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LOW-TEMPERATURE THERMODYNAMIC BOTTOMING CYCLES FOR FUSION REACTORS

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Possible application of thermodynamic bottoming cycles to fusion reactors is examined. Thermodynamic and cost data for many possible working fluids are incomplete. Geothermal research is the primary source of fluid data. Bottoming cycles should be fully integrated into the energy conversion system to achieve maximum effectiveness. Scavenging by a bottoming cycle of low-level energy may be possible but not attractive with present conceptual fusion reactor designs. The best use of a bottoming cycle with a fusion reactor appears to be in conjunction with a helium turbine.

INTRODUCTION

The concept of a low-temperature thermodynamic bottoming cycle for power plants is not new, but it has received much less attention than a high-temperature thermodynamic topping cycle mainly because the topping cycles promise much greater improvements in thermal efficiency than do the bottoming cycles. Recent problems with thermal discharges from power plants have increased the interest in bottoming cycles because the temperature rise of the circulating water through the condensor may be reduced by employing a bottoming cycle. For example, a modern fossil-fused power plant achieves a plant efficiency of about 40%, a light-water reactor achieves about 33%, and a High-Temperature Gas-Cooled Reactor (HTGR) achieves about 40%. For a 1000 MW(e) power plant of each type, the fossil-fueled power plant rejects about 1250 MW(th) to cooling water plus 250 MW(th) out the stack; the light-water reactor rejects about 2030 MW(th) to cooling water; and the HTGR rejects about 1500 MW(th) to cooling water. This heat rejection may be reduced by a bottoming cycle, but there has been little analytical work done in the effect of using a bottoming with a conventional steam cycle. In this application, the bottoming cycle essentially replaces the low-pressure steam turbine. The relatively large specific volume of steam at low temperature makes low pressure turbine and condenser costs uneconomical at low heat sink temperatures. There was some interest in this approach several years ago until the reliability of large, low-pressure steam turbines was proven. However, there continues to be interest in the combination of a bottoming cycle coupled to a gas turbine power plant. It is predicted that a bottoming cycle would be able to raise plant efficiencies by almost 44% for a simple gas turbine cycle, and by over 28% for a recuperated gas turbine cycle.[1]

The thermal efficiency of conceptual fusion reactors is largely determined by choice of blanket structural material. Three general classes of blanket structural metals have been considered: (1) Metals with low activation cross sections which minimize radiological hazards [primarily aluminum and vanadium alloys]. (2) Metals presently extensively used in the nuclear industry which minimize industrial technology requirements [the stainless steels and nickel-based alloys]. (3) Metals which allow high blanket temperatures which maximize thermodynamic efficiency [the refractory metal alloys]. The refractory metals may allow blanket coolant outlet temperatures as high as 1100°C which allows the use of a potassium or cesium topping cycle coupled with a normal steam cycle. This energy conversion system may allow a thermal efficiency of 44.8%, and for a 1000 MW(e) plant, requires that only 1232 MW(th) be rejected. However, a fusion reactor with a stainless steel blanket and only a conventional steam cycle may have a thermal efficiency of 26.5% to 18.8%.[2] The efficiencies for a 1000 MW(e) plant require a rejection of 2744 MW(th) to 4376 MW(th). Present thermal pollution regulations have essentially eliminated inexpensive waste heat rejection methods for large power plants, and present conceptual fusion reactor designs generally call for large power plants. Heat-sink limitations for waste heat by the year 2000 may force the design of very efficient fusion reactors, and one method of increasing the efficiency of a power plant may be by the use of a low-temperature bottoming cycle.

Bottoming cycles may be more applicable to fusion reactor designs than to other power plant types because of heat sources not found in other types of power plants. For example, there are relatively large energy sources from magnet coil cooling systems, cryogenic refrigeration systems, and neutron shield cooling systems in magnetically confined fusion reactors (and from laser cooling in laser fusion reactors) which may allow energy recovery using a bottoming cycle. In this instance, the problem is similar to that in recovery of energy from geothermal sources - the source is more concentrated but still at a relatively low temperature. The geothermal energy program is investigating the economic recovery of energy from a 149°C (300°F) energy source;[3] a fusion reactor bottoming cycle using energy from fusion reactor auxiliaries in this temperature range may be desirable and should be considered.

BOTTOMING CYCLE FLUIDS

The high-efficiency (with topping cycle) energy conversion system has been proposed for the Reference Theta-Pinch Reactor (RTPR).[4] For the topping cycle the working fluid is a metal such as mercury, potassium or cesium. The relative merits of each material is still subject to debate, but at least there are only two main contenders, potassium and cesium.[5] The working fluid in the bottoming cycle is normally thought to be an organic compound. Selection of the optimum bottoming cycle fluid is a process confused by the large number of possible fluids, the incomplete thermodynamic data for many of these fluids (a substantial part of the geothermal energy program has been to mathematically model the thermodynamic behavior of many fluids for study), and the ability to partially tailor a bottoming cycle to take best advantage of a particular fluid. However, a bottoming cycle working fluid must possess most of the requisite properties for usefulness in a power plant. These properties determine the cost and safety of using a bottoming cycle. The fluid should be inexpensive, safe, durable and non-toxic to minimize problems with the fluid itself. The power plant design elicits other desirable properties. The fluid should be non-corrosive and compatible with plant materials. To minimize piping and pump sizes, the fluid should have a low liquid specific volume. A low vapor pressure is desirable for a low pressure system, but the condensation pressure should be slightly above atmospheric to avoid condenser vacuum pumping. To allow use of small and efficient heat exchangers, the working fluid should have a low latent heat of vaporization, a high specific heat at constant pressure, and a high film coefficient. The fluid should have a high decomposition temperature. The vapor specific volume should be high for a large turbine output but not so high that excessively high turbine blade speed, large turbines, and large piping sizes are necessary. The pressure of vaporization should be near the top of the dome of the T-s diagram to maximize superheat and cycle efficiency. The turbine design and fluid properties should be matched so that the vapor pressure leaving the turbine is slightly above saturated vapor; this allows an efficient turbine. A working fluid with a relatively low heat of vaporization near the top of the T-s dome maximizes the turbine pressure ratio (and boosts turbine output). An ideal working fluid for a bottoming cycle would possess all of these desirable qualities. Stability in a radiation environment may also be desirable.

Many possible bottoming cycle fluids may be rapidly eliminated because of obvious drawbacks; however, elimination of fluids for obvious reasons still leaves many fluids to be considered in detail. Milora and Tester[6] screened 19 organic compounds and then did detailed thermodynamic cycle calculations on the seven most promising for two geothermal applications - a 150°C liquid dominated resource and a 250°C hot rock resource. Whitbeck[3] considered 22 organic compounds for a 149°C geothermal heat resource. Madsen and Ingvarsson[7] investigated nine organic fluids and selected three for further study. Vrable and Schuster[8] considered 68 working fluids and analyzed five for a bottoming cycle with a maximum temperature of 199°C using the exhaust from a HTGR gas turbine as the heat source. The Westinghouse ECAS Study[9] considered 45 low boiling point fluids and analyzed four for use with gas turbine bottoming cycles. The General Electric ECAS Study[10] analyzed three low boiling point fluids used with gas turbine bottoming cycles. Some of the fluids considered are listed in Table I with comments from References 3, 6, 7, 8, 9, and 10 on their applicability to bottoming cycle use.

TABLE I. Possible Bottoming Cycle Fluids

Refrigerant Number and/or Name	Reference Number and Comments	
R-11, Carrene 2	3	Turbine and turbine exhaust large
	6	Competitive with NH_3 in performance, best performance with $T_{max} = 270°C$.
	7,9	No specific comments
R-12	3	Generally good fluid; pumpwork slightly high.

TABLE I. Possible Bottoming Cycle Fluids continued

Refrigerant Number and/ or Name	Reference Number and Comments	
R-12 (Con't)	6	Should perform well in 200°C range; best performance with T_{max} = 193°C.
	7	Chosen for more detailed analysis for a T_{max} = 143°C cycle.
	9	Analyzed for a bottoming cycle with T_{max} = 371°C.
R-13	9	No specific comments
R-13B1	6	Best performance with T_{max} = 151°C.
	9	No specific comments.
R-21	3	Turbine and exhaust large; low efficiency for T_{max} = 143°C.
	7,9	No specific comments.
R-22	3	Pumpwork and pressures high.
	6	Best fluid for use between 160-230°C; best performance with T_{max} = 202°C.
	7,9	No specific comments.
	10	Max allowable fluid temperature = 221°C.
R-31/114	3	About 5% lower performance than R-660a for T_{max} = 143°C.
R-32	3	Pumpwork high; pressures high.
	6	Best performance with T_{max} = 175°C.
R-40, Methyl Chloride	3	Good performance with T_{max} = 143°C, but toxic.
	9	No specific comments.
R-113	3	Turbine and pipe sizes large.
	6	Best performance with T_{max} = 284°C; comparable in performance to ammonia.
	7	Chosen for more detailed analysis for a T_{max} = 143°C cycle.
R-114	3	Turbine and pipe sizes large.

TABLE I. Possible Bottoming Cycle Fluids continued

Refrigerant Number and/ or Name	Reference Number and Comments	
R-114 (Con't)	6	Best performance with T_{max} = 196°C and in supercritical mode.
R-115	3	Pump work and pressures excessive.
	6	Best performance with T_{max} = 124°C.
R-31/114	3	About 5% lower performance than R-600a for T_{max} = 143°C.
G-133	9	No specific comments.
R-142b	3	Low efficiency for T_{max} = 143°C; otherwise, good.
	6	Best performance with T_{max} = 200°C.
	9	No specific comments.
R-152a	3	Low efficiency for T_{max} = 143°C; otherwise, good.
	6	Best performance with T_{max} = 192°C and should perform well in 200°C range.
	9	No specific comments.
R-216	3	Large turbine required.
	6	Best performance with T_{max} = 228°C.
R-260, Propane	3	High pump work and pressures; low efficiency for T_{max} = 143°C.
	6	Best performance with T_{max} = 179°C, but should perform well in 200°C range.
	8,9	No specific comments.
RC-318	3	Large turbine and exhaust required.
	6	Best performance with T_{max} = 149°C.
	8	Chlorine free.
R-500, Carrene-7	3	High boiler pressure; high pump work.
	6	Best performance with T_{max} = 188°C, but should perform well in 200°C range.
	7	No specific comments.

TABLE I. Possible Bottoming Cycle Fluids continued

Refrigerant Number and/ or Name	Reference Number and Comments	
R-504	3	Excessive pump work.
	6	Best performance with T_{max} = 161°C; may be best fluid at about 150°C.
R-500, n-Butane	3	Generally good for T_{max} = 143°C (second best to R-600a).
	7	High flammability.
R-600a, Isobutane	3	Generally good for T_{max} = 143°C.
	6	Best performance with T_{max} = 198°C in a supercritical mode.
	7	Chosen for more detailed analysis in a T_{max} = 143°C cycle; high flammability and very high toxicity.
	8	Highest efficiency fluid analyzed for T_{max} = 199°C; higher than NH_3.
	9	No specific comments.
R-717, Ammonia	3	High pressure.
	6	Best performance with T_{max} = 295°C; superior to other fluids at over 250°C; best performance with a subcritical cycle.
	8	Chosen as best overall fluid for T_{max} = 227°C; should yield lowest capital costs of fluids considered (R-717, R-12, R-260, R-600a and R-1270).
	9	No specific comments.
R-718, Water	9	No specific comments.
	10	Used for bottom cycle with T_{max} = 538°C.
R-1270, Propylene	3	Excessive pump work.
	6	May perform well in 200°C range; best permance with T_{max} = 177°C.
	8,9	No specific comments.
C-15-12	8	Chlorine free.
Acetaldehyde	9	No specific comments.

TABLE I. Possible Bottoming Cycle Fluids continued

Refrigerant Number and/ or Name	Reference Number and Comments	
1-Butene	9	No specific comments.
2-Butene(cis)	9	No specific comments.
2-Butene(trans)	9	No specific comments.
1-Butyne	9	No specific comments.
2-Butyne	9	No specific comments.
Carbonyl Sulfide	9	No specific comments.
COS	9	No specific comments.
Cyclobutane	9	No specific comments.
Dibromodi-fluoromethane	9	No specific comments.
Dimethylamine	9	No specific comments.
Ethyl Chloride	9	No specific comments.
Ethyl Fluoride	9	No specific comments.
Ethylamine	9	No specific comments.
Ethylene-Fluoride	9	No specific comments.
Ethylene Oxide	9	No specific comments.
Fluorinol-85	10	Best performance with T_{max} between 316-399°C.
Hydrogen Sulfide	9	No specific comments.
Methanethiol	9	No specific comments.
Methyl Bromide	9	No specific comments.
Methl Ether	9	No specific comments.
Methylamine	9	Low turbine exhaust volume; Analyzed cycle T_{max} = 510°C
N_2O	9	No specific comments.
Octafluropropane	9	No specific comments.
Propadiene	9	No specific comments.
Propyl Fluoride	9	No specific comments.
SO_2	9	Excellent performance for T_{max} = 538°C
Trimethylamine	9	No specific comments.

Table I contains a listing of 59 possible bottoming cycle fluids. This listing is probably far from complete, and it evident that of those fluids listed, few have been subjected to more than a very preliminary analysis. While some fluids appear to be somewhat better for some applications, it is very probable that an optimum bottoming cycle working fluid has yet to be considered. Furthermore, except for the more commonly used refrigerants, working fluid costs tend to be very speculative for large quantities.

BOTTOMING CYCLE THERMODYNAMICS

The primary function of a bottoming cycle is to extract useful work from otherwise wasted energy ejected from a primary thermodynamic cycle. Therefore, a bottoming cycle is a scavenging cycle by nature. It must be designed to maximize the total energy conversion efficiency and output rather than maximize its own efficiency. For example, the thermodynamic efficiency of the primary cycle (η_p) is the ratio of useful output (P_p) to input energy (Q_p):

$$\eta_p = \frac{P_p}{Q_p} = \frac{Q_p - R_p}{Q_p} \qquad (1)$$

where R_p is the energy rejected by the primary cycle. The bottoming cycle then extracts a fraction (f) of R_p, turns it into useful energy (P_b), and rejects energy (R_b). The efficiency of the bottoming cycle η_b is:

$$\eta_b = \frac{P_b}{fR_p} = \frac{fR_p - R_b}{fR_p} \quad . \qquad (2)$$

Therefore, the overall thermodynamic efficiency of the energy conversion process (η) is:

$$\eta = \frac{P_p + P_b}{Q_p} \quad ; \qquad (3)$$

$$\eta = \frac{P_p}{Q_p} + \frac{P_b}{Q_p} = \eta_p + \frac{fR_p - R_b}{Q_p} \quad \frac{fR_p}{fR_p}$$

$$\eta = \eta_p + \eta_b f(1 - \eta_p) \quad . \qquad (4)$$

Equation (4) illustrates that if a bottoming cycle can be used to efficiently exploit the energy in the primary cycle flow stream, it is not always necessary to use means to improve the primary cycle efficiency to improve the overall thermodynamic efficiency (η). Equation (4) also illustrates that one should maximize $\eta_b f$ rather than just η_b. f is maximized by rejecting heat from the primary cycle at the lowest possible temperaure, and this temperature is limited by the heat exchanger design.[10] Since η_b generally decreases with a reduction in maximum temperature of the bottoming cycle, a value of $\eta_b f$ may be optimized for a given system.

A bottoming cycle for fusion reactor may result in an f effectively greater than 1. The energy extracted from the primary cycle of a fusion reactor by a bottoming cycle is $f_1 R_p$, and if energy (Q_s) is scavenged from other sources in a fusion reactor system, then the energy available to the bottoming cycle (fR_p) is:

$$fR_p = f_1 R_p + Q_s \quad ; \qquad (5)$$

$$\therefore f = f_1 + \frac{Q_s}{R_p} = f_1 + \frac{Q_s}{Q_p}\frac{1}{1 - \eta_p}$$

$$f = f_1 + \frac{Q_s}{Q_p}\frac{1}{1 - \eta_p} \quad .$$

If the ratio Q_s/Q_p is the scavenging parameter(s), then

$$f = f_1 + \frac{s}{1 - \eta_p} \quad . \qquad (6)$$

FUSION REACTOR APPLICATIONS

As indicated in the previous section, to achieve the maximum effectiveness a bottoming cycle should not simply be a replacement for the low-pressure steam turbine, but should require an extensive reappraisal of the entire energy conversion system. The design of a fusion power plant must provide the means to exploit the ability of a bottoming cycle fluid to scavenge low-temperature heat energy that is useless for a steam cycle. Some possible sources of energy are fluid pump cooling systems, heat rejection from refrigeration systems, neutral-beam injectors, direct energy convertors, vacuum pump cooling systems, and energy deposited in the magnet coils or coil shielding. Careful study is required to determine whether or not an energy-scavenging bottoming cycle is worth the effort. For example, the present blanket design of the Referenced Theta-Pinch Reactor (RTPR) results in a deposition of 850 MW(th) into the magnet coils and structure. This energy can be utilized only by increasing the magnet coil and structure temperature from 25°C to 150-200°C, but there are problems with this approach. The magnet coil and support structure are already highly stressed, and this increase in temperature would degrade the material properties by about 10%. The electrical resistivity of copper increases approximately 0.39% for every 1°C rise in temperature, and this would result in a 50-60% increase in coil resistance. An increase in temperature would also change the neutronic characteristics of the coil and structure. These problems may be resolved by further design and by the development of an inexpensive high-capacity energy supply, such as the Homopolar Energy Transfer System (HETS).[4]

As references 1, 8, 9 and 10 indicate, the most promising use of a thermodynamic bottoming cycle may be to recover waste heat from a gas-turbine exhaust. If helium is ultimately selected as the best fusion-reactor blanket heat removal medium, a directly coupled helium-gas turbine and a bottoming cycle appear to be an attractive energy conversion scheme. Vrable and Schuster[8] predict that a HTGR with a gas-turbine ammonia bottoming cycle with a wet cooling tower for waste heat rejection offers a 24% increase in plant output over a HTGR gas turbine with a dry cooling tower. Similar results could be expected from a fusion reactor helium turbine application.

The outlook is cloudy for the replacement of the low-pressure steam turbine with a bottoming cycle to improve the efficiency of this part of the energy conversion system. The Liquid Metal Fast Breeder Reactor (LMFBR) low-pressure turbine[11] will be approximately 18.6% efficient with an upper temperature of 173°C and a condensate temperature of 33.2°C. The ammonia bottoming cycle used with the HTGR gas turbine is 19.7% efficient with an upper temperature of 227°C and a condensate temperature of 35.4°C. The ammonia cycle uses 25.7°C cooling water; this high temperature was chosen to facilitate a performance comparison with a HTGR power plant with a dry cooling tower. The LMFBR uses 13.9°C cooling water. The design of steam system components is a well-established technology; whereas, the design of ammonia-vapor cycle components is primitive. Table II lists some differences between the LMFBR steam condenser and the HTGR ammonia condenser. Each condenser is single pass with shell-and-tube construction. The major difference between the two condensers is the operating pressure. Ammonia condensers operate at well above atmospheric pressure which eliminates the vacuum equipment necessary for a steam condenser. The exterior

TABLE II. Ammonia Vapor and Steam Condenser Characteristics

	Ammonia[8]	Steam[11]
Stated Parameters		
Heat Load, MW(th)	1677	755
Heat Transfer Surface Area, m^2	2.145×10^5	2.016×10^4
Avg. Condensate Temp., °K	308.4	306.2
Cooling Water Flow, kg/s	7.235×10^4	2.187×10^4
Cooling Water Inlet Temp., °K	298.7	286.9
Cooling Water Outlet Temp., °K	307.0	295.1
Condensate Pressure, MPa	1.39	0.005
Calculated Parameters		
Avg. Cond.-Cooling Water ΔT, °K	5.55	15.2
Overall Heat Transfer Coefficient, W/m^2-K°	1.408×10^3	2.464×10^3
Cooling Water Flow/Heat Load, Kg/s-W	4.314×10^{-5}	2.897×10^{-5}

shell of the ammonia condenser must be designed to accommodate the higher and reversed pressure stresses; however, the tube diameter and wall thickness can remain approximately the same as that for a steam condenser because the tube wall is loaded in compression rather than in tension. Note in Table II that the calculated temperature difference between the condensate and cooling water of the ammonia condenser is about three times greater for the steam condenser, but the overall heat transfer coefficient of the steam condenser is about twice that of the ammonia condenser. The heat transfer properties of saturated ammonia and water are not greatly different (except for the Prandtl Number), and the overall heat transfer coefficient of the two condensers should be similar. A more sophisticated design should raise the overall heat transfer coefficient of the ammonia condenser to near that of the steam condenser. This should result in the cooling water flow for the ammonia condenser being similar to that required for the steam condenser; therefore, it seems possible that the use of an ammonia bottoming cycle may reduce the waste heat problem.

The cost of a bottoming cycle for fusion reactors may be gauged from the estimates for the geothermal energy program. The estimated capital cost ranges from \$300 to \$700/kW, and the cost of electricity produced from geothermal energy would cost 1.56¢ to 4.30¢/kWh.[6] In contrast, the UMAK-1 capital cost is estimated at \$900 to \$1000/kW, and the electricity cost is 0.02¢/kWh.[12] The numbers are rather speculative in nature and contrast to about 1¢/kWh for coal, 2¢/kWh for oil, and 0.24-0.30¢/kWh for nuclear-produced electricity in 1974.[6]

CONCLUSIONS

Low-temperature bottoming cycles may have a place in the development of fusion technology, but preliminary scoping studies are hampered by a lack of basic thermodynamic data necessary to make more than a qualitative view. The geothermal energy program will likely produce the requisite thermodynamic data, cost data and calculational techniques to do a quantitative assessment in the future. To be effective, the bottoming cycle must be an integral part of the energy conversion system, not just a substitution or add-on cycle. If the bottoming cycle fluid can be used to scavenge heat in fusion reactor systems, the energy conversion system efficiency can be increased; however, present conceptual fusion-reactor designs do not lend themselves to effective energy scavenging by low-temperature bottoming cycles.

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