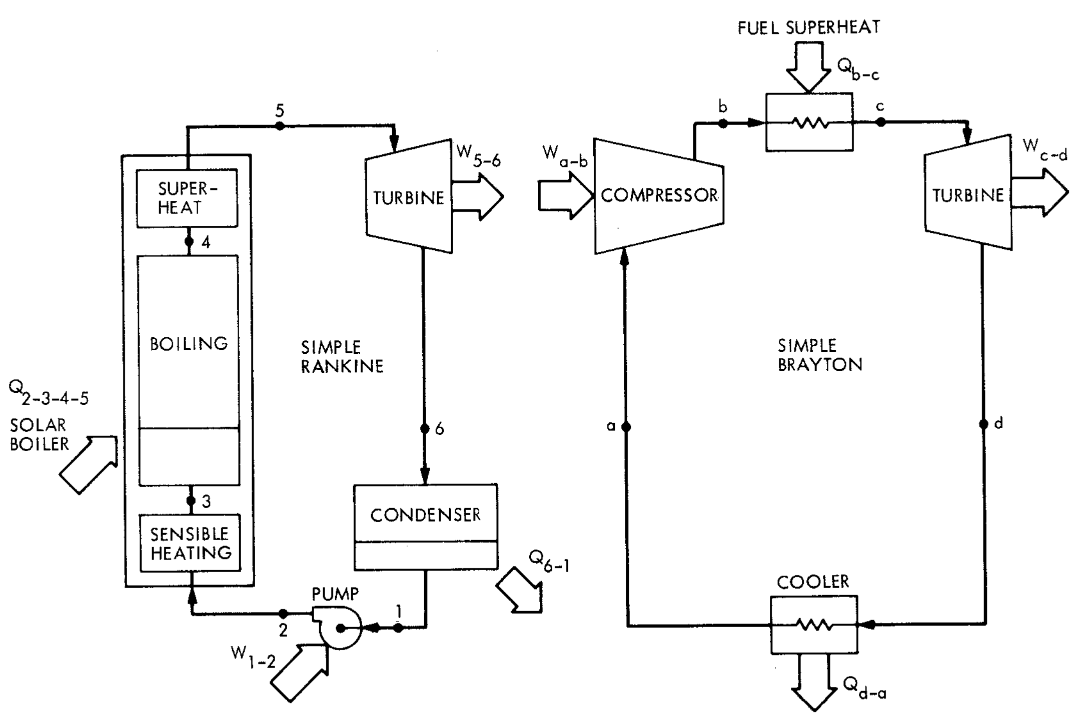

Fig. 2. Flow diagram of configuration 1: two physically separate cycles with common states 5 = b and 6 = a

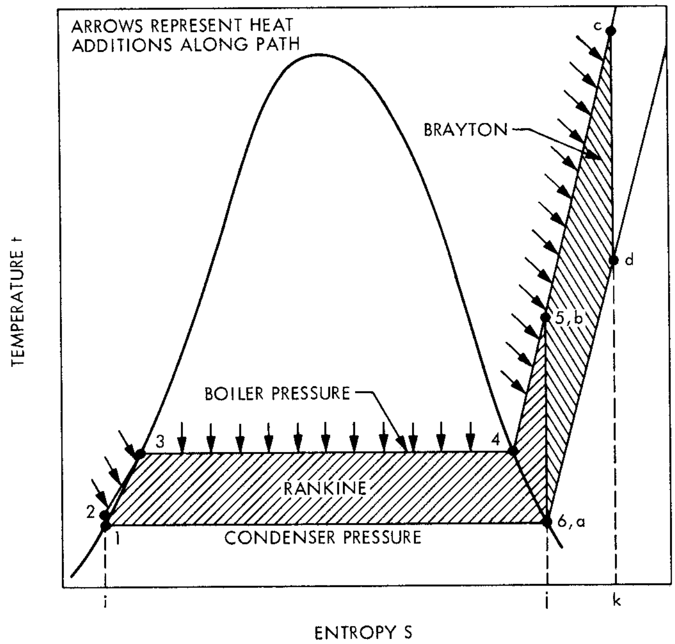

Fig. 3. Configuration 1 plotted on temperature-entropy diagram for water

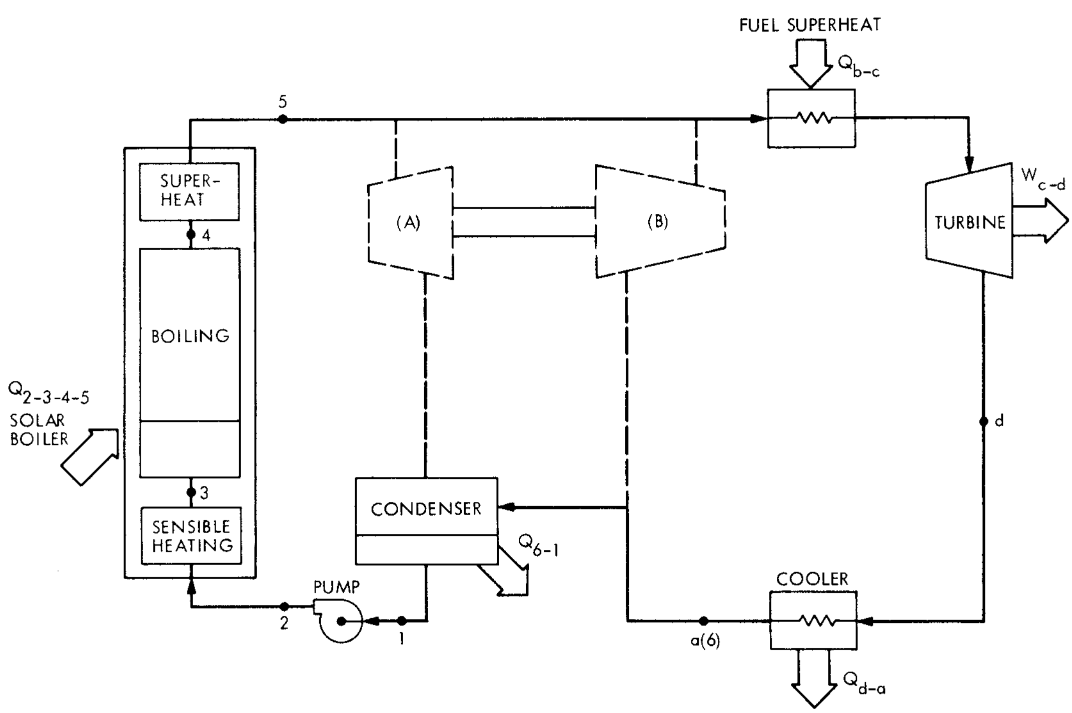

Fig. 4. Alternate flow diagram of configuration 1: deletion of Rankine turbine (A) and Brayton compressor (B) has no effect on the thermodynamics of the combined cycles

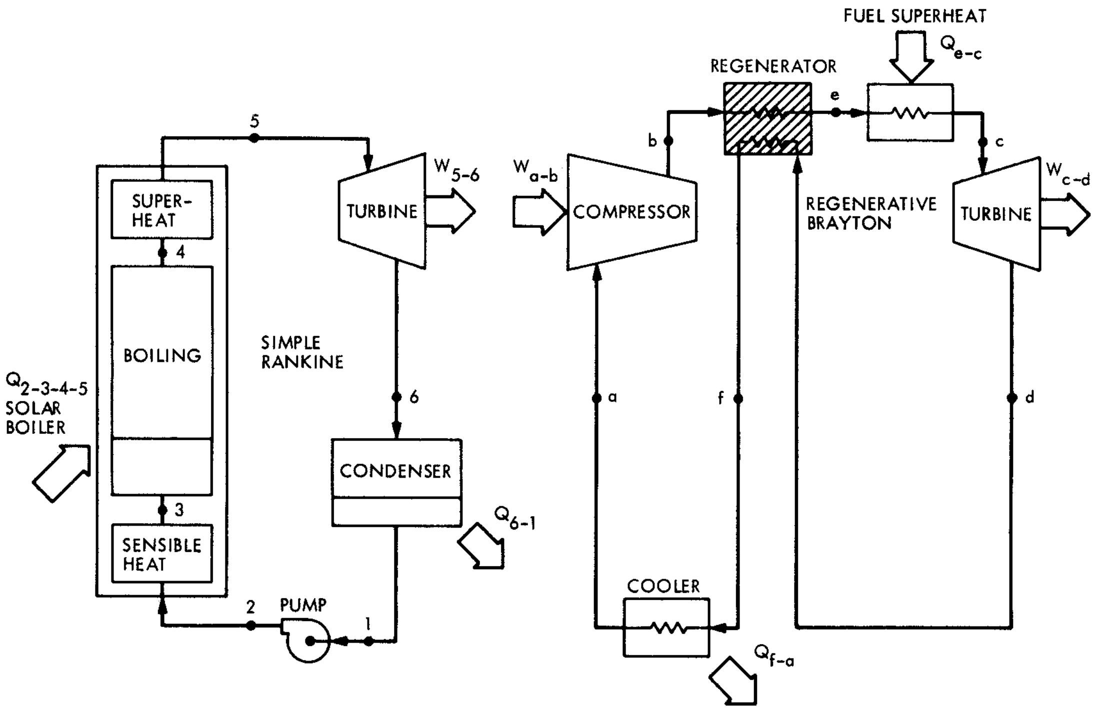

Fig. 5. Flow diagram of configuration 2

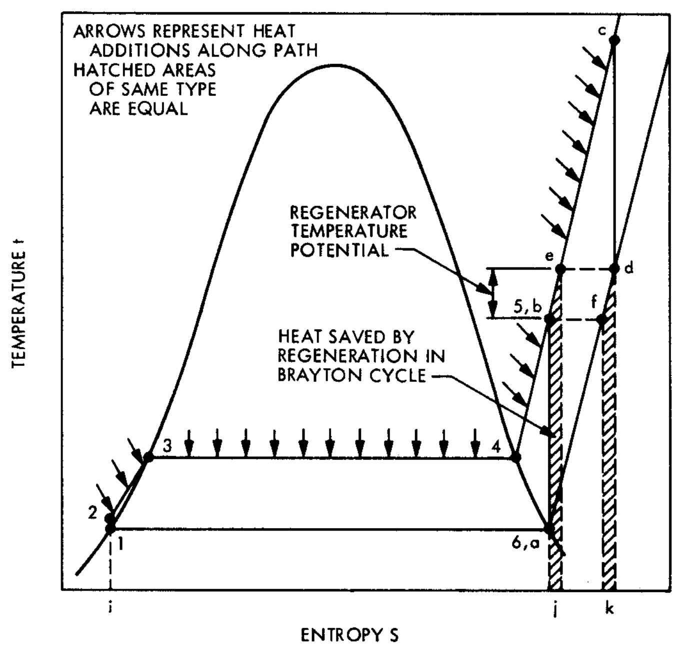

Fig. 6. Configuration 2 plotted on temperature-entropy diagram for water

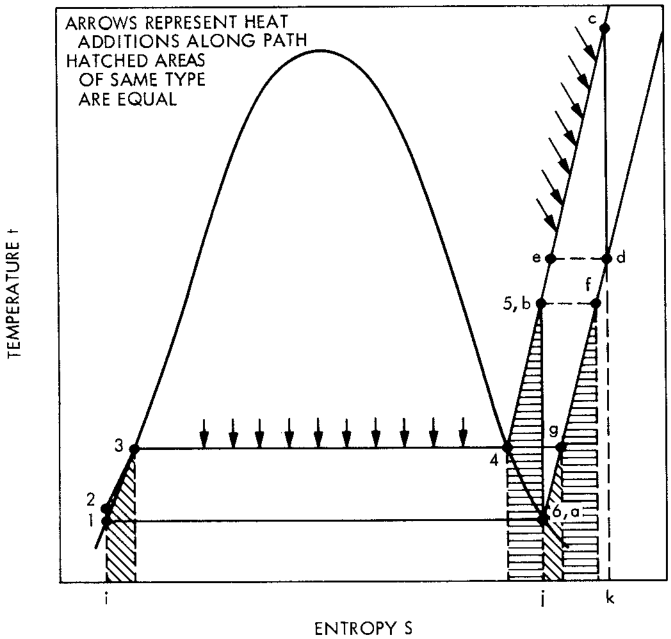

Fig. 7. Configuration 3 plotted on temperature-entropy diagram for water

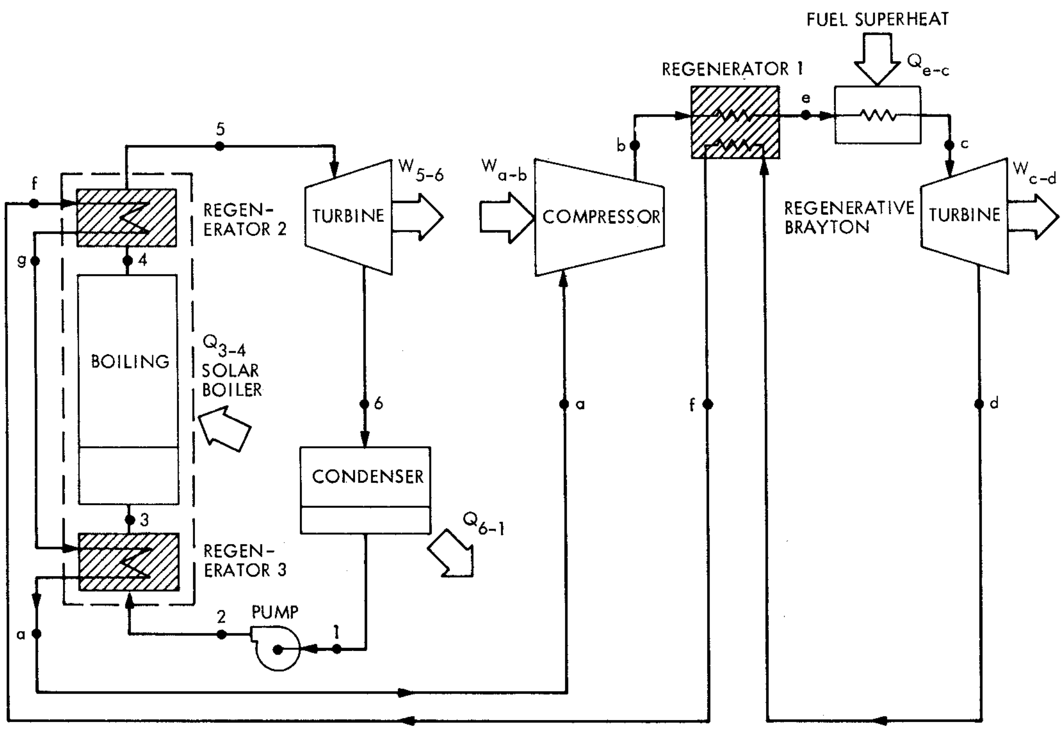

Fig. 8. Flow diagram of configuration 3

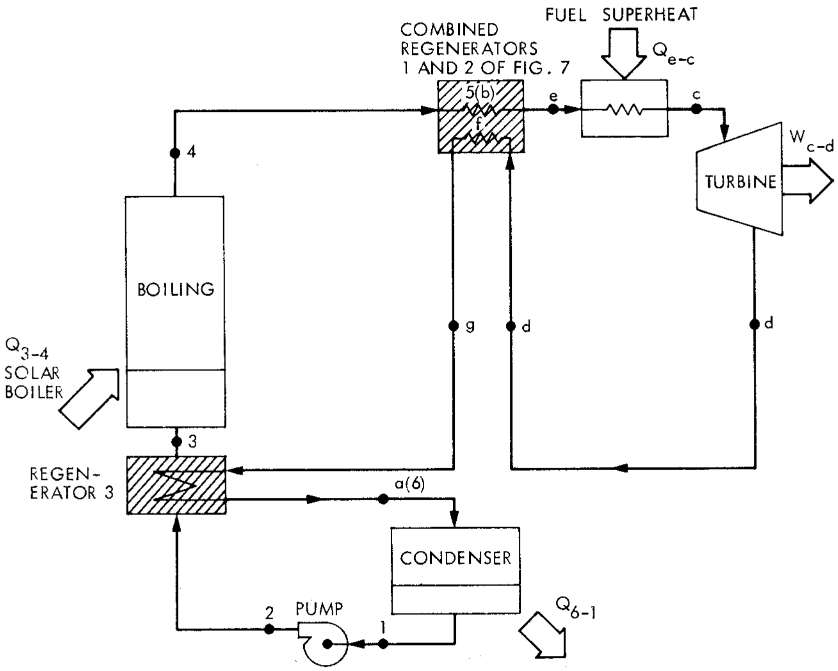

Fig. 9. Flow diagram of configuration 4: Rankine turbine/Brayton compressor deleted and regenerators 1 and 2 combined into one regenerator

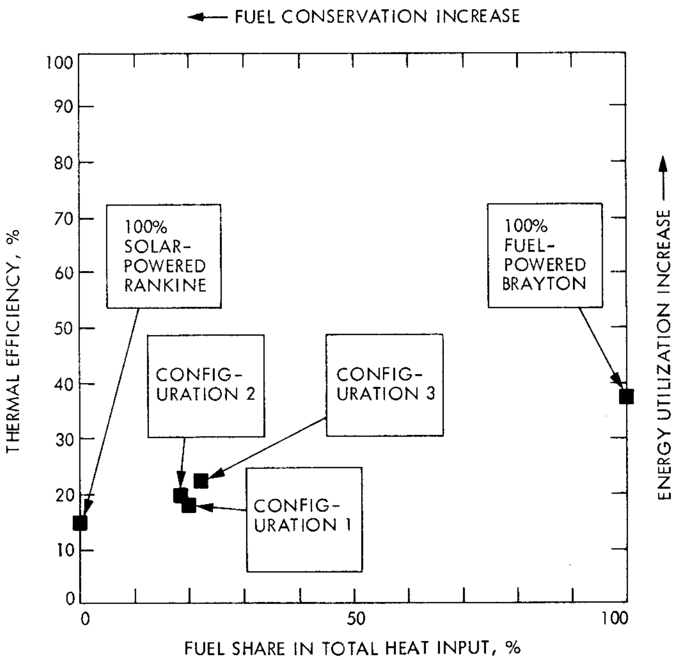

Fig. 10. Plot of thermal efficiency versus fuel share for various configurations in Table 1.