Feasibility of a Hybrid Cooling System in a Thermal Power Plant

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Abstract: The feasibility of introducing a hybrid cooling system in a thermal power plant is investigated with an aim to reduce water use with a minimum impact on plant performance. A number of cooling systems have been modelled including existing evaporative cooling system taking into account of a wide variety of ambient conditions at full load. Water consumption and plant performance for all cooling options considered are calculated and compared. The results show that a significant amount of water can be conserved with a minimal impact on plant performance by implementing a hybrid cooling system in opposed to evaporative cooling tower. This study also reveals that, although a significant reduction in water use with minimum effect on plant performance is possible with hybrid cooling, it is not a financially viable option.

Keywords: Power station, Cooling, Sustainable water consumption, hybrid cooling

1 Introduction

Water sustainability is a very prominent topic in the Australian public at present and will be well into the future. The accelerating growth of the country's population, industry and the visible change in climate is having a large impact on the availability of water. With water storage facilities at their all time low and the increasing demand, water prices are looking to more than triple in the near future. Therefore, any opportunities to conserve water would be in the best interests of the public and financial interests of industries. One of the largest consumers of water is fossil fuelled power stations. This is mainly due to their cooling systems. The power station considered (the reference plant) in this study has 2 units of 350 MW each at full load. For the year 2004-2005 over 50% of the water consumed from the nearby dam of the reference plant was for evaporative cooling tower [1]. Currently the reference plant consumes about 12,500 ML a year in its operations, around 85% of this is used in its cooling tower. The plant is estimated to use 2.1kL per MW of power generated. About 300 litres of water in a second is released into the atmosphere as vapour that could be used for domestic water or agricultural irrigation and with the price of water looking to escalate, the cost to produce electricity will greatly increase. Therefore, there is potential to conserve water and reduce operating costs by using an effective method of cooling. This paper presents the thermal performance of the plant for dry, wet and hybrid cooling systems. Then, an appropriate cooling method on the basis of the overall output and reduced water consumption of the power plant is identified.

2 Cooling Methods

Indirect cooling or closed cycle cooling systems work through the use of two cooling fluids. The first cooling fluid is usually water or a glycol mixture which is re-circulated between the condenser and cooling tower. It removes the remaining heat from the condenser and transfers it to the cooling tower where the second cooling fluid, being air transfers the heat energy into the atmosphere. There are two basic methods of indirect cooling, wet (evaporative) cooling and dry cooling. Both methods use various techniques and arrangements to achieve the purpose of cooling the circulating/cooling water with varying degrees of performance [2].

2.1 Evaporative (Wet) Cooling Systems

Evaporative or wet cooling systems (Figure 1) are a specialised heat exchanger where water and air brought into direct contact with each other to remove waste heat from the plant and expel it into the atmosphere. This takes place by the hot water being sprayed to form a rain droplet like pattern through the cool upward moving air. However, due to evaporation taking place in the process the mass flow of cool water leaving the tower is not equal the mass flow of hot water entering the tower. This difference is replaced by the make up water. Depending on the size of the plant and weather conditions this can be a very water consuming operation [3]. In Queensland the evaporation rates for 350 MW cooling systems are typically around 1.8 litres of water per kWh of power generated. To put this in a better prospective this is about 630 kilo litres an hour or 5500 ML per annum for a 350 MW plant [4].

Evaporation is the major source of water loss in evaporative cooling towers. However, it is not the only cause. Other causes of water loss are the blow down water, drift and blow out. The blow down water is used to clean any fouling, mineral deposits and impurities left in the tower after evaporation has occurred. This is done to maintain tower performance. Drift loss are water droplets that are carried out of the tower by the exhaust air and blow out loss are water droplets that are carried out of the inlet of the tower in windy conditions or by splashing and misting [5]. There are two common types of evaporative cooling tower systems; Natural draught cooling towers and Mechanical draft cooling towers. The selection of one or the other depends on the climatic location and the argument between the initial capital costs and the operational and maintenance costs.

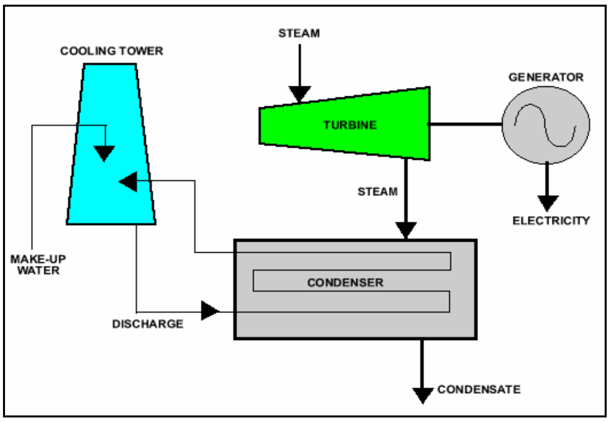


Figure 1: Schematic diagram of natural draft evaporative (wet) cooling system [4]

The Natural draft cooling relies on its hyperbolic nozzle shape and the density differential of the hot less dense air inside the tower and the cooler more dense air out side the tower. This is known as the Chimney Effect and the tower induces its own natural draft not relying on fans to mechanical induce it. Therefore operational and maintenance costs are low. However, the tower performance is better suited to cool, humid climates [3]. The mechanical draft cooling relies on large axial fans to induce air flow through the circulating water mist. Advantages of this system are that it can provide better controlled performance over a wide range of weather conditions. As a result lower cold water temperatures are available then a natural draft cooling tower on hot dry weather conditions. However, the down side is that power has to be put into system which increases operating costs. For a 450 MW station 1.5 – 2 MW are required to run the fans [4].

2.2 Dry Cooling Systems

Indirect dry cooling systems (Figure 2) work on the same principles as the air cooled (AC) condensers (like a large car radiator) or direct dry cooling [3]. In this case the circulating water is cooled instead of the steam. This system has the advantage over the AC condenser system in that it can be away from the turbine hall. It does not have to be as close to the turbine exhaust as possible. As with the indirect wet cooling systems previously discussed, there are two common types of dry cooling tower systems; Natural draught cooling towers and Mechanical draft cooling towers. Again the selection of one or the other depends on the climatic location and the argument between the initial capital costs and the operational and maintenance costs. Dry cooling is not as efficient as wet cooling. Indirect wet cooled system condensers usually have a backpressure in the range of 6 – 12 kPa and indirect dry cooled system condensers usually have a backpressure in the range of 15 – 20 kPa in their economic range and can see up to 50 kPa in hot conditions [6].

Trying to achieve better efficiency in dry cooling systems is often very expensive as the heat exchanger area is quite large a many large fans are required to meet operating conditions. The size of the cooling area varies inversely with the log mean temperature difference (LMTD) [6].

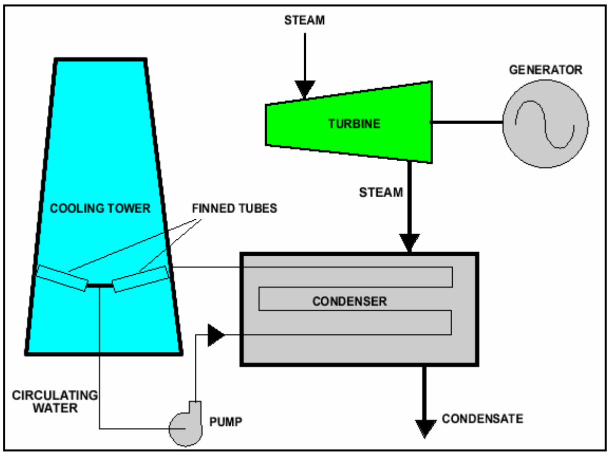


Figure 2: Schematic diagram of natural draft dry cooling tower [4]

2.3 Hybrid Cooling

The ultimate objective of the cooling system is to provide the condenser with the lowest possible sustainable temperature to archive the most economic turbine exhaust backpressure with the seasonal variations in the ambient temperature and relative humidity [2]. This study investigated the idea of combined wet and dry indirect cooling systems to utilise the advantages of both systems otherwise known as hybrid cooling. During periods of peak load and peak climate conditions a significant amount of heat is removed by sensible and evaporative heat transfer with hybrid cooling, this reduce the water consumption over conventional evaporative cooling. When the heat load and/or ambient conditions drop from peak design, water consumption is further reduced in the hybrid cooling system by regulating the load on the dry and wet units. A balance can be achieved by putting a greater load on the dry cooling unit and reducing the load on the wet cooling unit and if conditions permit water consumption can be totally eliminated by operating the dry cooling at 100% load [6]. There are two ways to maintain variations in hybrid cooling. They are parallel hybrid and series hybrid.

Parallel hybrid cooling works by the dry cooling unit being connected into the hot water section of the cooling water (CW) system at a tee piece before the inlet of the wet tower (Figure 3). The water cooled by the dry cooling unit is then mixed back with the cold water in the wet tower pond and should only slightly rise the overall cold water temperature. However, as this system decreases the flow in a natural draft (ND) cooling tower, the ND tower's cooling range will be widen giving a lower cold water temperature out of the wet cooling unit.

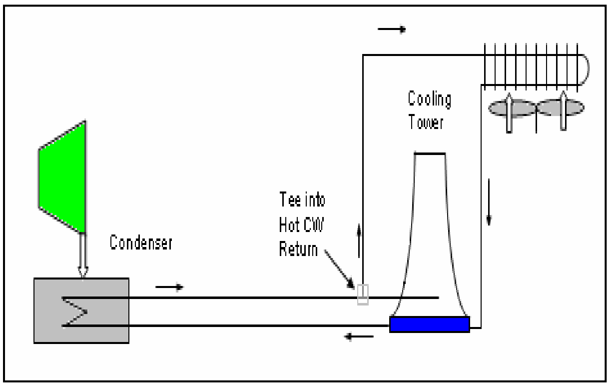
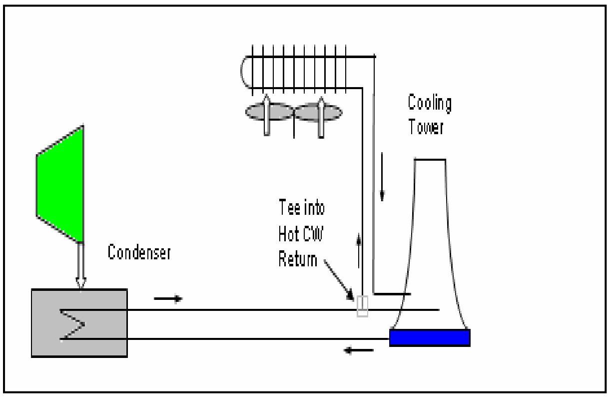


Figure 3: Parallel hybrid system (modified from [6])

In the series hybrid cooling system (Figure 4) the water cooled by the dry cooling unit is then mixed back with the hot water before the wet towers sprayers giving a cooler hot water temperature entering the cooling tower. The decrease in inlet water temperature in a ND cooling tower should theoretically increase the ND tower's cooling range resulting in a lower cold water



temperature out of the wet cooling unit.

Figure 4: Series hybrid cooling (modified from [6])

2.4 Factors Affecting Plant Performance

The exhaust pressure of the low pressure turbine strongly affects the efficiency of the steam cycle. Condenser backpressure determines the saturation temperature at which the expanded steam rejects its latent heat of vaporisation to the cooling system. Therefore, changes in backpressure affect the temperature of cycle heat rejection. This phenomenon of the effect of change in condenser saturation temperature on the work done in the cycle is displayed in Figure 5. Generally for higher cycle efficiency, a low exhaust pressure is sought after [2]. Therefore, as condenser backpressure is dependent on the cold CW temperature, the performance of the cooling system can have a large impact on the overall plant performance.

3 Modelling of the Cooling Systems

The following assumptions were made for modelling:

- Air exiting the natural draft cooling tower to be saturated.
- Hot CW = 40.1 ^{o}C
- Condenser design LMTD = 8.11
- Design HR = 15,996 kJ/kg
- Coal GHV = 20 MJ/kg
- Local atmospheric pressure = 98.5 kPa.

The modelling has been done to model the installed evaporative cooling system (base model) and the variations of series and parallel hybrid systems at ratios of 25%, 50% and 75% dry so a comparison can be made on the potential benefits and assess which system best suites the reference plant's operating conditions, objectives and budget.

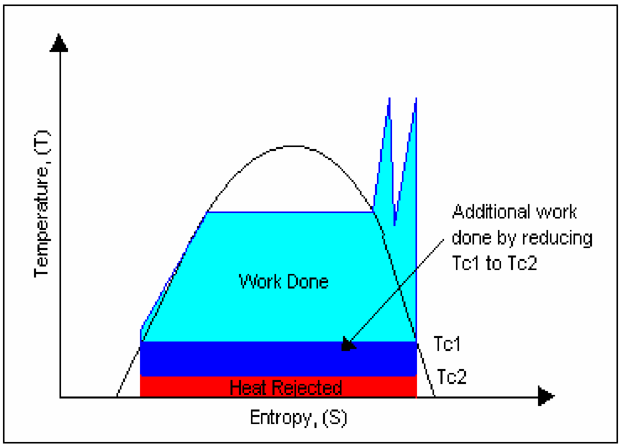


Figure 5: Steam cycle

3.1 Base Model

The base model of the evaporative cooling system was developed to verify the predicted results with the plant manuals and physical operating conditions. The results of the base model also served as the set point to which both the series and parallel hybrid systems results could be compared. The base model initially uses the plant's cooling tower performance curves from the plant manual to predict the re-cooled CW temperature for a given ambient and hot CW conditions [5]. The model then predicts the evaporation rate of the cooling tower. This is dependant on the draft and the humidity ratios of the air entering and exiting the tower, and given by [7],

$$Evaporation\ rate = G(1-\omega_1)(\omega_2-\omega_1) \qquad (1)$$

Where G is the air mass flow rate (kg/s), ω is the humidity ratio, and 1 and 2 denotes inlet and exit.

The ambient conditions, the properties of the air entering and exiting the tower, were found by assuming the exiting air to be saturated air and the mass energy balance equation given below [8],

$$L(h_{L1} - h_{L2}) = G(h_{G2} - h_{G1}) \quad (2)$$

$$G = \frac{L}{(L/G)} \quad (3)$$

$$(L/G) = \frac{L}{\sqrt[3]{HLRf}} \quad (4)$$

$$f = \frac{\Delta\rho}{(164 \times 10^{-8})\Delta h} \quad (5)$$

Where, *f* is the draft factor, H is the cooling tower height and R is the cooling tower range.

From the evaporation rate the blowdown rate was calculated by [6],

$$Blowdown = \frac{Evaporation\ rate}{(n-1)} \quad (6)$$

Where *n* is the number of CW cycles.

The make up flow rate is equal to the evaporation rate plus the blowdown rate plus drift losses. This is the water consumption of the cooling system. The model calculates the final re-cooled CW temperature assuming adiabatic mixing of the re-cooled CW and make up, and given by [7],

$$m_3 h_3 = m_1 h_1 + m_2 h_2 \quad (7)$$

where m is the mass flow rate of make-up water (kg/s).

From the final re-cooled CW temperature the condenser backpressure was calculated. The design LMTD = 8.11 for the reference plant's condensers [6]. As the inlet CW temperature varies so does the LMTD and can be expressed by,

$$LMTD_{corrected} = 8.11 \sqrt[4]{\frac{33.45}{\left(\frac{(T_1 + T_2)}{2}\right)}} \quad (8)$$

From the corrected LMTD the steam saturated temperature could be solved by,

$$LMTD_{corrected} = \frac{12.5}{2.3 \log \frac{(T_3 - T_1)}{(T_3 - T_2)}} \quad (9)$$

The steam saturated temperature was then used to find the condenser backpressure from the steam tables. The base model then uses the plant turbine exhaust correction curves from the plant manual to predict the change in heat rate (HR) [5]. The corrected heat rate was then found by,

$$HR_{corrected} = HR\left(1 + \frac{\Delta HR}{100}\right) \quad (10)$$

The corrected coal rate was found by,

$$Coal\ Rate_{corrected} = \frac{HR_{corrected}}{GHV_{coal}} \quad (11)$$

The prediction of the base model was compared with the on-site measured data. The results showed good agreement with the measured date.

3.2 Parallel Hybrid Model

The parallel hybrid model is a combination of the installed evaporative cooling system and the proposed dry cooling system which uses banks of 5MW air cooled heat exchangers (ACHE). A proportion of the hot CW is diverted through the dry cooling system and re-enters at the evaporative cooling tower's pond where the two re-cooled CW's and makeup water mixes in together. The *first step* of setting up parallel hybrid model was to predict the re-cooled CW temperature and the evaporation rate for the natural draft evaporative cooling tower at any ambient condition and reduced CW flow rate. The *second step* was to predict the re-cooled CW temperature for the dry cooling system. The predicted re-cooled CW temperature of the dry cooling system is dependant on the properties of the cold fluid (ambient air), the hot fluid (hot CW) and the properties of the ACHE. Three equations were used, that being the mass, energy balance equation for the cold and hot fluids and the heat transfer equation for the ACHE. All three equations were solved simultaneously to determine the exit conditions of both the cold and hot fluids,

$$LC_{p\ water}(T_1 - T_2) = GC_{p\ air}(t_2 - t_1) = UA \times LMTD \quad (12)$$

Where, T is the water temperature (^oK), t is the air temperature (^oK), C_p is the specific heat value (kJ/kg K), U is the heat transfer co-efficient (W/m^2K) and A is the area of heat exchanger (m^2).

The *third step* was to calculate the combined re-cooled CW temperature after the mixing of the evaporative re-cooled CW, the dry re-cooled CW and the make up. This was done again assuming adiabatic mixing of fluids as in base model. The *final step* of the parallel model, calculating the condenser backpressure and corrected heat rate, was performed the same way as in the base model.

3.3 Series Hybrid Model

The series hybrid model which is also a combination of the installed evaporative cooling system and the proposed dry cooling system which uses banks of 5MW air cooled heat exchangers. Again a proportion of the hot CW is diverted through the dry cooling system. However, in this case the dry re-cooled CW re-enters the hot CW before it enters the evaporative cooling tower. The *first step* of setting up series hybrid model was to predict the re-cooled CW temperature for the dry cooling system. This was carried out the same way as the second step of the parallel hybrid model. The *second step* was to calculate the re-cooled hot CW temperature before it entered the evaporative cooling system. This was done again assuming adiabatic mixing of fluids as in step three of both the base and parallel hybrid models. However, this was calculating the mixing of the dry re-cooled CW with the remaining hot CW. The *third step* was to predict the re-cooled CW temperature and the evaporation rate for the natural draft evaporative cooling tower at any ambient condition and re-cooled hot CW temperature. The *fourth step* of the series model was to calculate the re-cooled CW temperature combined with the make up temperature. The *final step*, calculating the condenser backpressure and corrected heat rate, was performed the same way as in the base model.

3.4 Ambient conditions

The design ambient conditions for the reference plant were 21.6° DBT, 17.5° WBT. This design point appears to have been determined from the annual average 9:00am conditions for local area when compared to the data from the bureau of meteorology [9]. This design ambient condition has been used for the modelling and comparisons of the hybrid systems as it was the year long average and design point of the existing system. However, average monthly ambient conditions were also modelled to demonstrate the performance at different times of the year and also demonstrate potential optimisation of the hybrid systems at different times of the day. The monthly average ambient conditions were the average mean 9:00am and 3:00pm dry and wet bulb temperatures for the month. For the different times of the day the daily temperature profiles. These points were used to create a sinusoidal wave to represent the daily temperature profile for a given month.

4 Results and Discussion

4.1 Coal & Water Consumption

The two major inputs to the operation of a coal fired power station are water and coal. From the modelling of the proposed hybrid systems it can be seen that there is a reduction in both of these inputs. There are significant reductions possible in the amount of water consumed for both parallel and series methods considering the existing systems consume around 12,000 ML/yr. However, from Table 1 it can be seen that series hybrid cooling conserves more water, although as the proportion of dry cooling increases this margin between series and parallel methods decreases.

Dry cooling systems are most effective at night and in winter when temperature differentials from hot CW to the inlet air temperatures are well over 20°C. While the peak day differences are much less, the daily average water conservation provides the achievable savings. There are also slight reductions possible in the amount of coal consumed for both parallel and series methods considering the existing systems consumes around 5 million tonnes per annum.

Table 1: Water reduction compared to base case

Ratio of Dry Cooling	Water (%)		Water (ML/yr)		Water (kL/MW)	
	Parallel	Series	Parallel	Series	Parallel	Series
25%	16%	23%	1839	2681	0..29	0.43
50%	38%	45%	4370	5179	0.71	0.84
75%	64%	65%	7381	7472	1.20	1.21

However, from Table 2 it can be seen that parallel hybrid cooling consumes less coal, although as the proportion of dry cooling increases this margin between series and parallel methods decreases. Coal consumption is directly proportional to the heat transfer rate of the plant and the efficiency of the turbine. The efficiency of the turbine can change with varying CW temperatures that rely on the ambient temperature. As a result, if the CW temperature is higher, the coal consumptions is higher. If the CW temperature is lower the coal consumption is lower. Therefore, the slight reductions in coal consumption are due to the slight reductions in CW temperature.

Table 2: Coal Reduction

Ratio of Dry Cooling	Coal (%)		Coal (t/yr)		Coal (kg/MW)	
	Parallel	Series	Parallel	Series	Parallel	Series
25%	0.16%	0.04%	7968	1897	1.30	0.31
50%	0.21%	0.09%	10176	4585	1.66	0.75
75%	0.18%	0.17%	8637	8158	1.41	1.33

4.2 Plant Performance

The performance of a cooling system in a coal fired power station is important as it has an impact on the whole stations performance. If its performance is lacking the end results are either more energy is needed to be put in (extra coal burnt) or loss of generation (reduced load). In this case there is a slight improvement in the performance of the cooling system for both of the proposed parallel and series methods, which has a flow on affect on the stations performance. From Table 3, it can be seen that parallel hybrid cooling has the greater improvement in performance although as the proportion of dry cooling increases this margin between series and parallel methods decreases.

Although there is slight improvement in station performance for the proposed hybrid cooling systems, this is out balanced by the requirement of additional auxiliary power to run the large number of fans on the dry cooling systems and the booster pump required for the series hybrid system. The additional auxiliary power required is shown in Table 4.

Table 3: Plant Performance

Ratio of Dry Cooling	Re-cooled CW Temperature (°C)			Condenser Backpressure (kPa)			Heat Rate (GJ/MWhr)		
	Exis.	Para.	Seri.	Exis.	Para.	Seri.	Exis.	Para.	Seri.
0%	28.67	-	-	9.34	-	-	16.04	-	-
25%	-	27.63	28.44	-	8.87	9.23	-	16.014	16.034
50%	-	27.30	28.09	-	8.73	9.07	-	16.007	16.026
75%	-	27.53	27.60	-	8.83	8.86	-	16.012	16.014

Table 4: Additional Auxiliary Power

Ratio of Dry Cooling	Additional Auxiliary Power (MW)		Additional Auxiliary Power (%)	
	Parallel	Series	Parallel	Series
25%	4.97	5.47	0.71%	0.78%
50%	9.94	10.44	1.42%	1.49%
75%	15.01	15.51	2.14%	2.22%

As a result of the additional auxiliary power required for the proposed hybrid systems there will be a loss of profitable generation from the existing system. The Loss of profitable Generation is shown in Table 5.

Table 5: Loss of Profitable Generation

Ratio of Dry Cooling	Loss of profitable Generation (MW/yr)	
	Parallel	Series
25%	43521	47901
50%	87042	91422
75%	131512	135892

However these losses of generation could be rectified by upgrading the LP turbines, giving overall a 3% increase of the turbine generator efficiencies. As a result the power station would then have a generating output of 2 x 360 MW units. This would cover the additional auxiliary power requirement and leave about an additional 5MW of profitable generation.

5.3 Financial Analysis

From the models, cost analysis calculations were conducted by using the Net Present Value (NPV) method to work out if any of the proposed models are a financially viable option. Due to confidentiality requirements no financial values are disclosed in this paper. Assuming a station life of 25 years from now, it was found that none of the proposed variations of parallel or series hybrid cooling was financially viable. This is mainly due to the high initial capital required and the small annual payback of reduced water consumption. For any of these options to become financially viable there would have to be a significant increase in the cost of water and electricity. By spending additional initial capital in upgrading the LP turbines, it is more financially viable then using the existing turbines as the loss in profitable generation. However, it is not enough to have the proposed hybrid systems paid off in the life of the station.

6 Conclusions

Theoretically a considerable reduction in water consumption with a minimal impact on plant performance is possible and there is potential optimisation of the parallel hybrid system to further reduce water consumption. However, due to the large number of fans on the dry cooling system additional auxiliary power is required causing a loss in profitable generation. The loss in profitable generation can however be overcome by upgrading the LP turbines changing the power station to 2 x 360 MW units. Even with the LP turbine upgrade, hybrid cooling is not a financially viable option for the reference plant.

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