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Evaluation of an Absorption Heat Pump to Mitigate Plant Capacity Reduction Due to Ambient Temperature Rise for an Air-Cooled Ammonia and Water Cycle

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Evaluation of an Absorption Heat Pump to Mitigate Plant Capacity Reduction Due to Ambient Temperature Rise for an Air-Cooled Ammonia and Water Cycle

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Abstract

Air-cooled geothermal plants suffer substantial decreases in generating capacity at increased ambient temperatures. As the ambient temperature rises by 50°F above a design value of 50°F, at low brine-resource temperatures, the decrease in generating capacity can be more than 50%. This decrease is caused primarily by increased condenser pressure. The use of mixed-working fluids has recently drawn considerable attention for use in power cycles. Such cycles are more readily amenable to use of absorption "heat pumps." For a system that uses ammonia and water as the mixed-working fluid, this paper evaluates the use of an absorption heat pump to reduce condenser backpressure. At high ambient temperatures, part of the turbine exhaust vapor is absorbed into a circulating mixed stream in an absorber in series with the main condenser. This stream is pumped up to a higher pressure and heated to strip the excess vapor, which is recondensed using an additional air-cooled condenser. The operating conditions are chosen to reconstitute this condensate back to the same concentration as drawn from the original system. We analyzed two power plants of nominal 1-megawatt (MW) capacity. The design resource temperatures were 250°F and 300°F. Ambient temperature was allowed to rise from a design value of 50°F to 100°F. The analyses indicate that use of an absorption heat pump is feasible; excess brine can be used to reduce condenser backpressure. For the 300°F resource, an increased brine flow of 30% resulted in an increase in net power of 21%. For the 250°F resource, the increase was smaller. However, these results are highly plant and equipment specific because evaluations must be carried out at off-design conditions for the condenser. Such studies should be carried out for specific power plants that suffer most from increased ambient temperatures.

Introduction

To promote wider deployment of geothermal energy, the U.S. Department of Energy (DOE) is pursuing development of medium- and lower-temperature applications for power generation. The brine temperature for a low-end resource may be as low as 250°F and that for a medium resource about 300°F. At these temperatures, to increase the power yield from the resource, use of non-azeotropic mixed-working fluids, such as ammonia and water, are being investigated [1].

Power plants that use an air-cooled condenser (ACC) and a low-temperature brine resource suffer substantial loss in capacity with increasing ambient temperatures. This loss arises primarily from an increased backpressure in the condenser because of the condenser's decreased capacity to reject heat at higher temperatures. This paper examines the influence of increased ambient temperature on power systems with resource brine temperatures of 300°F and 250°F. We evaluated a cycle that uses a mixture of ammonia and water as the working fluid.

Background

To mitigate the influence of ambient temperature swings, many methods are feasible. The basic problem boils down to that of supply-side adjustments to the power system in response to the heat-rejection environment. Many known methods for energy storage, such as electrical, pumped hydro, thermal, and chemical are applicable here. However, for geothermal systems, because a thermal reservoir is an integral part, we evaluated its use as the first-cut option. We duly note that this is one of many possible methods.

For the evaluation, we chose to look into a cycle that uses a mixture of ammonia and water as the working fluid. While DOE's geothermal program has paid substantial attention to the Kalina family of cycles, no such cycle is currently in operation in the United States. Deployment of Kalina plants in the United States is in planning stages. Two such systems have been built in Japan and Iceland [1]. However, details of their performance are not available in the literature. Much of the cycle information remains proprietary. On account of this, we chose to evaluate the Maloney-Robertson (MR) cycle [2] for this study. For geothermal systems, the MR cycle with an ammonia and water mixture yields good performance at low and medium brine temperatures. The results and conclusions drawn from this study are applicable, with minor differences in values, in principle to many cycles that use mixed-working fluids.

Assumptions

For reported analyses, we made the following assumptions: two resource temperatures of 300°F and 250°F (on the low end of the resource temperature), and air-cooled power-plant condenser(s). The design ambient temperature was set at 50°F. For performance evaluation at higher temperatures, we chose an elevated ambient temperature of 100°F.

Assumptions regarding the various components of the cycle are:
1) All heat exchangers are assumed to operate at an internal pinch point of 5°F.
2) The pumping causes a pressure rise of 120 pounds per square inch absolute (psia) for the brine.
3) The air-side pressure loss for the condenser is set at 0.5 in. of water.
4) All liquid pumps and air fans possess an overall efficiency of 80%.
5) Isentropic efficiency for the turbine is set at 86% and the generator efficiency at 92%.
6) Pressure losses in the working fluid loop are ignored.

Many of these assumptions are optimistic. We note that a 5°F pinch for the heat exchangers is a demanding specification. When a heat exchanger is designed to that specification, whether one can assert its performance to that specification in the field remains unproven. Routinely used field instrumentation is not capable of yielding results that can verify this performance within an acceptable engineering uncertainty. Furthermore, a vaporizer that can yield arbitrary quality for the exiting feed for mixed-working fluids at a theoretical minimum brine flow requirement remains to be demonstrated. This study also assumes the use of countercurrent flow in the air-cooled components. Such heat exchangers are not commercially available and require development for use with mixed fluids. Therefore, in comparing these results to actual operating plants, a certain amount of caution must be exercised. However, to keep this study comparable to many proposed systems, we used commonly quoted specifications for the cycle components as listed above.

In the rest of the paper, we refer to brine effectiveness in units of watt-hours per pound (Wh/lb) as the net cycle power output per unit of brine mass flow rate. Note that for the net output, all liquid feed pumping power, air fan power, and brine pumping power are subtracted from the gross electrical output. Additional losses related to plant auxiliaries are also taken out.

Baseline Design

A schematic diagram of the MR cycle [3] is shown in Figure 1. The feed pump delivers the feed (consisting of a mixture of light and heavy components) to the boiler at the required pressure. For our purpose, the feed is a mixture of ammonia and water. The feed exits the vaporizer at a quality less than one. It is separated into vapor (consisting mostly of light components) and liquid (consisting of heavy components) streams. The vapor expands through a turbine to produce power. The liquid from the separator heats the returning condensate in a recuperator. This cooled liquid is mixed with the turbine exhaust vapor to reduce the condenser pressure. An ACC turns the mixed vapor into liquid while rejecting heat to the ambient air.

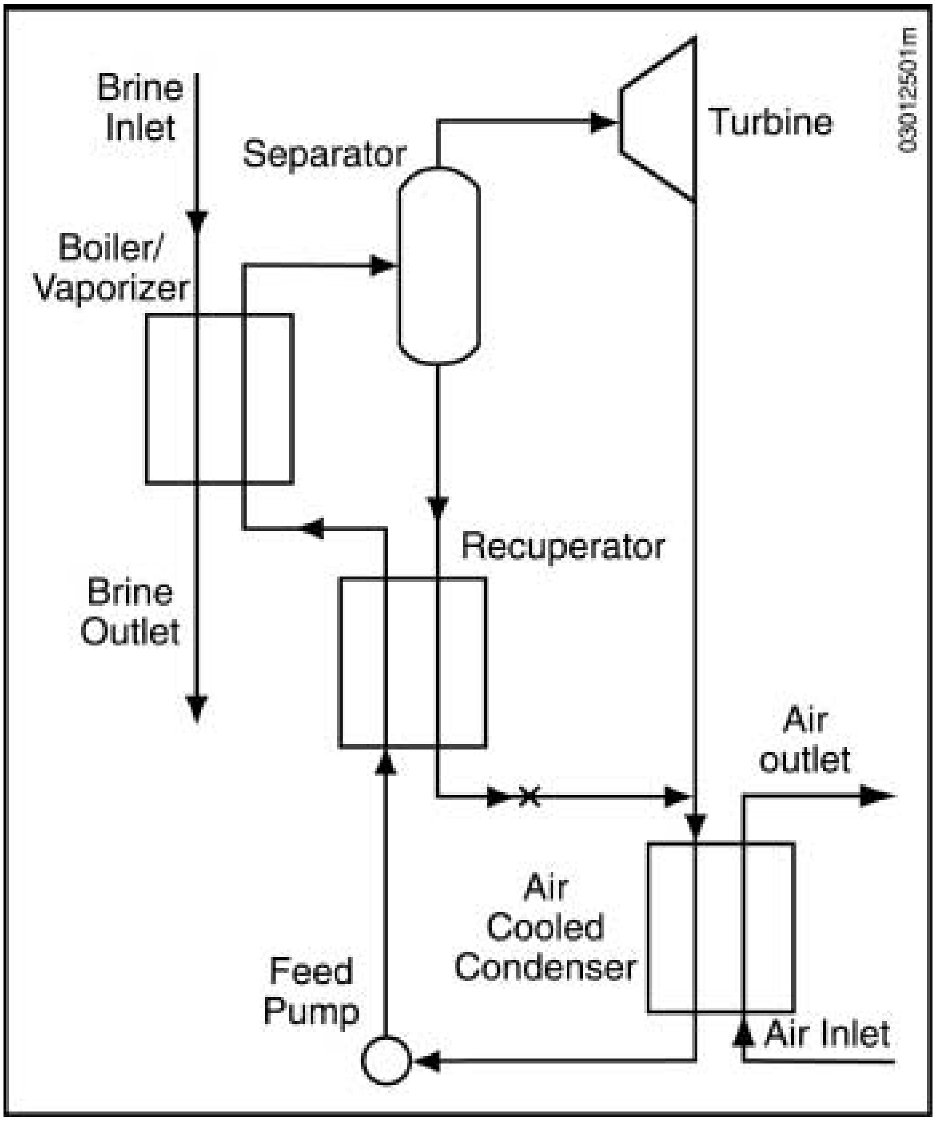

Figure 1. Schematic diagram of a Maloney-Robertson (MR) cycle.

For a Rankine cycle that uses a pure fluid, given fixed heating- and cooling-fluid flow rates, the pinch points at the boiler and condenser fix the boiling and condensing temperatures (or pressures). In contrast, in an MR cycle, one additional degree of freedom is available. Given boiler pressure, one can choose either the concentration of the working fluid or the quality of the mixture exiting the evaporator.

For these analyses, we chose to vary both the concentration of the working fluid and the quality of the exit mixture and allowed the boiler pressure to float. All analyses were conducted using a commercially available software, “ASPEN Plus” [3]. Figure 2 shows the results of the analysis for a brine resource temperature of 300^{o}F. In this figure, the variation of brine effectiveness is plotted as functions of ammonia concentration and vapor quality. For this figure, brine effectiveness does not include losses for brine pumping and plant auxiliaries.

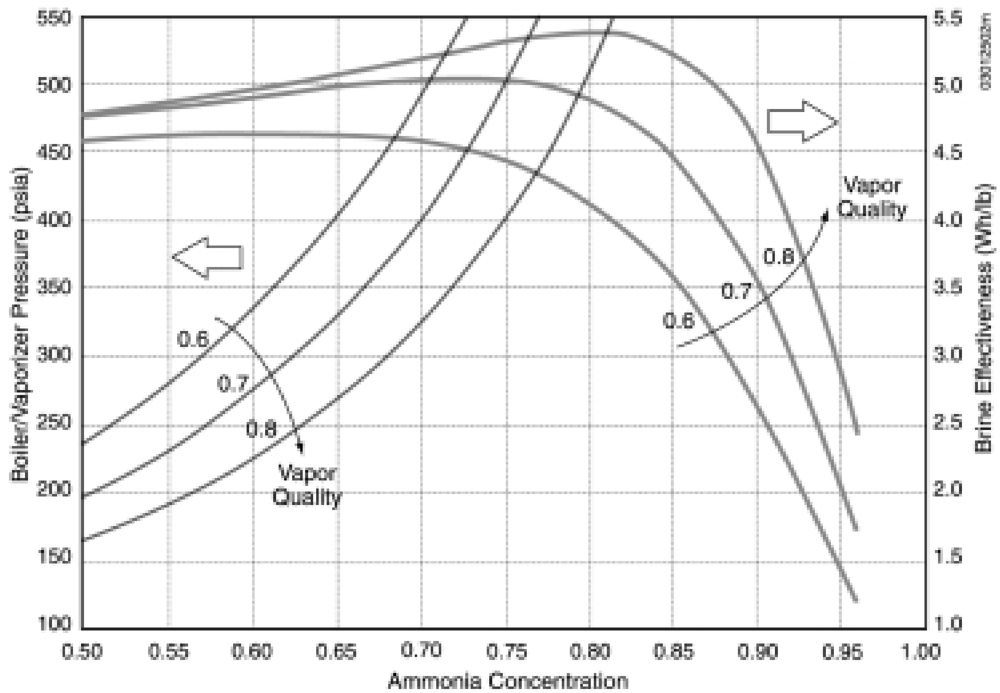

Figure 2. Variations of brine effectiveness as functions of feed concentration and boiler exit quality

We limited the range of ammonia concentrations from 0.5 to 0.95. In this range, the heating curve for the mixture is concave. (At lower concentrations, the heating curve becomes convex, thus requiring more brine flow). At a given vapor quality, with increasing concentration, the brine effectiveness increases first, reaching a maximum, and then decreases gradually. Increasing the vapor quality increases the concentration at which this maximum occurs. The family of curves for various qualities shows an envelope, which exhibits a gradual rise in effectiveness with increasing concentration. The envelope shows that the maximum effectiveness gradually rises from about 4.5 Wh/lb at a concentration of 0.5 to a value of 5.4 Wh/lb at a concentration of 0.8.

The corresponding variations of boiler (vaporizer) pressure against the two variables are also included in this figure. Increasing the ammonia concentration increases the pressure. Increased vapor quality decreases the pressure. Practical considerations (related to cost and safety) may impose limits on the high pressure. Limiting the boiler pressure to 300 psia and selecting a vapor quality of 0.7 and a concentration of 0.625 results in a near-optimum condition for the design.

With these chosen values, we generated details for plant components. Key results for the cycle are shown in Table 1, column (a).

Of specific interest is the design for the ACC. Detailed analysis indicates that an ACC's average logarithmic mean temperature difference (LMTD) is 9.58^{o}F and its UA-product requirement turns out to be 2.66x10^{6} Btu/hr^{o}F. Assuming a typical overall heat transfer coefficient U of 5 Btu/hrft^{2o}F and a typical finned-area-to-bare-tube-area ratio of 22 (see, for example, [4]), the required bare tube area for the ACC is 24,150 ft^{2}. For this condenser, the fan induces a volumetric flow of air at 18.7×10^6 ft^3/hr. The condenser pressure for the design is at 45 psia, and the net power from the plant comes in at 1 MW. The net brine effectiveness at design turns out to be 3.88 Wh/lb, taking into account all parasitic losses.

We characterized the performance of this plant at an ambient temperature of 100°F. The design was frozen with the specified areas and physical characteristics for all components. The volumetric flow of air induced by the condenser fan was also held constant.

Column (b) of Table 1 summarizes the performance of the plant at the higher ambient temperature. Unchanged are the brine inlet conditions, the feed flow rate and the turbine inlet vapor flow rate, temperature, pressure, and concentration. However, to condense the mixture at a higher temperature, the turbine exhaust pressure must now be substantially higher, at 108 psia. On account of this, the turbine gross power decreases. The overall brine effectiveness turns out to be approximately 2.00 Wh/lb. Net power output from the plant is now at 516 kW, showing a 48% reduction from design. Such a decrease in performance is typical in binary power plants that use ACCs [5].

The influence of a higher ambient temperature is primarily to increase the condenser pressure in the cycle. Increased condensate temperature also raises the brine outlet temperature and causes a reduction in heat input to the plant. For results presented here, we attempted to decrease the condenser pressure rise by diverting some of the vapor out of the condenser into an auxiliary absorber placed in series with the condenser.

For 100°F-ambient operation, we evaluated the influence of varying condenser backpressure on the power plant and its leftover vapor. Figure 3 shows the change in net output (expressed as a fraction of its operating value of 516 kW) as a function of varied condenser pressure. Net power increases almost linearly with decreasing condenser pressure. Also shown in this figure is the uncondensed vapor flow, again expressed as a fraction. For condenser pressures greater than 108 psia, the entire incoming vapor is condensed (i.e., none remains uncondensed). For lower condenser pressures, the uncondensed fraction of vapor increases almost linearly with decreasing condenser pressure. At a pressure of 85 psia, we are left with 14% uncondensed vapor and a nearly 27% increase in net power.

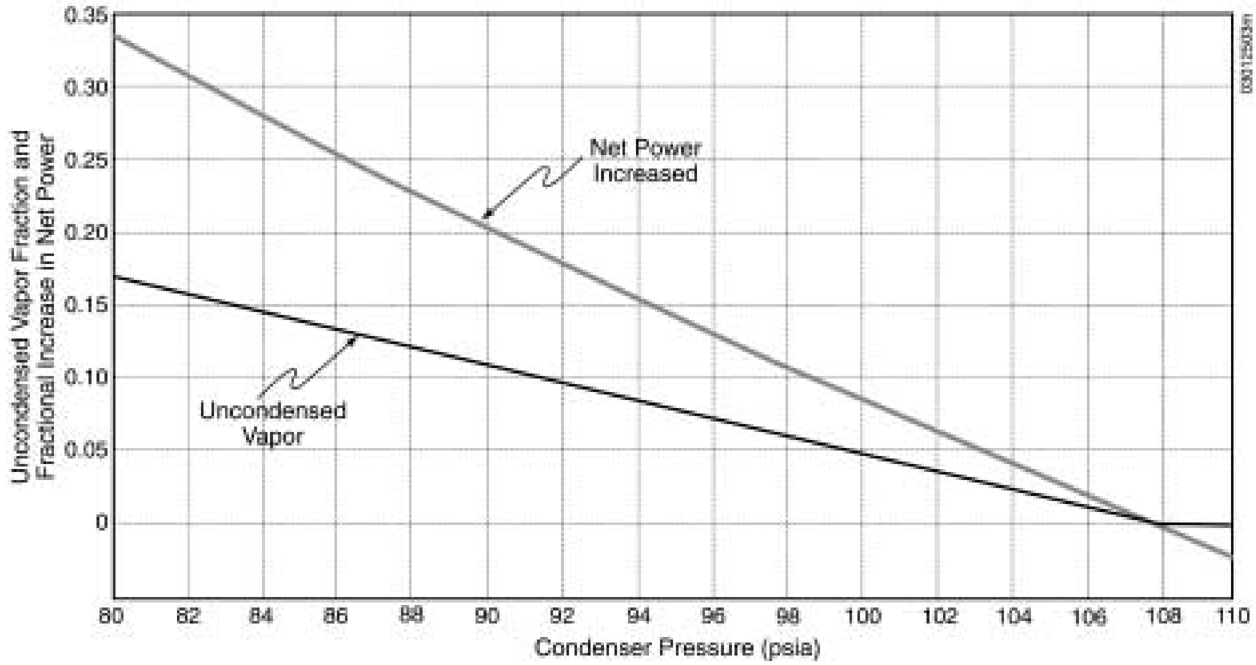

Figure 3. Influence of condenser backpressure on net power and uncondensed vapor.

The next section describes the absorber, which handles the uncondensed vapor and its influence on the performance of the overall plant.

Absorption Heat Pump

A schematic diagram of a single-stage absorption heat pump is shown in Figure 4. In this case, the system consists of an isothermal (air-cooled) absorber, a pump, a recuperator, primary heater, separator, an additional auxiliary air-cooled condenser, and a trim cooler. The absorber is placed downstream of the plant's main ACC. Uncondensed vapor from the ACC is absorbed in a steam of a lean mixture (of appropriate concentration) in the absorber. The concentration of the vapor entering the absorber is adjusted to a value such that it can be regenerated downstream at the heater. Mixing some of the condensate from the ACC with the uncondensed vapor provides the means to adjust this concentration.

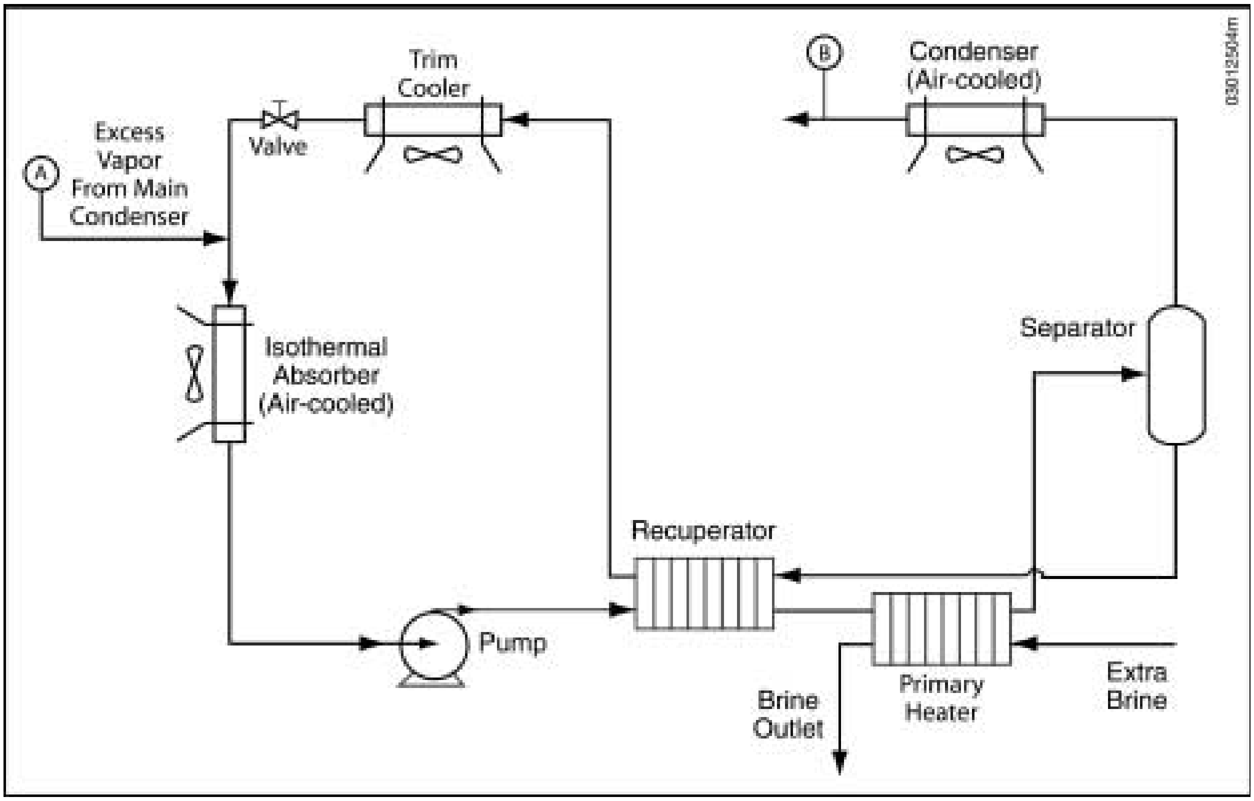

Figure 4. Schematic diagram of an absorption heat pump for handling uncondensed vapor from plant's main ACC

The pump pressurizes the absorber effluent to the primary heater (or regenerator) pressure. The mixture then flows through a recuperator and then into the regenerator, where it is heated using extra brine. Regenerated vapor is condensed in the auxiliary ACC for return to the main plant. The liquid from the separator returns to the absorber for reuse (after being cooled in a trim cooler and reduced in pressure to that of the absorber). Substantial flexibility exists for the selection of operating conditions for the absorption system. We report results for one set of operating conditions chosen to illustrate the system behavior.

For the example studied, we reduced the main ACC pressure from 108 psia to 85 psia. To reduce this pressure, we removed 14% of uncondensed vapor from it. This vapor is essentially pure ammonia, and is difficult to reconstitute for return. To facilitate reconstitution, some of the condensate is mixed with this vapor before sending it to the absorber. For evaluation of this system, we chose a regeneration pressure of about 223 psia, and a condensing temperature of 120°F.

Results

Table 1, column (c) provides a summary of the evaluations. By reducing the turbine exhaust pressure to 85 psia, net power output from the power cycle increases from 516 kW to 621 kW. The operation of the absorption heat pump together with additional brine pumping uses up 34 kW. The net result is an increased yield of 105 kW or nearly 20%. The corresponding increase in brine flow is 29%.

Table 1. Details of Power System Operating Conditions

Description		Units	(a) 50ºF	(b) 100ºF	(c) 100ºF (with absorber)	Remarks
			Operation at Ambient Temperature			
Brine Inlet						
	Flow rate	(lb/hr)	258200	258200	332800	29% Excess Brine used.
	Temperature	(°F)	300	300	300	
Brine Outlet						
	Temperature	(°F)	187	194	194	Excess brine leaves at 189ºF
Feed						
	Flow rate	(lb/hr)	51500	51500	51500	
	Temperature	(°F)	58	102	102	
	Pressure	(psia)	45	108	85	
	Concentration	(---)	0.625	0.625	0.625	
Turbine Inlet						
	Vapor flow rate	(lb/hr)	35730	35730	35730	
	Temperature	(°F)	295	295	295	
	Pressure	(psia)	303	303	303	
	Concentration	(---)	0.794	0.794	0.794	
Turbine Exhaust						
	Temperature	(°F)	177	226	212	
	Pressure	(psia)	45	108	85	
Air-cooled Condenser						
	Air flow rate	(lb/hr)	1453240	1323630	1323630	Volume flow remains constant.
	Air Inlet Temperature	(°F)	50	100	100	
	Air Outlet Temperature	(°F)	123	179	166	
Power (Electrical) Summary						
	Gross	(kW)	1126	637	776	
	Brine pump	(kW)	37	37	48	Excess brine requires 11 kW
	Feed pump	(kW)	26	21	21	
	Fans	(kW)	23	23	23	
	Miscellaneous	(kW)	40	40	40	
	Absorption Heat Pump Operation	(kW)			23	Includes power for pump and all fans
Net Power Output		(kW)	1000	516	621	Additional power = 105 kW Net yield is 20% power
Overall Brine Effectiveness		(Wh/lb)	3.873	1.998	1.866	

For operation at 100°F ambient temperature, a set of similar evaluations of an absorber for a plant using brine at a temperature of 250°F yielded a net increase of 13% in power with 26% excess brine.

Operation of an absorption heat pump to alleviate reduction in plant output is possible. However, the costs related to introduction and operation of such equipment and the corresponding returns on investment remain to be ascertained.

Recommendations

In this study, we investigated the use of an absorption heat pump to mitigate the influence of ambient temperature rise. A plant that uses an ammonia and water mix as the working fluid was investigated. The results indicate that use of an absorption system can result in additional net power at high ambient temperatures.

There are many other methods that can mitigate the influence of ambient temperature rise. Such methods also deserve to be evaluated. However, because these studies invariably require off-design performance evaluations for components, the results will be plant specific. Potential revenue from increased production capacity at high ambient temperatures will play a major role in an assessment of the cost benefits.

A generalized approach to address this problem does not result in useful guidelines or conclusions. For particular applications to particular plants, off-design case studies must be conducted at their site-specific conditions.

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