A national laboratory of the U.S. Department of Energy
Office of Energy Efficiency & Renewable Energy

National Renewable Energy Laboratory

Innovation for Our Energy Future

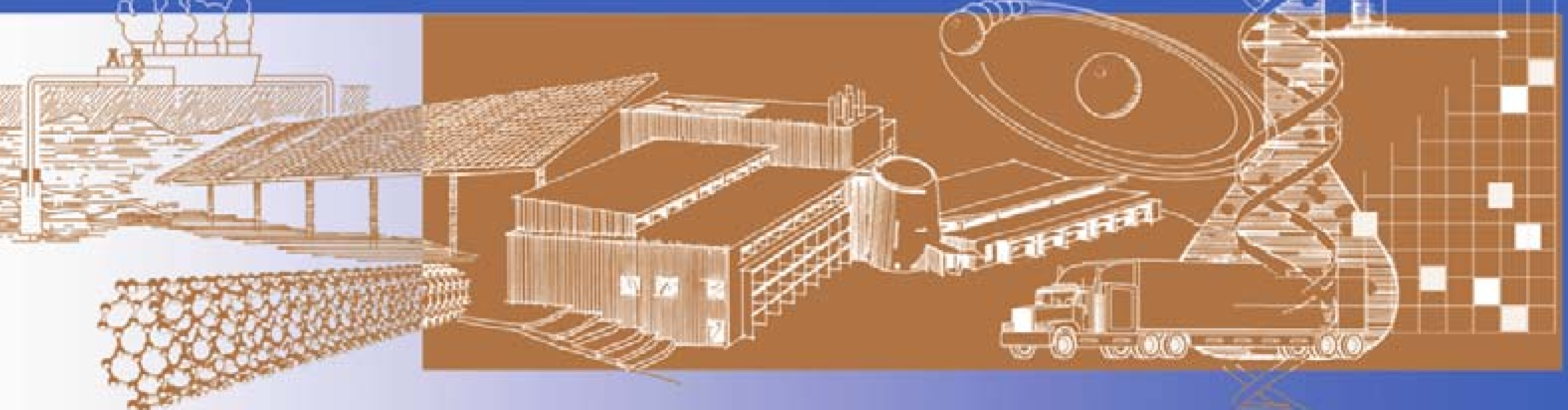
Nexant Parabolic Trough Solar Power Plant Systems Analysis

Task 2: Comparison of Wet and Dry Rankine Cycle Heat Rejection January 20, 2005 – December 31, 2005

B. Kelly
Nexant, Inc.
San Francisco, California

Subcontract Report
NREL/SR-550-40163
July 2006

NREL is operated by Midwest Research Institute • Battelle Contract No. DE-AC36-99-GO10337



Nexant Parabolic Trough Solar Power Plant Systems Analysis

Task 2: Comparison of Wet and Dry Rankine Cycle Heat Rejection January 20, 2005 – December 31, 2005

Subcontract Report
NREL/SR-550-40163
July 2006

B. Kelly
Nexant, Inc.
San Francisco, California

NREL Technical Monitor: H. Price
Prepared under Subcontract No. LDC-5-55014-01

National Renewable Energy Laboratory
1617 Cole Boulevard, Golden, Colorado 80401-3393
303-275-3000 • www.nrel.gov

Operated for the U.S. Department of Energy
Office of Energy Efficiency and Renewable Energy
by Midwest Research Institute • Battelle

Contract No. DE-AC36-99-GO10337

Available electronically at http://www.osti.gov/bridge

Available for a processing fee to U.S. Department of Energy
and its contractors, in paper, from:
U.S. Department of Energy
Office of Scientific and Technical Information
P.O. Box 62
Oak Ridge, TN 37831-0062
phone: 865.576.8401
fax: 865.576.5728
email: mailto:reports@adonis.osti.gov

Available for sale to the public, in paper, from:
U.S. Department of Commerce
National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
phone: 800.553.6847
fax: 703.605.6900
email: orders@ntis.fedworld.gov
online ordering: http://www.ntis.gov/ordering.htm

Printed on paper containing at least 50% wastepaper, including 20% post-consumer waste

Contents

Task 2
Comparison of Wet and Dry Rankine Cycle Heat Rejection

1. Introduction

The efficiency of a Rankine cycle is defined, in large part, by the pressure and the temperature of the steam both entering and leaving the turbine. The cycle efficiency can be improved either by raising the pressure and the temperature at the inlet to the turbine, or decreasing the pressure and the temperature at the outlet.

The steam conditions at the turbine outlet are defined by the temperature at which the steam is condensed and the latent heat of vaporization can be transferred to the environment. The lowest ambient temperature available is the wet bulb temperature; thus, most power plants use an evaporation process to provide the cooling water source for the condenser. However, the principal heat transfer mechanism in a wet cooling tower is evaporation. As a result, approximately 1 pound of water must be evaporated for each pound of steam condensed, and the water consumption in a large power plant can be significant. For example, an 80 MWe parabolic trough solar plant, operating with a capacity factor of 27 percent, will consume about 725,000 tons of water per year.

For sites which have a limited supply of water, heat can be rejected to the environment by condensing turbine exhaust steam at the dry bulb, rather than the wet bulb, temperature. For desert sites, design values for the dry bulb and the wet bulb temperatures are about 104 °F and 68 °F, respectively. Compared with a turbine inlet temperature of 703 °F, a difference of 36 °F in the steam condensation temperature does not appear significant. However, the work performed in the turbine expansion process is defined as $\int \nu \, dP$, where ν is the fluid specific volume and dP is the change in pressure. With a turbine inlet pressure of 1,450 lb_f/in^2, and an outlet pressure of 1.07 lb_f/in^2 defined by a condensation temperature of 104 °F, a theoretical overall pressure ratio of 1,360 can be achieved. However, an outlet pressure of 0.34 lb_f/in^2, defined by a condensation temperature of 68 °F, results in an overall pressure ratio of 4,260. Granted, the theoretical pressure ratios cannot be achieved due to economic limits on heat exchange area. Nonetheless, it is clear that small changes in the condensation temperature can have a large influence on the expansion ratio, and therefore the work performed by the steam.

An economic analysis was conducted to determine 1) the preferred design conditions for a dry cooling tower, and 2) the anticipated increase in the levelized cost of energy due the selection of a dry, rather than a wet, cooling tower. For the purposes of the analysis, the power plant was assumed to be an 80 MWe parabolic trough facility located near Barstow, California. The study was conducted through the following steps:

- A model of an 80 MWe Rankine cycle using an air cooled condenser was developed using the GateCycle program (Reference 1). Six models were developed, with initial temperature differences between 24 °F and 49 °F. (Initial temperature difference is defined as Dry bulb temperature - Steam condensation temperature.) For each of the six models, estimates of turbine output and cooling fan power demand were made for dry bulb temperatures between 40 °F and 130 °F.

- A histogram of the hourly dry bulb temperatures, at direct normal radiation values above 250 W/m^2, was assembled for Barstow. For each of the six models, the predicted turbine output and fan power demand at each of the 21 ambient temperatures in the histogram was multiplied by the number of hours at each temperature, and the outputs summed to estimate the annual plant performance.

- Capital cost estimates for the air cooled condenser in each of the six models were developed. The relative economic benefits among the six models were plotted for a range of energy values between $60/MWhe and $140/MWhe to determine the sensitivity of the preferred initial temperature difference on the selling price of electric energy.

- A second model of an 80 MWe Rankine cycle, this using a wet cooling tower, was developed with the GateCycle program. One case was developed, with a condenser cooling water temperature range of 22 °F and a cooling tower approach to the wet bulb temperature of 12 °F.

- A histogram of the hourly dry bulb temperatures and coincident relative humidities, at direct normal radiation values above 250 W/m^2, was assembled for Barstow. From these data, an equation relating dry bulb temperature and annual average relative humidity was developed.

- Estimates of turbine output, cooling fan power, and water consumption for the wet heat rejection case were made for combinations of dry bulb temperatures between 40 °F and 130 °F and the corresponding relative humidities. From this, equations for estimating Rankine cycle performance were developed for use in the Excelergy computer program. The annual net electric output and water consumption of the plant were then estimated.

- Estimates of the capital costs and the operating costs for the six plants with a dry heat rejection system, and the one plant with the wet heat rejection system, were developed. The costs were used as inputs to an annual cash flow analysis to determine the levelized energy costs for the seven cases.

- For the dry heat rejection case with the lowest energy cost, additional GateCycle calculations were performed to estimate the turbine output if the maximum exhaust pressure was limited to 8 in. HgA during those periods in which the dry bulb temperature exceeded 110 °F. Equations of turbine output and fan power demand as a function of dry bulb temperature were incorporated in the Excelergy computer program, from which the annual net electric output and levelized energy cost were estimated.

2. Dry Heat Rejection

The procedure for determining the performance and the operating cost for the air cooled condenser in a plant with dry heat rejection is outlined below.

2.1 Rankine Cycle

The Rankine cycle design closely followed that developed by Fichtner for the 55 MWe AndaSol project in Spain. The cycle is a conventional, single reheat design with 5 closed and 1 open extraction feedwater heaters. The live steam pressure and temperature are 1,450 lb_f/in^2 and 703 °F, respectively, and the reheat steam temperature is 703 °F. The GateCycle flow diagram is shown in Figure 1.

Cold and hot reheat steam pressures, feedwater heater extraction pressures, feedwater heater terminal temperature differences, and feedwater heater drain cooler approach temperatures were taken from the Fichtner flow diagram. Pressure losses in the steam lines to the feedwater heaters were set to zero, as implied in the Fichtner diagram. The condenser pressure was set to 1.23 lb_f/in^2, or 2.5 in. HgA.

Turbine expansion efficiencies, and the required live and reheat steam flow rates to achieve a gross output of 88.0 MWe, were calculated by GateCycle. Simultaneously, the low pressure turbine exhaust loss was adjusted manually to yield a gross cycle efficiency of 0.377.

2.2 Air Cooled Condenser Sizes

Air cooled condenser heat transfer areas were calculated for the 6 initial temperature differences of 24 °F, 29 °F, 34 °F, 39 °F, 44 °F, and 49 °F. The calculations were based on a dry bulb temperature of 106 °F, which is not exceed for all but 1 percent of the hours each year at Barstow. An allowance for subcooling the water leaving the condenser by 2 °F is provided to ensure the flow to the condensate pump is single-phase.

The design parameters for the heat exchangers are listed in Table 1, and the calculated areas and fan power requirements are shown in Table 2. As expected, both the heat transfer areas and the fan power requirements are inversely proportional to the initial temperature difference.

2.3 Dry Bulb Temperature Distribution

A list of the hourly dry bulb temperatures was derived from the Excelergy weather file DAG_TMY2_hr. The list was sorted into a series of 21 bins representing 5 °F increments in temperature between 20 °F and 125 °F. The summations were limited to those hours in which the plant was in operation by selecting temperatures only for direct normal radiation values above 250 W/m^2. The resulting histogram is shown in Figure 2.

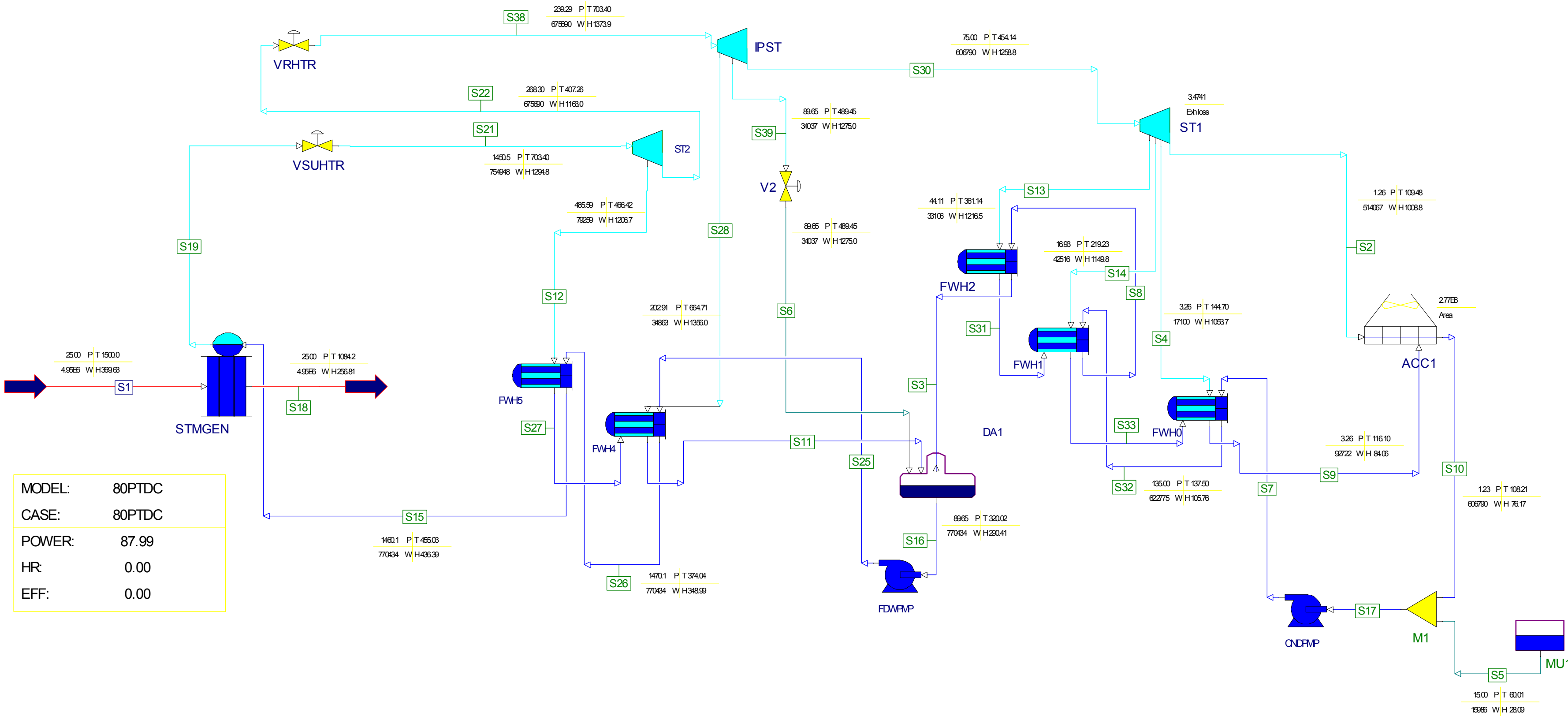


Figure 1 GateCycle Flow Diagram
88 MWe Rankine Cycle with Dry Heat Rejection

Table 1
Air Cooled Condenser Design Parameters

Parameter	Value
Condenser pressure, in. HgA	2.5
Tubes	
- Diameter, in.	1.0
- Wall thickness, in.	0.05
- Arrangement	Staggered
- Rows perpendicular to air flow	3
Fins	
- Type	Round
- Diameter, in.	2.74
- Thickness, in.	0.04
- Fins per inch, each	9
Air velocity, ft/sec	11.5
Overall heat transfer coefficient, Btu/hr-ft^2-F	7.79 [1]
Exit subcooling, °F	2

Note 1: Based on sum of outside tube surface area and fin area

Table 2
Air Cooled Condenser Surface Areas and Fan Power

Initial temperature difference, °F	Heat transfer area[1], ft^2	Total fan power, kWe
24	4,378,469	4,561
29	3,596,010	3,752
34	3,046,976	3,181
39	2,644,881	2,751
44	2,325,841	2,430
49	2,078,986	2,167

Note 1: Sum of outside tube surface area and fin area

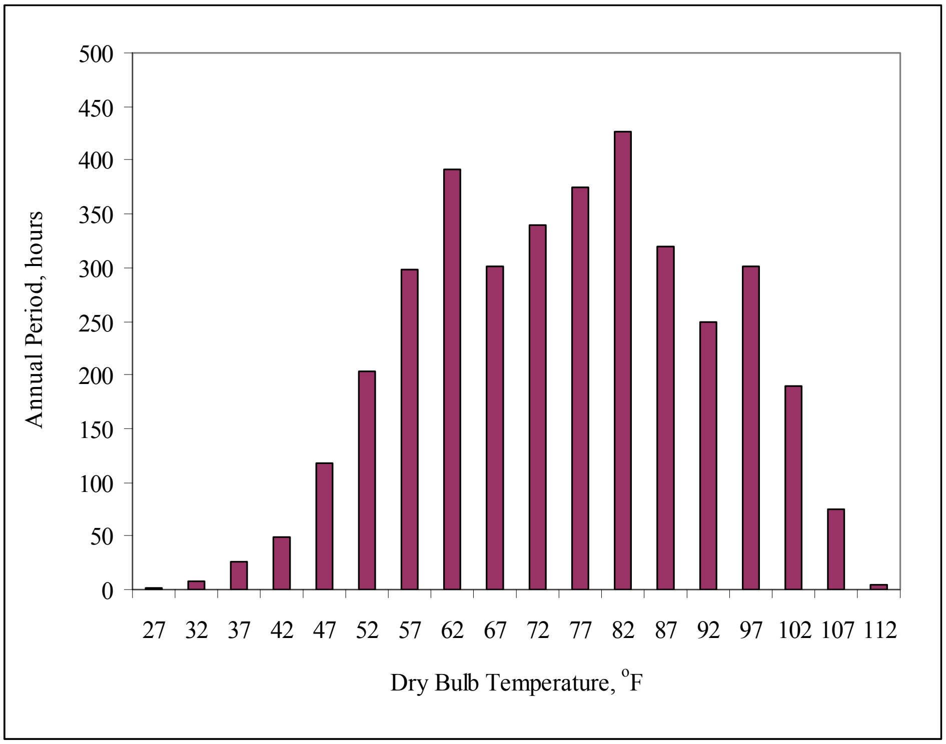


Figure 2 Dry Bulb Temperature Distribution for Barstow, California

2.4 Parametric Studies

For each of the air cooled condenser areas listed in Table 2, the performance of the turbine was modeled for a series of dry bulb temperatures in the range of 40 °F to 130 °F. From this, equations were developed for estimating turbine output as a function of the dry bulb temperature. Gross turbine outputs and fan power demands were then calculated for each of the 21 temperature bins. The gross power outputs and the fan power demands were multiplied by the hours in each bin, and then summed over the year to estimate the annual gross output, the annual fan energy demand, and the net plant output. The results are shown in Table 3.

The gross output increases uniformly as the initial temperature difference decreases. However, the highest net output occurs with an initial temperature difference of 29 °F; the incremental increase in the fan energy demand at 24 °F compared to 29 °F is higher than the incremental increase in gross output.

The line labeled "Net incremental output, MWhe" is the net output compared to the net output at a selected reference initial temperature difference of 44 °F. The line labeled "Allowable incremental capital cost, $" is calculated as follows:

$$\frac{\textit{(Net incremental output, MWhe)(Energy value, \$/MWhe)}}{\textit{Fixed charge rate}}$$

Table 3
Air Cooled Condenser Parametric Studies

	Initial Temperature Difference, F											
	24 F		29 F		34 F		39 F		44 F		49 F	
Dry Bulb Temperature Distribution	MWe	MWhe	MWe	MWhe	MWe	MWhe	MWe	MWhe	MWe	MWhe	MWe	MWhe
0 hours from 20 F to 24.9 F	103.6	0	101.5	0	94.8	0	94.3	0	86.1	0	84.2	0
1 hours from 25 F to 29.9 F	98.6	99	96.9	97	92.6	93	92.4	92	87.7	88	86.6	87
7 hours from 30 F to 34.9 F	95.0	665	93.8	657	91.2	638	91.1	638	88.9	622	88.4	618
26 hours from 35 F to 39.9 F	92.6	2,408	91.8	2,387	90.4	2,352	90.5	2,352	89.7	2,333	89.5	2,327
49 hours from 40 F to 44.9 F	91.1	4,466	90.6	4,441	90.1	4,416	90.2	4,418	90.2	4,422	90.1	4,415
118 hours from 45 F to 49.9 F	90.3	10,661	90.1	10,627	90.0	10,626	90.0	10,624	90.4	10,668	90.2	10,646
204 hours from 50 F to 54.9 F	90.0	18,366	89.9	18,336	90.1	18,375	90.0	18,352	90.2	18,406	89.9	18,338
298 hours from 55 F to 59.9 F	90.0	26,814	89.9	26,791	90.1	26,841	89.8	26,767	89.7	26,735	89.2	26,571
392 hours from 60 F to 64.9 F	90.0	35,297	90.0	35,267	89.9	35,255	89.5	35,092	88.9	34,845	88.1	34,532
301 hours from 65 F to 69.9 F	90.1	27,115	89.9	27,072	89.6	26,966	89.0	26,784	87.8	26,420	86.7	26,105
340 hours from 70 F to 74.9 F	90.0	30,596	89.7	30,503	89.0	30,250	88.2	29,977	86.4	29,376	85.1	28,942
375 hours from 75 F to 79.9 F	89.7	33,626	89.2	33,457	88.0	33,018	87.0	32,642	84.8	31,800	83.3	31,249
427 hours from 80 F to 84.9 F	89.1	38,033	88.4	37,747	86.8	37,066	85.6	36,555	83.0	35,449	81.4	34,760
320 hours from 85 F to 89.9 F	88.2	28,209	87.2	27,917	85.3	27,283	83.9	26,844	81.1	25,952	79.4	25,408
249 hours from 90 F to 94.9 F	86.9	21,641	85.7	21,351	83.4	20,778	81.9	20,397	79.1	19,697	77.4	19,266
301 hours from 95 F to 99.9 F	85.4	25,693	84.0	25,269	81.4	24,507	79.8	24,010	77.1	23,203	75.4	22,690
189 hours from 100 F to 104.9 F	83.5	15,790	81.9	15,482	79.3	14,981	77.5	14,654	75.1	14,197	73.5	13,888
75 hours from 105 F to 105.9 F	81.5	6,115	79.7	5,980	77.1	5,781	75.3	5,649	73.3	5,495	71.7	5,380
5 hours from 110 F to 114.9 F	79.4	397	77.5	388	75.0	375	73.3	366	71.6	358	70.2	351
0 hours from 115 F to 119.9 F	77.3	0	75.4	0	73.2	0	71.5	0	70.2	0	68.9	0
0 hours from 120 F to 124.9 F	75.4	0	73.6	0	71.8	0	70.3	0	69.2	0	68.0	0

3,677 total annual operating hours												
Gross output, MWhe		325,990		323,769		319,601		316,213		310,065		305,571
Fan energy, MWhe		16,771		13,795		11,695		10,116		8,935		7,967
Net output, MWhe		309,219		309,974		307,906		306,097		301,131		297,604
Net incremental output, MWhe		11,615		12,370		10,302		8,493		3,526		0
Energy value, $/MWhe		120		120		120		120		120		120
Allowable incremental capital cost, $		9,291,937		9,895,614		8,241,463		6,794,619		2,821,155		0
Condenser area, ft^2		4,378,469		3,596,010		3,046,976		2,644,881		2,325,841		2,078,986
Condenser cost, $		21,783,431		18,490,143		15,753,691		13,574,076		11,951,296		10,885,353
Incremental capital cost, $		10,898,078		7,604,790		4,868,338		2,688,723		1,065,943		0
Net cost benefit, $		-1,606,141		2,290,824		3,373,125		4,105,897		1,755,212		0

For the purposes of the study, a fixed charge rate of 0.15 has been assumed.

The line labeled “Condenser cost, $” is derived from the data shown in Figure 4 (Reference 2).

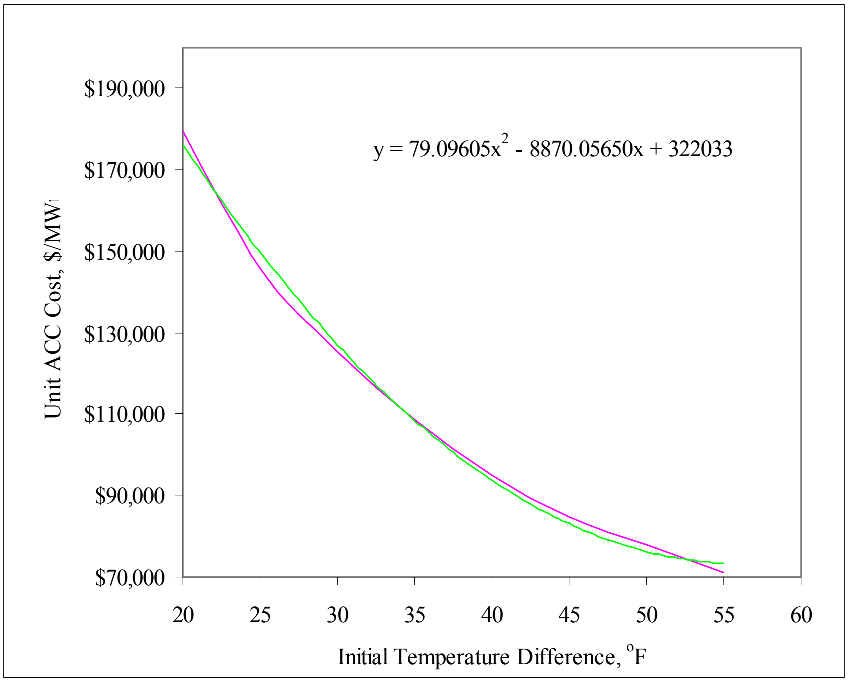


Figure 4 Unit Air Cooled Condenser Cost as a Function of Initial Temperature Difference

The “Incremental capital cost, $” is the capital cost of the condenser compared to the capital cost of the condenser at the selected reference initial temperature difference of 44 °F. The “Net cost benefit” is the incremental capital cost minus the allowable incremental capital cost.

The results for the six initial temperature differences, at each of five energy values between $60/MWhe and $140/MWhe, are illustrated in Figure 5. The ordinate locations of the curves are arbitrary since the adoption of a reference initial temperature difference of 44 °F was also arbitrary. The curves show the optimum initial temperature difference is likely to be in the range of 35 °F to 40 °F. Further, the curves show the optimum temperature difference is, to a large degree, insensitive to the selling price of the electric energy.

As noted in Table 3, the analyses assumed the Rankine cycle operated at full load throughout the year. This simplifying assumption was made to determine the sensitivity of the preferred initial temperature difference on the overall plant economics. A more detailed performance and economic analysis, based on the Excelergy program, is discussed in Section 4.

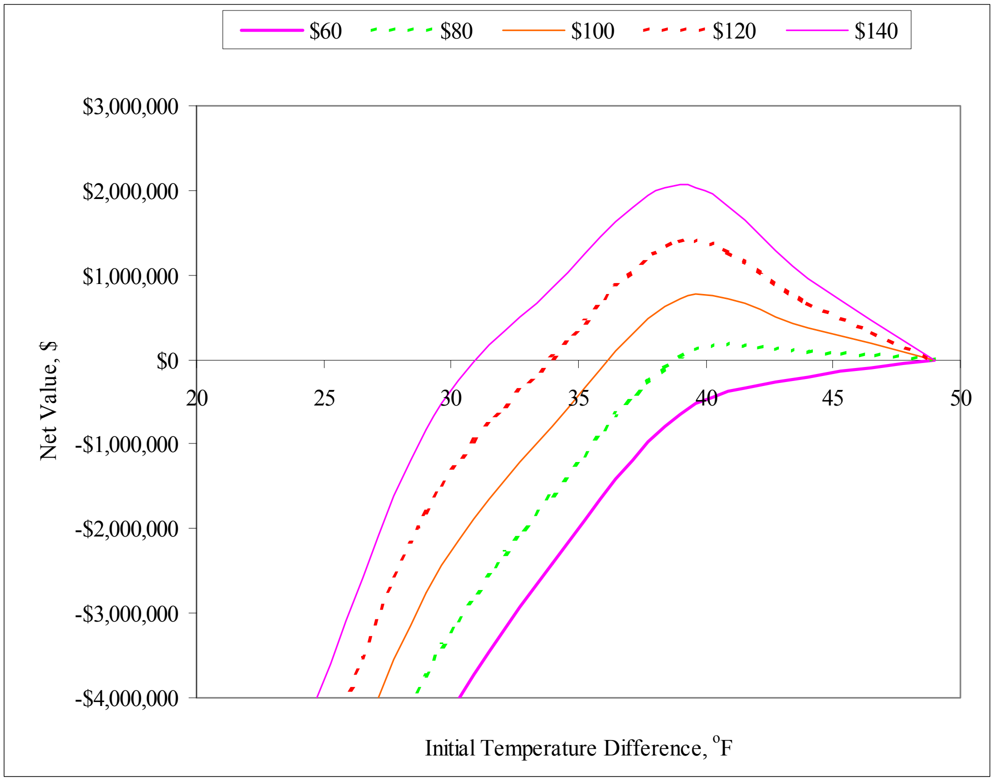


Figure 5 Net Value to Plant as a Function of Initial Temperature Difference

2.5 Additional GateCycle Calculations

With an air cooled condenser area of 2,645,000 ft^2, corresponding to the nominal preferred initial temperature difference of 39 °F, the condenser pressure reaches 8 in. HgA with an ambient temperature of about 108 °F. With an ambient temperature of 130 °F, the condenser pressure reaches (on a theoretical basis) 14.2 in. HgA.

In practice, the condenser pressure is likely to be limited to a value of about 8 in. HgA to prevent excessive aerodynamic loads on the last stage blades. To model the effect of this constraint on the output of the turbine, a series of additional GateCycle calculations were made at ambient temperatures of 110 °F, 120 °F, and 130 °F with the main steam flow rate reduced to the point where the condenser pressure does not exceed 8 in. HgA.

The results are illustrated in Figure 6. The upper lines represents the turbine output with no constraints on the condenser pressure, and the lower line shows the turbine output with the pressure limited to 8 in. HgA. With ambient temperatures of 120 °F and 130 °F, the turbine output must be reduced by a significant 31 percent and 55 percent, respectively. However, a review of the dry bulb temperature histogram for Barstow in Figure 2 shows only a limited number of hours with temperatures above 110 °F. As a result, the effect on the annual plant output due a constraint on the condenser pressure should be minor. The effect is explored on a quantitative basis in Sections 4 and 5 below.

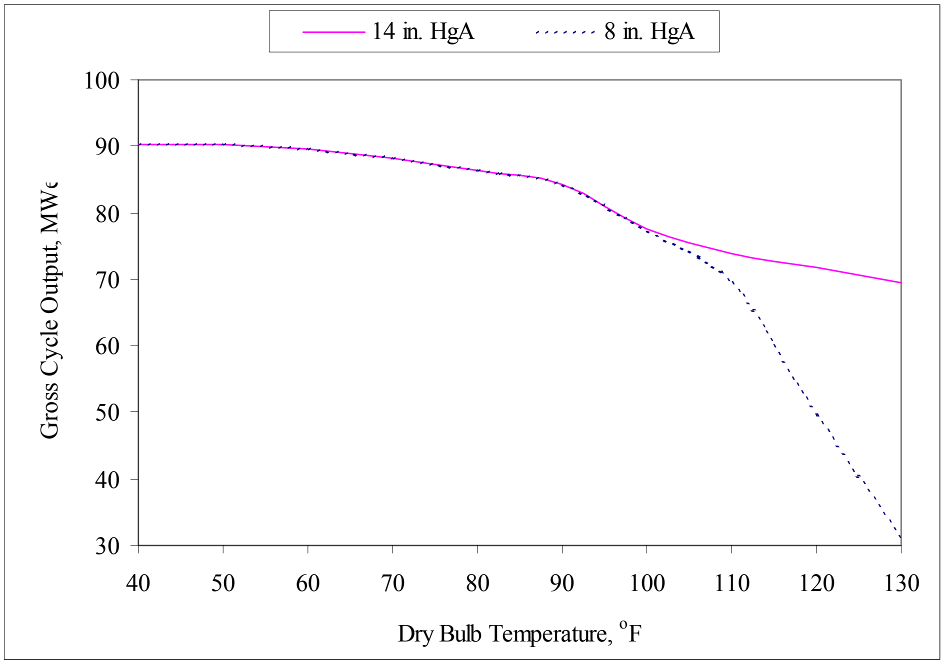


Figure 6 Gross Turbine Output
with Allowable Condenser Pressures of 14 in. HgA and 8 in. HgA

3. Wet Heat Rejection

The procedure for estimating the performance of a plant with a wet mechanical draft cooling tower is outlined below.

3.1 Rankine Cycle

The Rankine cycle design for a plant with a wet heat rejection system follows very closely the design for a plant with a dry heat rejection system. The principal changes are the deletion of the air cooled condenser, and the addition of a surface condenser, the wet cooling towers, a circulating water pump, and a makeup water source. The GateCycle flow diagram is shown in Figure 7.

The design condenser pressure is 1.23 lb_f/in^2 for both the dry and the wet heat rejection systems; thus, the gross cycle efficiency is 0.375 for both plants.

3.2 Wet Cooling Tower Capacity

The capacities of the wet cooling tower were selected by the GateCycle program, based on a dry bulb temperature of 106 °F and a coincident wet bulb temperature of 68 °F. This combination of temperatures is not expected to be exceeded for all but 1 percent of the hours each year. The characteristics of the cooling tower are summarized in Table 4.

Table 4
Wet Cooling Tower Design Parameters

Parameter	Value
Condenser pressure, in. HgA	2.5
Approach to wet bulb temperature, °F	12.3
Circulating water range, °F	21.1
Cells, each	3
Total fan power, kWe	881
Water consumption, lb_m/hr	
- Blowdown	154,694
- Evaporation	531,620
- Drift	22,513

- Total	708,827

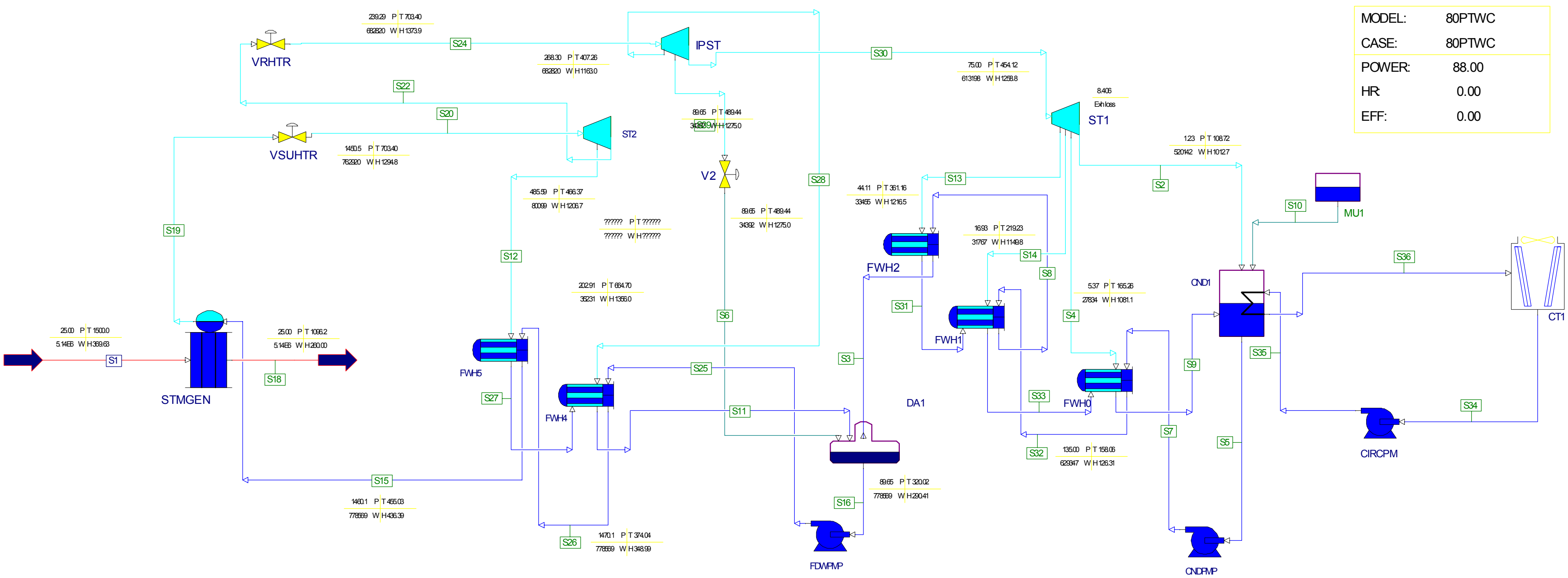


Figure 7 GateCycle Flow Diagram
88 MWe Rankine Cycle with Wet Heat Rejection

3.3 Annual Performance Calculations

The performance of a wet cooling tower is calculated by the GateCycle program using a combination of the dry bulb temperature and the relative humidity. However, the data contained in the Excelergy weather file DAG_TMY2_hr lists dry bulb temperature and dew point temperature. The relative humidities were calculated from the dew point temperatures using the standard expression, as follows:

$$\varphi = \frac{\text{Steam saturation pressure at dew point temperature, } lb_f/in^2}{\text{Steam saturation pressure at dry bulb temperature, } lb_f/in^2}$$

For the hours in which the direct normal radiation exceeded 250 W/m^2, the dry bulb temperatures were plotted against the corresponding relative humidities, which yielded the distribution shown in Figure 8.

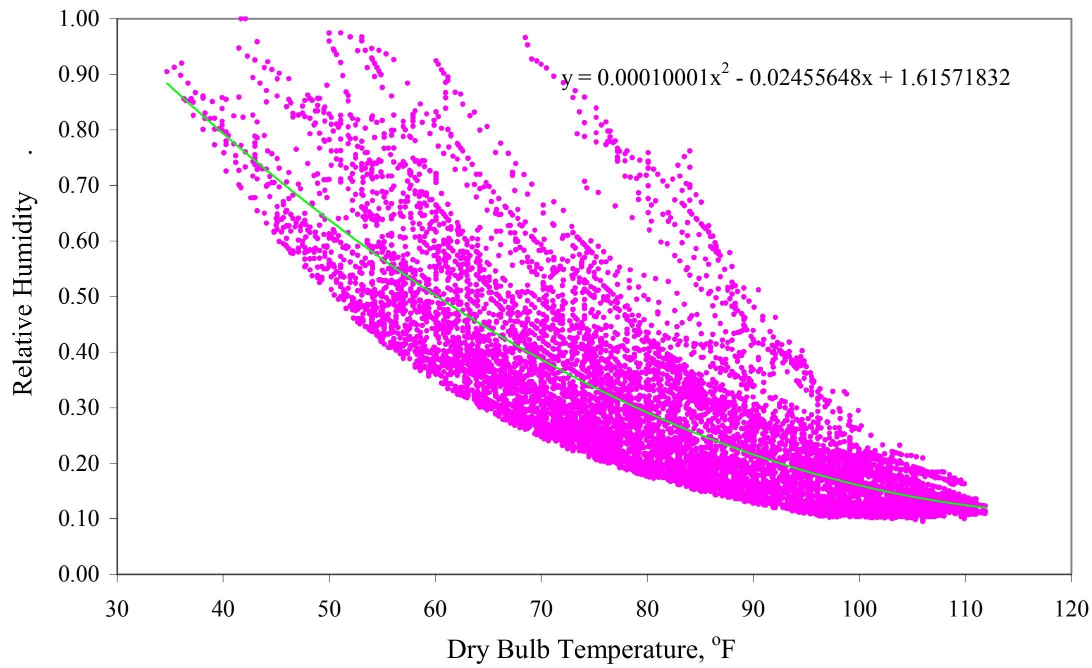


Figure 8 Dry Bulb Temperatures and Calculated Coincident Relative Humidities

Using the equation shown in Figure 8, a representative annual relative humidity was developed for each dry bulb temperature. From this information, the gross plant output and cooling tower water consumption were calculated for a series of dry bulb temperatures between 40 °F and 120 °F. The results, summarized in Table 5, show the plant output to be essentially invariant with the ambient temperature. However, the cooling tower water use varies inversely with the ambient temperature.

Table 5
Gross Plant Output as a Function of Dry Bulb Temperature
Wet Heat Rejection

Dry bulb temperature, °F	Relative humidity	Gross turbine output, MWe	Cooling tower makeup[1], lb_m/hr
40	0.79	88.5	383,000
50	0.64	88.5	456,000
60	0.50	88.5	527,000
70	0.39	88.4	584,000
80	0.29	88.4	635,000
90	0.22	88.3	686,000
100	0.16	88.2	737,000
110	0.12	88.1	789,000
120	0.11	88.0	841,000

Note 1: Sum of blowdown, evaporation, and drift losses.

Strictly speaking, numerous relative humidities are associated with each dry bulb temperature, as illustrated in the data points of Figure 8. Fortunately, for sites with low relative humidities during the summer, the performance of the Rankine cycle is essentially invariant with the dry bulb temperature. As a result, assigning only one relative humidity to each dry bulb temperature should result in an annual energy estimate which is very close to a more complex analysis involving a three-dimensional surface fit of gross output as a function of dry bulb temperature and relative humidity.

A combination of an essentially constant turbine output and an inverse relationship between ambient temperature and water use should be characteristic of a desert location. In essence, the cooling tower always transfer heats to the environment under favorable conditions: When the ambient temperature is high, the relative humidity is low; and when the relative humidity is high, the ambient temperature is low. However, at other plant locations in which the relative humidity is not a strong function of the ambient temperature, the turbine output is likely to decline on hot days.

4. Annual Plant Performance

The net electric outputs for the plants with dry and wet heat rejection were estimated using the Excelergy computer program. The plant designs were based on the characteristics listed in Table 6.

Table 6
Plant Design Parameters for Use In Excelergy

Parameter	Value
Collector type	LS-2+
Collector field aperture area, m^2	534,230
Solar multiple	1.45
Gross plant output, MWe	88.0
Gross cycle efficiency	0.377
Solar field design parameters	Default
Solar field parasitic power demand	Default
Power block design parameters	Default
Power block parasitic power demand	Default [1]

Note 1: With separate calculations for cooling tower and circulating water pump auxiliary power consumption

4.1 Dry Heat Rejection

The gross output of the Rankine cycle was calculated using the standard Excelergy format, as follows:

$$Nth = .Qtpb / Qdesign$$
$$Nel = T2EPLF0 + (T2EPLF1)(Nth) + (T2EPLF2)(Nth)^2 + (T2EPLF3)(Nth)^3 + (T2EPLF4)(Nth)^4$$
$$.EgrSol = Edesign * Nel$$

where .Qtpb is the thermal power to the steam generator at each time step, Qdesign is the design thermal power to the steam generator, .EgrSol is the gross turbine output at each time step, and Edesign is the design gross turbine output. The part load thermal-to-electric coefficients T2EPLF0 through T2EPLF4 are the default Excelergy values; i.e., the ratio of part load to full load Rankine cycle efficiency is assumed to be independent of the heat rejection system.

The effect of the ambient temperature on the gross cycle output is also modeled using the standard Excelergy format, as follows:

$$Ntc = TempCorr0 + TempCorr1 * Ttc + TempCorr2 * Ttc^2 + TempCorr3 * Ttc^3 + TempCorr4 * Ttc^4$$
$$.EgrSol = .EgrSol * Ntc$$

where Ttc is the dry bulb temperature, and the five coefficients TempCorr0 through TempCorr4 are derived from a fourth order polynomial fit of GateCycle calculations of the gross turbine output plotted as a function of the dry bulb temperature.

The gross turbine output assumes all 14 cooling tower fans are in operation when the turbine is operating at or near full load, regardless of the ambient temperature. In principal, it may be possible to turn off some of the fans at low ambient temperatures to reduce the parasitic energy demand. To explore the potential energy savings, a series of GateCycle calculations were performed for the following conditions: 60 °F ambient temperature; 50 percent relative humidity; and 8 to 14 fans in operation. The results are illustrated in Figure 9. With 11 to 14 fans in operation, the reduction in parasitic energy demand associated with isolating a fan was essentially equal to the reduction in the gross output of the turbine due to an increase in the condenser pressure, and the net output of the plant remained nearly constant. However, with fewer than 11 fans in operation, the performance degradation due to the increase in the condenser pressure was larger than the savings in fan energy, and the net output decreased. For the purposes of the study, the fan power calculation assumed that all fans were in operation whenever the turbine was in operation at or near full load.

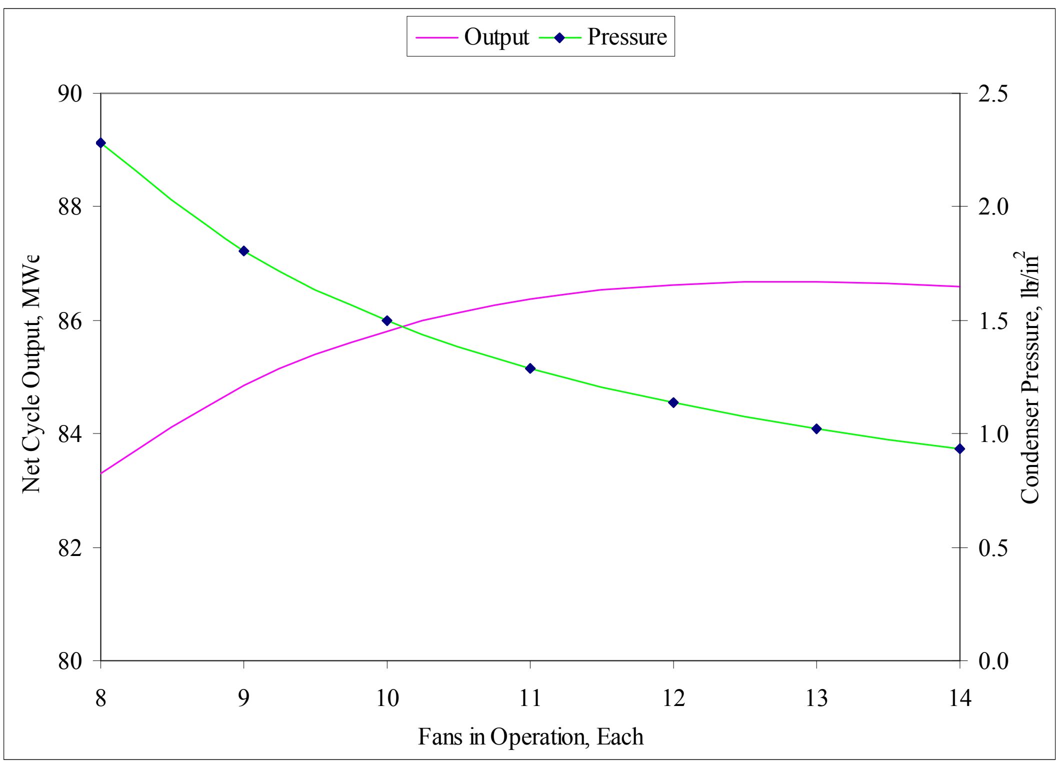


Figure 9 Plant Performance and Air Cooled Condenser Operation

On a related point, the GateCycle calculations for fan power consumption are based on an air velocity at the entrance to the tube bank of 11.5 ft/sec. As such, the mass flow rate and fan power consumption are inversely related to the ambient air temperature. For example, with an ambient temperature of 40 °F, the power demand of each fan is 219 kWe; at 140 °F, the demand falls to 182 kWe. A polynomial expression is included in the Excelergy parasitic block calculations to model this effect, as follows:

$$.EparCt = (CtPar)[(CtParF0 + (CtParF1)(AmbTemp(D, H, T)) + (CtParF2)(AmbTemp(D, H, T))^2)] (.PbLoad)$$

where .EparCt is the cooling tower energy demand at each time step, CtPar is the fan power demand at the design point, the coefficients CtParP0 through CtParF2 adjust the fan power demand with the ambient temperature, and .PbLoad is the ratio of gross electric output at each time step to the design electric output. As such, the number of cooling tower fans in operation is assumed to be proportional to the turbine output.

4.2 Wet Heat Rejection

The gross output of the Rankine cycle was calculated using a modified Excelergy format for wet cooling towers. The effect of the wet bulb temperature on the Rankine output was modeled as follows:

$$Ntc = TempCorr0 + TempCorr1 * Ttc + TempCorr2 * Ttc^2$$
$$.EgrSol = .EgrSol * Ntc$$

where Ttc is the dry bulb temperature, and the three coefficients TempCorr0 through TempCorr2 are derived from a second order polynomial fit of GateCycle calculations of the gross turbine output plotted as a function of the dry bulb temperature. Each dry bulb temperature is assumed to have a corresponding relative humidity, as illustrated in the trend line of Figure 8.

At the design point, the parasitic energy consumption was estimated to be 881 kWe for the cooling tower fans, and 653 kWe for the circulating water pumps. For combinations of ambient temperature and Rankine cycle output other than the design point, the energy demand was calculated as follows:

$$.EparCt = (CtPar)[(CtParF0 + (CtParF1)(AmbTemp(D, H, T)) + (CtParF2)(AmbTemp(D, H, T))^2)] (.PbLoad)$$

where the coefficients CtParP0 through CtParF2 are based on GateCycle calculations which adjust the fan and the circulating water pump power demands with the ambient temperature.

4.3 Annual Performance Comparison

The results of the annual performance calculations for the dry and the wet heat rejection cases are shown in Table 7. Cases 1 through 6 use dry heat rejection systems, with initial temperature differences of 24 °F to 49 °F, respectively. Case 7 is the same as Case 4, but with the condenser pressure limited to 8 in. HgA.. Case 8 uses a wet heat rejection system.

The dry heat rejection cases deliver 91 to 96 percent of the annual electric energy supplied by the wet heat rejection case, and have annual solar-to-electric efficiencies 0.5 to 0.7 percentage points lower. However, the annual water use for the dry cases is only about 8 percent of that for the wet case.

Table 7
Annual Energies for Plants with Dry and Wet Heat Rejection

Case	1	2	3	4	5	6	7	8
Type of cooling tower	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Wet
Initial temperature difference, F	24	29	34	39	44	49	39 [1]	N/A
Fan power, kWe	4,541	3,800	3,194	2,724	2,399	2,194	2,724	1,559
Cooling tower fan energy, MWhe	11,820	9,823	8,138	6,857	5,901	5,311	6,860	4,124
Net energy generation, MWhe	192,933	193,282	191,651	190,263	186,324	183,506	190,346	201,177
Annual solar-to-electric efficiency	0.1294	0.1296	0.1285	0.1276	0.1249	0.1231	0.1276	0.1349
Raw water use, m^3	57,451	57,140	56,485	55,957	54,972	54,285	55,974	742,368

Note 1: Maximum condenser pressure limited to 8 in. HgA

5. Economic Analysis

Levelized energy costs were calculated for the plants with dry and wet cooling using the year-by-year cash flow analysis within Excelergy. The input financial parameters to the model are listed in Table 8.

Table 8
Financial Parameters for Levelized Energy Cost Calculations

Parameter	Value
Interest during construction	
- Construction period, years	2
- Interest rate, percent	7
Operation and maintenance cost, $ million	
- Dry cooling [1]	4.720 to 4.793
- Wet cooling	4.778
Cost of water, $/1000 gallons	1.40
Effective income tax rate, percent	40.0
Debt financing	
- Interest rate, percent	6
- Period, years	20
- Minimum coverage ratio	1.4
- Nominal fraction of total investment, percent [2]	56
Investment tax credit, percent	10
Depreciation period, years	5
Equity financing	
- Required return, percent	15
- Nominal fraction of total investment, percent [2]	44
Discount rate, percent	
- Nominal	10.1
- Real	7.6

Notes:
1) Varies with the size and capital cost of the air cooled condenser
2) Actual value varies by plant, based on debt coverage ratio, depreciation schedule, investment tax credit, and equity financing requirements

The capital cost for the plant with wet cooling was developed from the default values in Excelergy. As a point of reference, the default estimate within Excelergy for 'General Balance of Plant and Cooling' was compared with an independent estimate of the wet heat rejection system from References 2 and 3. The results are shown in Table 9.

Table 9
Comparison of Wet Heat Rejection System Costs

	Excelergy	Refs. 2 and 3
General BOP and Cooling	$6,792,000	
Surface condenser		$1,650,000
Wet cooling tower		$1,316,000
Cooling tower basin		$553,000
Circulating water pumps		$60,000
Circulating water pipe		$81,000
Raw water well and well field		$496,000
Evaporation pond		$1,561,000
	-------------	-------------
Total	$6,792,000	$5,717,000

Assuming the Excelergy estimate includes balance of plant items other than the wet heat rejection system, such as a compressed air system, the two estimates are in reasonable agreement.

The capital costs for the seven plants with dry cooling were developed by subtracting the 'General Balance of Plant and Cooling' estimate from the default Excelergy values, and then adding the cost of the air cooled condensers. To this was added an allowance of $1 million for those items within 'General Balance of Plant and Cooling' which were not associated with the wet heat rejection system.

The annual operation and maintenance costs for the plant with both dry and wet heat rejection systems were developed from the default values in Excelergy.

The results of the levelized energy cost calculations are shown in Table 10.

Table 10
Levelized Energy Costs for Plants with Dry and Wet Heat Rejection Systems

Case	1	2	3	4	5	6	7	8
Type of cooling tower	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Wet
Capital cost, $ 1000	290,958	286,202	282,256	279,120	276,862	275,315	279,120	267,747
O&M cost, $ 1000	4,763	4,744	4,728	4,715	4,705	4,698	4,716	4,778
Levelized energy cost, $/kWhe	0.1400	0.1379	0.1375	0.1373	0.1393	0.1408	0.1373	0.1270
Energy cost penalty, percent	10.2	8.6	8.3	8.1	9.7	10.9	8.1	Base

Thus, the use of a dry heat rejection system imposes a nominal 8 to 9 percent penalty on the levelized cost of energy.

With further optimization efforts, the expected penalty could perhaps be reduced to values in the range of 7 to 8 percent. Potential improvements include the following:

- Reducing the capital cost of the air cooled condenser by optimizing the tube and fin geometry in conjunction with the design air velocity and the fan power demand

- Reducing the parasitic energy demand by optimizing a schedule for fan speed settings as a function of turbine output and ambient temperature.

As noted in Table 8, the cost of raw water is estimated to be \$1.40 per 1000 gallons. On a conceptual level, the cost for water could rise to the point where the cost of energy for a plant with wet cooling is equal to the cost of energy from a plant with dry cooling. A brief economic analysis shows the required cost of water to be \$14.80 per 1000 gallons, which is about a factor of 10 higher than current prices.

On a point related to the selection of the optimum initial temperature difference for the air cooled condenser, initial considerations might lead to the selection of a low value for the design initial temperature difference. The cost of energy from a solar project is higher than from a fossil-fired plant; thus, small approach temperatures for the heat exchangers should be justified. However, the capacity factor of a solar power plant without thermal storage is no higher than 28 percent. As a result, there are only a limited number of hours in a year in which the capital investment in the larger heat exchanger can be recovered. This characteristic, coupled with the limited number of hours in a year in which the ambient temperature exceeds 110 °F, leads to the selection of an air cooled condenser with a relatively high initial temperature difference, and relatively high turbine performance penalties on hot days.

6. References

1) GateCycle Program, Version 5.20, GE Enter Software, Inc. and the Electric Power Research Institute

2) Maulbetsch, John S., (Electric Power Research Institute, Palo Alto, California), "Comparison of Alternate Cooling Technologies for California Power Plants - Economic, Environmental, and Other Tradeoffs", Public Interest Energy Research, California Energy Commission, Report 500-02-079F, February 2002

3) Letter from Bill Powers (Powers Engineering, Los Angeles, California) to Kent Zammit (Electric Power Research Institute, Palo Alto, California), Subject: Comments on February 2002 CEC/EPRI Document - "Comparison of Alternate Cooling Technologies for California Power Plants - Economic, Environmental, and Other Tradeoffs", March 29, 2003

REPORT DOCUMENTATION PAGE	*Form Approved* *OMB No. 0704-0188*

1. REPORT DATE *(DD-MM-YYYY)*	2. REPORT TYPE	3. DATES COVERED *(From - To)*
July 2006	Subcontract Report	January 20, 2005 – December 31, 2005

4. TITLE AND SUBTITLE
Nexant Parabolic Trough Solar Power Plant Systems Analysis: Task 2: Comparison of Wet and Dry Rankine Cycle Heat Rejection

5a. CONTRACT NUMBER
DE-AC36-99-GO10337

5b. GRANT NUMBER

5c. PROGRAM ELEMENT NUMBER

6. AUTHOR(S)
B. Kelly

5d. PROJECT NUMBER
NREL/SR-550-40163

5e. TASK NUMBER
CP06.1301

5f. WORK UNIT NUMBER

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
Nexant, Inc.
101 Second St., 11th Floor
San Francisco, CA 94105-3672

8. PERFORMING ORGANIZATION REPORT NUMBER
LDC-5-55014-01

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Renewable Energy Laboratory
1617 Cole Blvd.
Golden, CO 80401-3393

10. SPONSOR/MONITOR'S ACRONYM(S)
NREL

11. SPONSORING/MONITORING AGENCY REPORT NUMBER
NREL/SR-550-40163

12. DISTRIBUTION AVAILABILITY STATEMENT
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

13. SUPPLEMENTARY NOTES
NREL Technical Monitor: H. Price

14. ABSTRACT *(Maximum 200 Words)*
Subcontract report by Nexant, Inc., regarding a system analysis comparing solar parabolic trough plants with wet and dry rankine cycle heat rejection.

15. SUBJECT TERMS
concentrating solar power; solar parabolic trough; solar; solar thermal electricity

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
Unclassified	Unclassified	Unclassified	UL		19b. TELEPONE NUMBER (*Include area code*)