A Novel High-Temperature Ejector-Topping Power Cycle

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A novel, patented topping power cycle is described that takes its energy from a very high-temperature heat source and in which the temperature of the heat sink is still high enough to operate another, conventional power cycle. The top temperature heat source is used to evaporate a low saturation pressure liquid, which serves as the driving fluid for compressing the secondary fluid in an ejector. Due to the inherently simple construction of ejectors, they are well suited for operation at temperatures higher than those that can be used with gas turbines. The gases exiting from the ejector transfer heat to the lower temperature cycle, and are separated by condensing the primary fluid. The secondary gas is then used to drive a turbine. For a system using sodium as the primary fluid and helium as the secondary fluid, and using a bottoming Rankine steam cycle, the overall thermal efficiency can be at least 11 percent better than that of conventional steam Rankine cycles.

1 Introduction

To increase thermal power generation efficiency, there is a continuing effort to increase the cycle's top temperature. While the temperature of combustion is in excess of 1700°C, most of the power generated in the world is by steam Rankine cycles where the maximal steam temperature is about 560°C (1050°F) (Babcock and Wilcox, 1978), and gas turbines operate up to about 1200°C (2200°F). These temperature limitations are incurred due to the combined effects of temperature, pressure, and dynamic forces on the cost and life of the plant. The critical components are the turbine blades and bearings, and the superheater piping in steam cycles.

Another problem that limits the use of higher temperature (when coal is the fuel) in present-day power plants is ash deposition. The temperature in the superheater must be low enough so that the ash entrained by the burned gas will stay solid and would not stick to the tubes. On the other hand, at temperatures higher than those that can be used in current steam plants, above 870°C (1600°F) (Babcock and Wilcox, 1978) the molten ash has a low enough viscosity to flow off the tubes without accumulating. This phenomenon is indeed used in slag-tap and cyclone type furnaces to remove a part of the ash from the flue gases. Parts of such furnace walls, which are water cooled, are kept hot enough so that the ash deposited on them flows down into a slag tank. These types of boilers discharge less ash to the plant environment and need smaller ash separation equipment. If the temperature of the superheater convection tube-bank could also be high enough, the flue gases will be even cleaner and perhaps no particle emission control equipment would be needed.

The high combustion temperatures in regular or cyclone furnaces [in some cases over 1700°C (3000°F)] require a large and costly radiant space for cooling the burned gases to thermomechanically acceptable levels.

In summary, it is highly desirable to operate at higher temperatures that not only will improve efficiency but will also reduce the size and cost of furnaces and will help solve the problem of controlling flue gas ashes. Construction materials are available today, and better ones are continuously becoming available, which are sufficiently durable for extended periods of time at the higher temperatures required, if operated under low stress. Unfortunately, the high steam pressures in modern Rankine plants, which can reach 24 MPa (3500 psia), limit the possibility of raising the boiler temperature considerably. Blade, bearing, and dynamic problems also limit gas turbine operating temperatures.

Topping Rankine cycles were developed (but not on a commercial scale), which allow the use of high-temperature boilers at relatively low pressures. The topping cycle fluid (such as mercury) has a vapor pressure much lower than that of water, and can thus be boiled at temperatures closer to that of combustion (about 1500 K) but at pressures much lower than that of water. The vapor expands in a turbine and then condenses while transferring its heat to the water in the lower temperature Rankine cycle. The efficiency of such topping cycles is potentially indeed higher than that of conventional nontopping cycles, but the topping cycles still require a very high-temperature turbine, which is not available.

The novel cycle described here (Fig. 1) and patented (Freedman, 1989) is also a topping one, but instead of expanding the very hot gases in a turbine, it uses them to compress another gas in an ejector. Most noteworthy, the combustion gases, which are too hot to be used in a turbine, are cooled thereby to a level acceptable for use in present day turbines by a process that produces compression work of another gas. From the Second-Law viewpoint, the straight cooling of combustion gases from the combustion temperature to that acceptable for turbine operation, as practiced in conventional boilers, destroys completely the exergy contained between these two tem-

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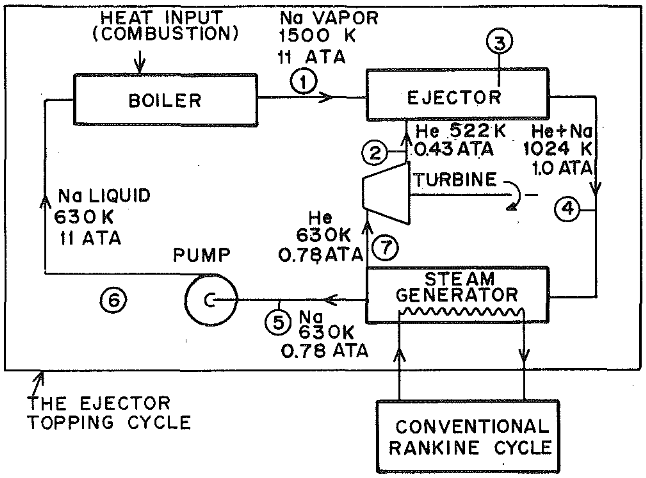

Fig. 1 Flow diagram of a high-temperature ejector-topping cycle

peratures. Here we accomplish the same cooling with concomitant production of useful work, clearly an important exergetic improvement. The ejector, as opposed to turbines, can operate at very high temperatures because of its inherently simple construction and absence of moving parts, which result in very low mechanical stresses and high reliability.

Although ejectors have relatively low efficiencies, the ejector-based topping cycles may have an overall higher efficiency than that of the above-described current turbine-based topping cycles, because of two major advantages: (1) The ejector can tolerate higher temperatures than a turbine can, and (2) it could use working fluids that have thermophysical properties superior to those that can be used in turbine topping cycles.

In the following, some aspects of the proposed system are discussed and an illustrative example is analyzed. The fluids chosen in this example were helium (the secondary fluid) and sodium (as the primary fluid). It is believed that helium is a good choice as a turbine fluid because of its low molecular weight and its high specific heat ratio, although it is harder to compress in the ejector. Sodium was chosen because there is experience in using it in high-temperature energy systems, and its thermophysical properties seem to be more promising than those of other candidate materials. It is possible that other working fluid pairs are better, and a thorough thermoeconomic selection study is desirable.

2 The Ejector

2.1 Introduction. The ejector uses energy from the sodium vapor to compress the helium from the pressure at the turbine outlet on back to the pressure of the turbine inlet (Fig. 1). In addition, it also supplies the pressure needed to overcome the pressure drop in the heating side of the steam generator. The compression is done by direct momentum exchange between the helium and the sodium through mixing and shear forces. Some important features of the ejector in the proposed system are described below. More information about ejectors in general can be found in Bonnington and King (1972), Bonnington and Hemmings (1976), and ASHRAE (1988).

2.2 Pressures. The mixing takes place in a convergent mixing section where the helium enters at subsonic velocity and the sodium at supersonic velocity. The mixing is done at a pressure that is lower than the pressure at the turbine outlet, and kept constant during the mixing.[2] There is an optimal mixing pressure for each situation that can be calculated using basic ejector theory. The pressure of the sodium vapor is reduced to the mixing pressure, which may be 30 times lower, by expanding the sodium vapor in a supersonic nozzle. After the mixing, the gas mixture is also supersonic and is compressed and slowed down, first by shock waves in a supersonic diffuser and then by expansion in a subsonic diffuser.

Increasing the pressure and temperature of the sodium increases the compression of the helium. The compression ratio of the ejector, i.e., the ratio of the helium pressures after and before the compression, is determined in an overall system analysis, and will be discussed below.

2.3 Condensation Processes. Sodium condensation may occur in the supersonic nozzle where its temperature may drop below the saturation pressure. This condensation will not be completed because of the tendency of the high-velocity gases to delay the condensation and to leave the vapor in a supercooled state (determining the fraction that does condense may need experimentation). If the amount of sodium condensing is too large, the heat released in the process may choke the flow and the ejector will not be able to compress the helium. If, on the other hand, only a small fraction of the sodium will condense, the ejector performance may even be slightly improved (ASHRAE, 1988). Condensation may also occur near the ejector outlet. In either case, this problem can be taken into consideration in the analysis, and averted to large extent in practice (say, by causing condensation to take place as near to the ejector outlet as possible).

2.4 Configuration. Many configurations of the ejector are possible, including the cylindrical and ring shapes shown in Fig. 2. Each of them contains the four basic elements: mixing section, supersonic nozzle(s), supersonic diffuser, and subsonic diffuser. The flow in the ejector can be axial or axial with rotation around the axis—"cyclone type." The cylindrical axial flow ejector is the only type that is well known and has a large theoretical and empirical data base. Cylindrical cyclone-type ejectors were described in the literature but their theoretical treatment is much more complicated and empirical data are scarce.

2.5 Ejector Computation. Exact computation of even the simplest kind of ejector needs sophisticated computer codes, which must be augmented by empirical correlations when condensation occurs during the process (cf. Mikkelson and Addy, 1974; VKI, 1975; Ortwerth, 1978; ESDU, 1986). In the analysis performed here, and based on the well-validated approaches and empirical data described in the above-mentioned references, the foundation was a one-dimensional inviscid approximation of adiabatic ideal gas flow. State-of-the-art empirical correction coefficients were used to obtain more realistic re-

[2] There are ejectors in which the pressure is not constant during the mixing.

Nomenclature

A = area
C_p = specific heat at constant pressure
C_v = specific heat at constant volume
H = enthalpy
k = specific heat ratio = C_p/C_v
m = mass flow rate
m_i = mass of species i
M = Mach number
P = pressure
T = temperature
V = volume
W = molecular weight
W_t = turbine work output
α = compressibility
ΔH = enthalpy difference
χ = molar mass ratio

Subscripts

a = air for combustion
g = gas
o = stagnation
tot = total

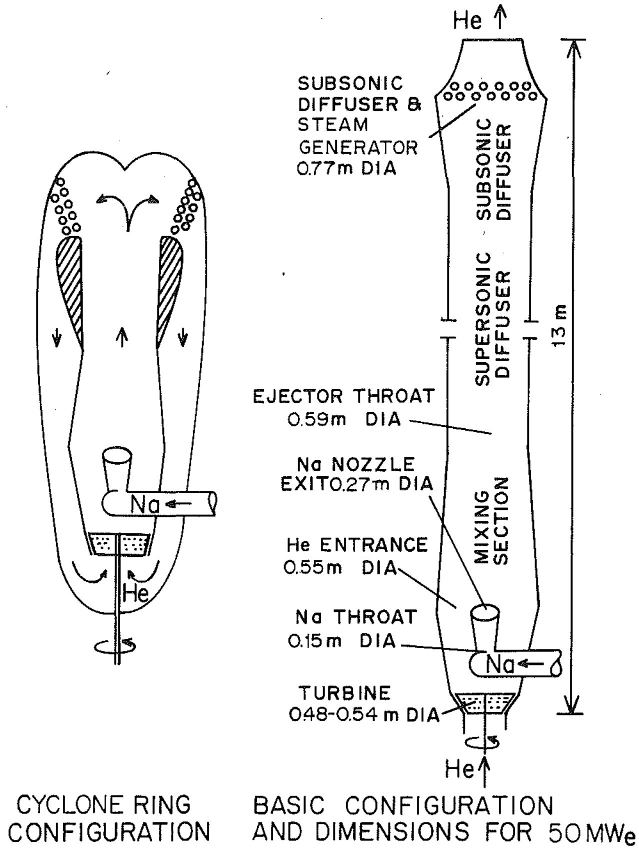

Fig. 2 Two ejector configurations

sults. The computer program developed solves a conventional model of supersonic constant pressure ejectors, which uses the laws of conservation of mass, momentum, and energy together with one-dimensional gas dynamic relationships between the flow parameters before and after a normal shock wave.

The following computational procedure was followed: (1) The parameters for the primary and secondary flows at the entrance are chosen, and the momentum of the flows is calculated, (2) it is assumed that there exists a downstream section in which the flow is totally mixed and has uniform properties, consistent with the assumption of a constant pressure duct, and the momentum there is computed by using the momentum conservation equation, (3) the temperature and mass flow rate of the mixed flow are computed using the conservation equations of mass and energy, (4) the pressure and velocity of the mixed flow are then computed, (5) if the mixed flow is supersonic, it is assumed that it decelerates through a normal shock wave and its exit pressure is thus found, (6) the pressure is then reduced by 25 percent to account for deviations from ideal ejector theory (see more below), checks are made to ensure that the entropy has not increased, and this whole procedure is repeated by systematic corrections in the inlet parameters until the exit pressure equals the desired one.

As stated above, a conservative empirical value of 25 percent stagnation pressure loss due to friction inside the ejector that is not accounted for by the theory was employed in the program. Condensation effects in the supersonic nozzle are assumed to reduce the mass of helium that can be compressed by 5 percent from the mass computed by the program.

The vapor pressure of the sodium needs to be known for calculating the equilibrium state of the mixture at the ejector outlet. This is given by Eq. (1) taken from Fink and Leibovitz (1981):

$$\ln P = 18.832 - 13113/T - (1.0948)\ln T + (1.977)10^{-4}T \quad (1)$$

where P = vapor pressure of the sodium, ATA, and T = temperature, K.

The total pressure P_{tot} is the sum of the partial pressures; according to Dalton's law it is related to the molar mass ratio by

$$\chi = \frac{m_{\text{Na}}}{m_{\text{He}}} = \left(\frac{P_{\text{Na}}}{P_{\text{tot}} - P_{\text{Na}}}\right) \frac{W_{\text{Na}}}{W_{\text{He}}}. \quad (2)$$

In cases where the mixture temperature is below the dew point, the latent heat of evaporation of sodium and the average specific heat capacity of the mixture need to be known for determining the equilibrium state. The calculation is done by guessing an equilibrium temperature, and calculating the sodium concentration at this temperature and the amount of sodium that needed to be condensed to reach this concentration. A check is made then to find whether the heat added by condensation increased the mixture temperature to the guessed one, and the initial guess is corrected if needed.

3 The Steam Generator

The steam generator produces the steam needed for the lower-temperature conventional Rankine cycle by heat transfer from the mixture of sodium and helium emerging from the ejector in a process that condenses the sodium and cools the helium and the condensed sodium. The pressures and temperatures of the steam are those used in boilers, superheaters, reheaters, and economizers of conventional power plants. The heat transfer rates in the steam generator are much higher than in fuel-fired boilers and are higher even than those found in liquid sodium steam generators, thus reducing significantly the required heat transfer area needed. These and other features of the steam generator are discussed below in more detail.

On the steam side the pressure is high and the heat transfer rates are similar to those found in conventional boilers. The main difference is in the coefficient of heat transfer on the gas side. This side is a partial condenser, where a large fraction of the heat transfer is by condensation of the sodium: In the example calculated in Section 6 below it is 75 percent of the total heat. The condensation heat transfer coefficient between pure sodium and steam generator tubes is two orders of magnitude larger than the heat transfer coefficient between combustion gas and the tubes. When the sodium vapor is mixed with the helium, the heat transfer coefficient of the mixture is between that of helium and that of condensing sodium. As a result, in parts of the steam generator where the sodium concentration is not too low, the gas side resistance to heat transfer can be neglected. In the steam generator section where water is evaporated, the heat transfer rate is determined, therefore, mainly by the thermal conductivity of the tubes. In the steam generator region where the sodium concentration is low, the heat transfer coefficient is that of the high-speed helium, which is still 10 to 20 times higher than that of combustion gases in furnaces. These high velocities are possible because the low density of the helium makes it possible to reach higher velocities than can be reached by air for a similar pressure drop. A part of the heat is transferred by the condensed (liquid) sodium, which also has a very high convective heat transfer coefficient. It should be noted that at the same time the tube design and materials must be such as to resist erosion by the sodium droplets.

The tubes in the steam generator are free from the ash deposition and the associated lowering of the heat transfer coefficients, which take place in fuel-fired systems.

The selection of the temperature of the gas leaving the steam generator and entering the turbine (station 7 in Fig. 1) is determined primarily by the need to condense and thus separate all the sodium. Higher temperatures not only allow undesirable condensation of sodium in the turbine, but also transfer more of the heat to the lower temperature Rankine cycle, and thus the overall system improvement, achieved by the topping cycle, is reduced.

It should be noted that the coexistence of sodium and water in the same heat exchanger (albeit separated by tube walls) requires special safety measures to prevent their direct contact. There is, however, significant experience with heat exchangers that use liquid sodium and water, particularly from the liquid metal fast breeder reactor programs in a number of countries, and both knowledge and safety are continuously being improved (Bliem et al., 1985).

Measures are needed to prevent sodium droplet entrainment by the helium before it enters the turbine. The droplets may damage the turbine and also increase the sodium concentration in the mixture passing through the turbine and compressed by the ejector, thus reducing the system efficiency. A two-step separation process is possible: first, cooling of the gas mixture to below the sodium dew point alongside with maximization of the contact between the mixture and the condensing liquid sodium, followed by inertial or electrostatic separation of the condensed sodium droplets from the mixture. The separation process produces, however, a pressure drop that would otherwise have been used to produce more power by the turbine, and thus a proper tradeoff between these two effects is needed. It may be noted that we have accounted for the incomplete sodium separation by assuming that 5 percent was not condensed (Section 6).

4 The Sodium Boiler

4.1 Furnace Features. The sodium is evaporated in a fossil fuel-fired furnace, which is operated at higher temperature than used for conventional steam boilers. To achieve the high temperature, combustion is proposed to be done in a cyclone furnace (although other furnaces are also possible), a small chamber in which a high-speed rotating air stream is used to burn the fuel in an almost adiabatic process. The air needed for the combustion is heated in a high-temperature air heater by exchanging heat with the burned gases leaving the furnace. The combustion temperature is determined by the temperature of the entering air, the quality of the fuel, the amount of excess air, and the amount of heat lost through the walls. Conventional cyclone furnaces operate at temperatures of about 2300°C, and can reach 3500°C and more if special high-temperature air heaters are used (Bliem et al., 1985).

It should be noted that operation at such temperatures tends to increase NO_x production, which could be controlled by catalytic methods for NO_x separation from the stack gas or by the use of advanced combustor concepts, such as the radiatively-conductively stabilized burner described by Tang et al. (1980, 1981). It should, however, be noted that the higher efficiency of the proposed system also results in a proportionally smaller quantity of fuel and combustion products, including CO_2.

The furnace walls are covered, on the fire side, by an insulating layer of molten ash, and its temperature is kept at the desired level by the boiling sodium on the other side. The hot gas leaving the cyclone chamber heats the main convection bank in which the sodium is evaporated. The combustion temperature, amount of excess air, temperatures of the burned gases entering the air heater and of the air leaving it, and other relevant parameters, can be determined from an overall heat balance, as shown in Section 6 below.

When compared to conventional fuel-fired steam boilers, the higher conductivity and the heat of evaporation of sodium, and the fact that the fraction of heat transferred in the boiling process is about 75 percent of the total heat exchanged for sodium and only about 50 percent for H_2O, it was found that the overall heat exchange area needed in the sodium furnace is ten to twenty times smaller than that needed to generate steam for the same amount of heat transferred (see Section 6 below). In addition, the specific heat of Na vapor is large (twice that of high temperature steam), and thus the Na mass flow rate is relatively small.

The lower pressure and density of the sodium vapor make the volume needed for its containment quite large. In summary, the sodium boiler is characterized by a small heat exchange area, low sodium mass flow rate, low pressure, high temperature, and high vapor volume.

4.2 Construction Materials. Even the newest metal alloys suffer from a relatively short operating life at these high temperatures, where creep has a very significant effect on reducing the allowable temperature and stress (cf. AGARD, 1966). Higher temperature operation can be attained by a combination of proper design and materials. An example of improved design is a fire-tube boiler in which the tensile loads on the hot surfaces of the tubes are replaced by compressive loads. Compressive yield strength is generally higher than tensile yield strength and the effect of creep is less significant. This can be accomplished by passing the hot combustion products through tubes that are immersed in liquid sodium. Based on knowledge of the properties of the high-temperature alloys that could be used, it is estimated that such a fire-tube boiler can operate at 1500 K (2240°F, which results in pressure of 1.08 MPa) for a number of years without failure (cf. AGARD, 1966).

Higher temperatures may be attained with nonmetals: graphite, ceramics, or composite materials. Graphite has excellent stability at high temperature and it does not creep. Its yield strength increases with the temperature and its value is approximately 5000 psi (33.3 MPa), high enough to be used for pressure tubes up to 10 MPa with reasonable wall thickness. Unfortunately, it is chemically attacked by the hot sodium and by oxygen. The only way it can be used is by isolating it from contact with the surrounding gases. Coating is a successful way to prevent oxidation of graphite at high temperatures (Hausner, 1966) and continuous improvements are being made in graphite coatings for uses such as re-entry vehicles and rocket nozzles. It might also be possible to isolate graphite tubes by metal alloy sheathes, which might retain isolation qualities despite the fact that they are plastic at those high temperatures.

Ceramic materials are continuously being improved for use in high-temperature energy systems, and impressive advances have been made in recent years (ASM, 1985). It seems reasonable that ceramic materials for the high-temperature, low-pressure boilers will be developed earlier than the materials for use in high-temperature turbines. There is a good chance that operation at 2000 K and higher would be possible in the near future by using new construction materials.

5 Pump and Turbine

The work required by the pump is very small compared to the work done by the turbine and it can be calculated using the compressibility (α) of sodium liquid. Knowing the compressibility and density of liquid sodium (6×10^{-3} K^{-1}, and 850 kg/m^3 at 600 K, respectively, Fink and Leibovitz, 1981) we can calculate the enthalpy difference of the liquid sodium before and after the pump, i.e., the work done by the pump assuming ideal isothermal compression:

$$\left(\frac{\partial H}{\partial P}\right)_T = V - T\left(\frac{\partial V}{\partial T}\right)_P = V(1 - T\alpha),$$

$$\Delta H = \int_{P1}^{P2} V(1 - T\alpha)dP = V(1 - T\alpha)(P_2 - P_1) = 0.42 \text{ kJ/kg}. \tag{3}$$

This is only about 0.1 percent of the work done by the turbine, as will be shown in the next section. Liquid sodium pumps, of several types, are described by Sittig (1956).

Table 1 Fluid parameters at the various stations in Fig. 1

Fluid		T K	P MPa	W	k	m kg/s	M	A cm²
1	Na, gas	1500	0.108	23	1.5	1.0	0.2	28
2	He	522	0.043	4	1.67	0.340	0.7	123
3	He + Na,g	871	0.10	10	1.605	1.34	0.4	121
4	Na;g + Na,l + He	1024	0.10	10	1.605	1.34	0.4	266
5	Na,l	630	0.08	23	----	1.0	----	
6	Na,l	630	1.08	23	----	1.0	----	
7	He	630	0.08	4	1.67	0.340	0.4	105

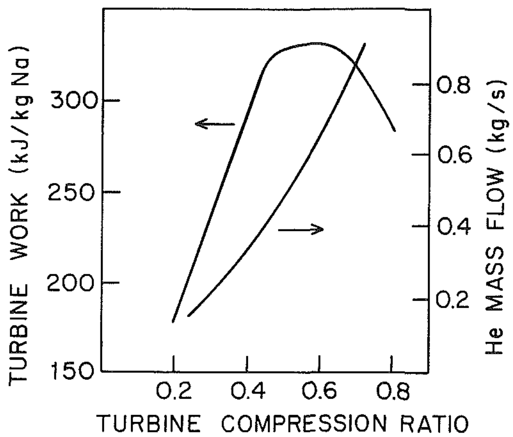

Fig. 3 Turbine work and helium mass flow rate as a function of ejector compression ratio for the example analyzed

Because the turbine is operated at a relatively low compression ratio, only a few stages will be needed along the flow direction and it thus should not be too large in that direction. The high specific volume requires, however, a relatively large cross-sectional area for the flow and hence a larger diameter.

6 Analysis of an Example

6.1 The Approach and Basic Assumptions. To provide quantitative performance predictions of this novel cycle, a full analysis of a conceptual system, using helium and sodium and producing 50 MWe, was performed employing for the ejector analysis the computer program described in section 2.5 above. The performance of the other components and of the entire system was calculated by employing mass and energy balances. The cycle and its main parameters are described in Fig. 1 and Table 1 (to simplify the description, the air preheater is not shown in Fig. 1). These describe a system that may be the closest to practical realization at this time, and the parameters have values that, although sometimes arbitrary, are used in the modeling of conventional systems. A conservative estimate of all losses was taken, and no optimization was performed yet. Consequently, it is quite likely that the real losses and the use of other combinations of parameters will yield better performance results. The results presented below were typically rounded to the nearest significant digit.

The first step in the analysis was to choose the compression ratio of the ejector. The compression ratio of the turbine is that of the ejector, minus the pressure drops in the steam generator and droplet separator. As can be seen from the results for this example shown in Fig. 3, a compression ratio of about 0.6 gives the maximal turbine work. Lower compression ratios require smaller mass flow rates of He (and thus result in smaller-size systems). We have chosen a compression ratio of 0.43, where the work output is reduced by only 3 percent from the maximal possible, but which allows a reduction of almost 50 percent in the helium mass flow rate from that needed for the maximal work output.

Table 2 Ejector parameters as computed by the computer program (entries marked with an asterisk are input parameters)

Parameters locations		m kg/s	Po MPa	P MPa	To K	T K	M·	W	k	A cm²
Before mixing start	Na	1.0 *	1.08 *	0.029	1500 *	451	3.048	23. *	1.5 *	36.2
	He	0.380	0.043 *	0.029	522 *	448	0.7	4. *	1.67 *	135.
At ejector outlet		1.38 *	0.1 *		871			10	1.605	
At sodium nozzle throat		1.0	1.08		1500		1.0	23.	1.5	9.6
At ejector throat		1.38	0.153	0.029	871		1.688	10	1.605	135

Using this information and the ejector-design computer program described in Section 2.5, the results shown in Table 2 were obtained. The sodium mass flow rate was chosen as 1 kg/s for simplicity.

The work done by the turbine, W_t, was computed by assuming isentropic expansion of an ideal gas. Gas turbines operating at similar pressure ratios attain isentropic efficiencies of up to about 95 percent (cf. Hill and Peterson, 1965), but since the operating pressure in this cycle is lower than in conventional gas turbines, the isentropic turbine efficiency in this example was assumed to be 80 percent.

Pressure losses in the steam generator and sodium droplet separator were assumed to be 22 percent of the compression ratio generated by the ejector. The turbine expansion ratio will be, therefore, 0.55.

It was assumed that the fuel is high-quality coal with an HHV of 32,000 kJ/kg, burned with 15 percent excess air, that the combustion efficiency of the sodium boiler is conservatively 0.82 and that the heat losses through the furnace walls are about 3 percent (Babcock and Wilcox, 1978).

To effect complete evaporation of the sodium in the boiler, it was assumed that the temperature of the combustion products leaving the sodium boiler is 50 K higher than the boiling temperature in the boiler; in our example it will be, therefore, 1550 K.

The ejector for such topping cycles is rather large, and the heat loss through its exterior surface was thus evaluated. The average temperature on the ejector inner wall is 700 K and the temperature difference across the wall is, therefore, 400 K, about half of the temperature difference in conventional steam boilers. It was assumed conservatively that the heat loss per unit area was approximately 0.4 kW/m^2 (250 Btu/ft^2/hr), typical to conventional boilers (Babcock and Wilcox, 1978). Based on the dimensions shown in Fig. 2, the surface heat loss is of the order of 10 kW, negligible relative to the 50 MW produced.

Consistent with the conservative approach used in the evaluation of the proposed cycle, it was assumed that its bottoming Rankine cycle has state-of-the-art high efficiency, including a supercritical boiler with feedwater heating (all internal heat sources that can be used in the cycle are used regeneratively) and no reheat. While reheat, which is usually used in conventional Rankine systems, would decrease the amount of steam produced, the overall cycle efficiency would almost not change at all.

6.2 Properties. The specific heat of He is practically constant in the entire operating temperature range, so $C_p = 5.2$ kJ/kg°C and $C_p/C_v = 1.67$ were used (Reynolds and Perkins, 1977; Rohsenow and Hartnett, 1973). The average specific heat of the combustion products (with 5 percent moisture) at the high-temperature side of the boiler was taken as 1.46 kJ/kg K

Table 3 Main results of the sample system analyzed

Power, kW/(kgNa/s):	
heat supplied by the coal	5730
helium-turbine work output	238
bottoming Rankine cycle work output	2124
air preheat available from the stack gas	3380
Mass flow rates, (kg/s)/(kgNa/s):	
the combustion products	3.24
the combustion air	3.06
bottoming cycle steam	2.2
The temperature of the combustion products, K:	
entering the sodium boiler, T_{g1}	2300
leaving the sodium boiling section	1550
leaving the sodium sensible-heating section in the boiler, T_{g3}	1290
leaving the stack	420
The temperature of the combustion air, K:	
ambient	300
entering the Na boiler furnace	1110
The conditions of the bottoming Rankine cycle:	
feedwater inlet to steam generator	400 K
steam exit	810 K, 24 MPa

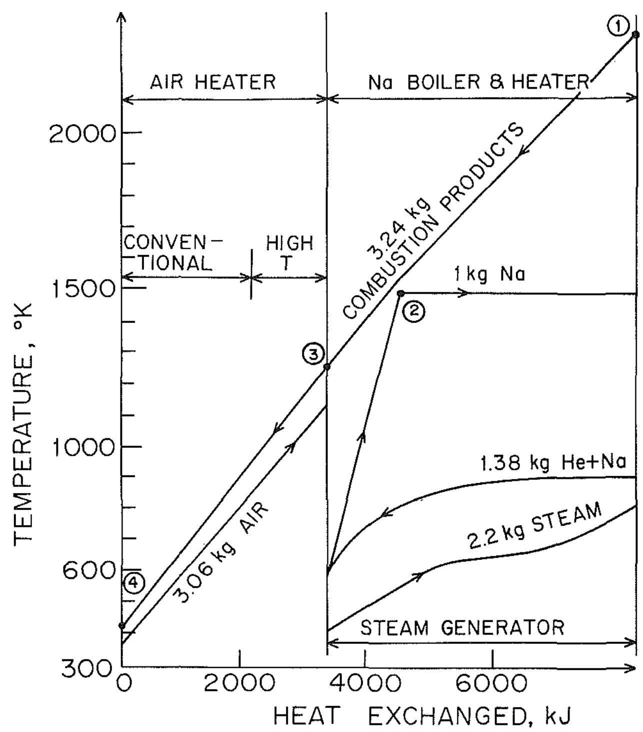

Fig. 4 Thermal balance of the system

and at the low-temperature end as 1.34 kJ/kg K (Babcock and Wilcox, 1978). The temperature dependence of the specific heat of air, both in the combustion air preheater and in the combustion chamber, were taken into account. The properties of sodium as a function of temperature were obtained from Fink and Leibovitz (1981).

6.3 Results and System Efficiency. The results of the analysis of this specific power cycle configuration are summarized in Table 3 and Fig. 4. Since the combustion air is preheated to 1110 K, a temperature higher than that in conventional Rankine power plants, it is noted that in the air-heater area and the air-heater blower, power would increase by about 30 percent. At the same time, as discussed in Section 4.1 above, the heat transfer rates in the proposed sodium boiler are much higher than those in steam boilers, resulting in a reduction of boiler air blower power consumption. This fact, and the relatively small energy consumption of the air blowers in the conventional case, lead to the conclusion that the air preheat in the proposed bottoming cycle would have an insignificant effect on the overall system efficiency improvement.

Table 4 Inefficiency sources and their values in the example analyzed; * indicates inclusion in the overall efficiency of the lower temperature

Inefficiency source	value	remarks
Pressure loss inside the ejector	25% of the work done in an ideal top cycle	literature-based estimate section 2.5
Pressure loss in the steam generator and droplet separator .	22% "	estimate
Decrease in helium mass flow rate due to condensation in primary nozzle and incomplete separation	10% "	estimate
Loss in possible turbine work due to less than optimum compression ratio used to reduce system size	3% "	see Section 6.1
Turbine efficiency	80%	"
Heat loss from ejector external surface	0.002% of the total heat supplied by the fuel	"
Combustion efficiency	82% *	"
Excess air needed for the combustion	15% of the stoichiometric amount *	"
Sodium boiler heat exchange efficiency	94%	Table 3
Air heater efficiency	80% *	"

Table 5 Features of different 50 MW sodium-helium ejector-topping cycles (the middle column, in bold letters, is the case fully worked out in the paper)

Boiler Pressure, MPa	0.2	1.08	**1.08**	5.0	5.0
Boiler Temperature, K	1250	1500	**1500**	2000	2000
Ejector Outlet Pressure, MPa	0.02	0.02	**0.1**	0.1	0.5
Efficiency improvement over 40% steam plant	6 %	9 %	**6 %**	11 %	8 %
Diameter of Ejector Throat, cm	128	160	**57**	74	29

Based on the calculated results, about 20 kg/s of sodium are needed for the production of the designated 50 MWe by the ejector-topped power plant, requiring 111,000 kW of fuel heat. Assuming that the bottoming steam cycle has an overall efficiency of 40 percent (this includes the power consumption by the preheat and combustion air blowers, water pumps, and other auxiliaries) and if no topping cycle is used, the work done by the steam turbine, W_s, with the same amount of heat input is therefore 44,400 kW.

If the topping cycle is used, the work done by the helium (topping cycle) turbine, W_{He} = 4764 KW, and the work done by the steam cycle turbine, W_{st}, is

$$W_{st}=(111{,}000-4{,}764)0.4=42{,}494. \tag{4}$$

The net power gain due to the addition of the ejector topping cycle is thus 4764 − (44,400 − 42,494) = 2858 kW, and the improvement relative to the conventional steam cycle without topping is thus 100 (2858/44,400) = 6.4 percent. The assumed and calculated system inefficiencies are summarized in Table 4.

Performance estimations under a variety of operating conditions shown in Table 5 have indicated possible efficiency improvements of up to 11 percent.

7 Conclusions

The concept of a novel high-temperature ejector-topping cycle was described. It was shown that with this method the efficiency of heat cycles can be improved by allowing the use of temperatures higher than those that current gas turbines can tolerate. This is made possible by using at these highest temperatures an ejector, which inherently has no moving parts, and by operating the entire topping cycle under relatively low pressures. Even with currently available construction materials, operating temperatures of over 1500 K can be reached, at pressures of about 1.1 MPa. Using sodium as the primary ejector fluid and helium as the secondary, an analysis of the cycle at these conditions indicated a 6.4 percent efficiency improvement over a conventional steam Rankine cycle without topping. Additional calculations indicated efficiency improvements up to 11 percent. No attempt was made as yet to optimize the fluids or operating conditions, and it is expected that even better efficiency improvements can be attained.

Thermodynamically, it is noteworthy that in this cycle the combustion gases, which are too hot to be used in a turbine, are cooled to a level acceptable for use in present day turbines by a process that produces compression work of another gas. From the Second-Law viewpoint, the straight cooling of combustion gases from the combustion temperature to that acceptable for turbine operation, as practiced in conventional boilers, completely destroys the exergy contained between these two temperatures. Here we accomplish the same cooling with concomitant production of useful work, clearly an important exergetic improvement.

Other improvements were noted in that the high-temperature operation allows the use of a much smaller furnace (the sodium boiler) with practically no ash accumulation or emission problems. This holds good promise for coal utilization.

This work is a description of the first effort to reduce the proposed concept to practice. It was shown that this cycle is technologically feasible, but its economic attractiveness still needs to be investigated. The exploration of other ways to utilize the concept and other working fluids is recommended and is indeed under way. One preferable research direction is the use of air and/or steam as the working fluids of the topping cycle, and there is a good chance that this direction will yield practical and improved systems.

References

AGARD, 1966, *Materials Properties Handbook, Vol. 4: Heat Resisting Alloys*, North Atlantic Treaty Organization Advisory Group for Aerospace Research and Development, Neuilli-sur-Seine, France.

ASHRAE, 1988, *Equipment Handbook*, American Society of Heating, Refrigerating, and Airconditioning Engineers, Atlanta, GA 30329.

ASM, 1985, *Engineering Application of Ceramic Materials—Source Book*, American Society for Metals, Metals Park, OH 44073.

Babcock and Wilcox, 1978, *Steam, Its Generation and Use*, Babcock & Wilcox, 161 E. 42nd St., New York, NY 10017.

Bliem, C., et al., 1985, *Ceramic Heat Exchanger Concept and Materials Technology*, Noyes Publications, Park Ridge, NJ.

Bonnington, S. T., and King, A. L., 1972, *Jet Pumps and Ejectors—A State of the Art Review Bibliography*, BHRA Fluid Engineering, Cranfield, Bedford, United Kingdom.

Bonnington, S. T., and Hemmings, J. A. G., 1976, *Jet Pumps and Ejectors*, BHRA Fluid Engineering, Cranfield, Bedford MK430AJ, United Kingdom.

ESDU, 1986, *Ejectors and Jet Pumps—Design and Performance for Compressible Air Flow*, Engineering Science Data Unit, Item No. 84029, 251–259 Regent Str., London W1R 7AD, United Kingdom.

Fink, J. K., and Leibowitz, L., 1981, "Calculation of Thermophysical Properties of Sodium," *Proc. Eighth ASME Symp. on Thermophysical Properties*, J. V. Sengers, ed., ASME, New York, Vol. 2.

Freedman, B.-Z., 1989, "Use of Ejectors for High Temperature Power Generation," U.S. Patent No. 4,843,823 (assigned to the University of Pennsylvania).

Hausner, H. H., 1966, *Coatings of High Temperature Materials*, Plenum Press, New York.

Hill, P. G., and Peterson, C. R., 1965, *Mechanics and Thermodynamics of Propulsion*, Addison-Wesley, Reading, MA.

Mikkelson, C. D., and Addy, A. L., 1974, "An Investigation of Gas Dynamic Flow Problems in Chemical Laser Systems," Report UILU-ENG-74-4009, Department of Mechanical and Industrial Engineering, Univ. of Illinois at Urbana.

Ortwerth, P. J., 1978, "On the Rational Design of Compressible Flow Ejectors," AIAA Paper No. 78-1217.

Reynolds, W. C., and Perkins, H. C., 1977, *Engineering Thermodynamics*, McGraw-Hill, New York.

Rohsenow, W. M., and Hartnett, J. P., 1973, *Handbook of Heat Transfer*, Ch. 2, McGraw-Hill, New York.

Sittig, M., 1956, *Sodium, Its Manufacture, Properties and Uses*, Reinhold Publishing Corp., New York.

Tang, H., Churchill, S. W., and Lior, N., 1980, "The Formation of Thermal and Fuel NO_x for Radiantly Stabilized Combustion," *Proc. Combustion Institute's 18th Int. Symp. on Combustion*, pp. 73–80.

Tang, H., Churchill, S. W., and Lior, N., 1981, "The Effect of Fuel of Fuel-Sulfur on NO_x Formation in a Refractory Burner," in: *Research Trends in Air Pollution Control: Scrubbing, Hot-Gas Clean-up, Sampling and Analysis*, R. Mahalingham and A. J. Engel, eds., AIChE Symposium Series, No. 211, Vol. 77, pp. 77–86.

VKI, 1975, *Ejectors*, VKI Lecture Series 79, von Karman Institute.