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FINAL REPORT

U.S.DOE NETL Award DE-FE0031833
GTI ENERGY PROJECT NUMBER 22654

Enhanced Cooling Tower Technology for Power Plant Efficiency Increase and Operating Flexibility

Reporting Period:

October 1, 2019 through September 30, 2022

Report Issued:

December 26, 2022

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File Name: Final Report DE-FE0031833 (GTI Energy 22654)

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Acknowledgement

The authors acknowledge a generous support of U.S. Department of Energy's National Energy Technology Laboratory as well as Utilization Technology Development NFP (UTD) and Sustaining Membership Program (SMP) towards to research and concept design development of this Enhanced Cooling Tower technology.

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Executive Summary

The Final Report describes the results of the work completed during the "Enhanced Cooling Tower Technology for Power Plant Efficiency Increase and Operating Flexibility" project. The objectives of work were to develop and demonstrate the prototype (up to 100 kWth) of an economically viable "all weather" Sub-Dew Point Cooling Tower (SDPCT) with inlet air cooling and dehumidification including testing in laboratory-controlled environment for simulating various ambient conditions. The inlet air is cooled and dehumidified by a Pressure Dehumidifying System (PDHS) installed at the cooling tower inlet.

To achieve the project objective, the following activities were performed:

- Refinement of the proposed SDPCT concept as per typical coal power plant and preliminary performance characterization of the individual components and advanced techniques.
- Design, preliminary engineering and manufacturing review of SDPCT prototype for efficient heat and mass transfer with minimized air pressure drop.
- Multi-level design optimizations at component, system and power plant levels through thermo-fluid and energy flow modeling – Phase One.
- Detailed engineering, fabrication, purchasing and demonstration of a PDHS prototype under various simulated ambient conditions at GTI Energy test facility – Phase Two.
- SDPCT performance characterization by Baltimore Aircoil Company's (BAC) in-house modelling software based on the PDHS prototype test results – Phase Two.
- Techno-economic assessment (TEA) and sensitivity studies by Worley/Edvisian to guide design optimizations – Phase Three.

During the implementation of the project, the following results and findings of the project were obtained.

- The proposed concept of the SDPCT for a coal power plant was refined. According to the concept, the sub-dew point water temperature at the outlet of the cooling tower is achieved by using a PDHS. The PDHS precools and dehumidifies ambient air supplied to the SDPCT. The air cooling and dehumidification is accomplished by a near-atmospheric pressure regulation technique and efficient heat exchange components (patent pending). The air in the PDHS is slightly pressurized (10-20% above the ambient atmospheric pressure), and the air dew point is increased, thus making it easier to remove moisture from the air. The use of an expander in the PDHS compensates for the power consumed by the blower.
- The pilot-scale PDHS prototype has been tested at GTI industrial laboratory. Various process temperatures and flow rates were simulated in the pilot-scale PDHS test. Three selected representative cities of the U.S. major climate zones for coal power plants and the range of the wet bulb temperature were used in the experiments.
- Experiments have confirmed that the cooled air is dehumidified when the air temperature falls below the local dew point. Estimated water savings in the cooling tower, estimated from the results of experiments, amounted to more than 20%.
- The measured coefficient of performance (COP) of the pilot-scale PDHS prototype ranged 2.0 and 12.0 depending on ambient conditions.

The PDHS mathematical model developed by GTI has been incorporated into BAC's in-house modeling software to simulate PDHS enhanced cooling towers (SDPCT). The computational analysis carried out by BAC, based on the experimental data, confirmed the following:

- water savings 20%-33% for all selected climate conditions
- up to 100% water savings is achievable at favorable conditions
- higher dehumidification rate at higher blower pressure ratio
- sub-dew point cooling: 2.5°F - 4.2°F below ambient dew point was achieved
- COP: up to 4.35
- cooling tower with the PDHS can run at much lower temperature of cold water (45°F - 70°F) compared to the cooling tower without the PDHS (80°F - 91°F)

In the Techno-Economic Assessment (TEA), two SDPCT cases were compared with the referenced 650MW coal power plant. The following findings were obtained:

- 0.36-1.06% net plant efficiency gain
- makeup water reduction leads to reduced water cost saving of 47.1 $/MWh without condenser upgrade and 59.6 $/MWh with condenser upgrade (from 1-pass to 2-pass condenser design)
- CAPEX is 104.1 $/MWh for PDHS retrofit without condenser upgrade and 115.6 $/MWh with condenser upgrade
- LCOE is 443.9 $/MWh without condenser upgrade and 149.8 $/MWh with condenser upgrade.

Chapter 1 Introduction

Background

The U.S. Department of Energy (DOE) has been supporting technologies that can enhance the performance and economics of future new coal power plants. For Fiscal Year 2019, DOE solicited applications to develop technologies that will enhance the performance and cost-effectiveness of coal-based power generation.

The DOE National Energy Technology Laboratory (NETL) has an extensive program looking into more efficient water use at power plants [1]. Research is desired that lowers the overall water usage and/or impact on water quality from power plants through advances in cooling technology. Emphasis is placed on near-term solutions that have the potential to assist existing coal-fired power plants to operate in a more water efficient way. New methods are needed to be economically viable in the near term.

One Area of Interest (AOI) of the DOE solicitation includes Coal Power Plant Cooling Technology with lower cost, higher performance, and decreased water consumption. This AOI supports the Water Security Grand Challenge, a White House initiated, U.S. Department of Energy led framework to advance transformational technology and innovation to meet the global need for safe, secure, and affordable water. Technologies are requested that can enhance the flexibility, efficiency, and maintainability of existing recirculating cooling towers. These technologies should enhance the tolerance of fossil power generation to reduced water availability scenarios such as droughts or competing needs like agriculture.

The main objective of this project is to develop a coal power plant cooling technology that provides higher performance of a power plant, decreases water consumption, enhances flexibility and improves the efficiency of existing recirculating cooling towers by precooling and dehumidifying air and controlling parameters of the air under cyclic and part-load operation.

The initial objective of the project efforts is to complete preliminary process design, modeling, and laboratory experimental equipment design. The subsequent objective is to assemble the laboratory equipment and develop the project test plan. The final objective is to conduct laboratory testing, complete data analysis, and conduct a techno-economic analysis (TEA) including scale-up recommendations.

The goal of this project is to demonstrate the prototype (up to 100 kW) of an economically viable "all weather" Sub-Dew Point Cooling Tower (SDPCT) with inlet air dehumidification including testing in laboratory-controlled environment for simulating various ambient conditions.

Sub-Dew Point Cooling Tower Technology (SDPCT) Initial Concept

Improving the efficiency of power plant cooling towers provides immediate benefit to the overall plant. Specifically, increasing the cooling tower efficiency decreases the evaporative water loss, while simultaneously increasing the net power from the same amount of coal (or natural gas) combusted in the power plant. Cooling towers are a type of heat exchanger combined with evaporation. In that context, modern cooling towers are as close to maximum efficiency as possible with approach temperatures of 5 to 7°F. New cooling tower approaches with closer approach temperatures and thus higher efficiency have been proposed, but they require complex retrofits, high energy costs, or a full replacement of the cooling tower with a new unit. Among the new approaches is the M-cycle dew point cooling tower fill [2], which has been studied by EPRI and GTI Energy, and hybrid cooling technology, e.g., thermosiphon cooler studied by EPRI and Johnston Controls. The project team proposed to demonstrate and characterize the performance of a sub-dew point cooling tower enhancement technology. The process is described below. No modifications are required to the existing cooling towers, no new materials are required to develop or obtain. The proposed enhancement assumes using available off-shelf commercial hardware installed outside the cooling tower or partially integrated with the structural design. These features and flexible operation lower technical risk and improve the chances of early adoption. The plant modification will be easy to install. To achieve higher cooling tower efficiency, the proposed enhancement technology changes how ambient air is introduced into the cooling tower. Instead of the usual practice of introducing ambient air directly into the cooling tower, the ambient air is now pre-cooled and dehumidified. This leads to lower water temperatures, thus harvesting water from ambient air that directly reduces the make-up water and, with better cooling, offers higher power generation efficiency reducing evaporative losses.

The proposed sub-dew point cooling tower technology (patent pending) employs an innovative flow arrangement called a Pressure Dehumidifying System (PDHS) coupled with effective and efficient heat and mass transfer so the air is cooled and dehumidified prior to entering the cooling tower fill (Figure 1). The air cooling and dehumidification is accomplished by a near-atmospheric pressure regulation technique and efficient heat exchange components. The blower in the system slightly pressurizes the incoming air (15-20% above the ambient atmospheric pressure) and increases the air dew point, thus making it easier to remove moisture from the air. The expander is used to offset the power consumed by the blower, making this a low power system.

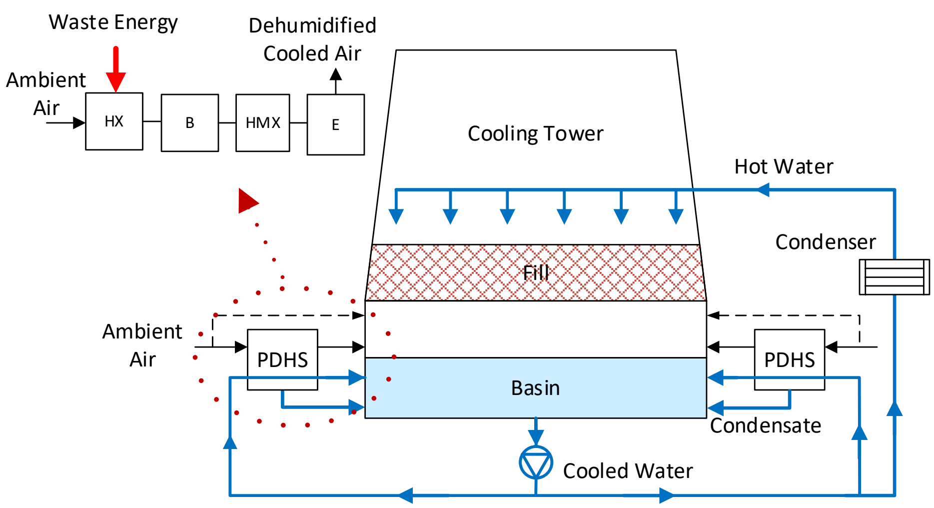


Figure 1. Sub-dew point cooling tower technology approach: PDHS – Pressure Dehumidifying System, HX – Heat Exchanger, B – Blower, HMX – Heat-Mass Exchanger, E - Expander (initial concept from proposal)

Figure 2 shows the full implementation of the sub-dew point cooling tower technology employing the PDHS in a coal power plant. It provides a comparison of process parameters, water evaporation savings, and power increase for one ambient air condition and the same coal feed rate. By implementing the advanced technology to retrofit a cooling tower, the plant efficiency is increased by ~3%, makeup water is reduced by more than 20%, and evaporation losses are reduced.

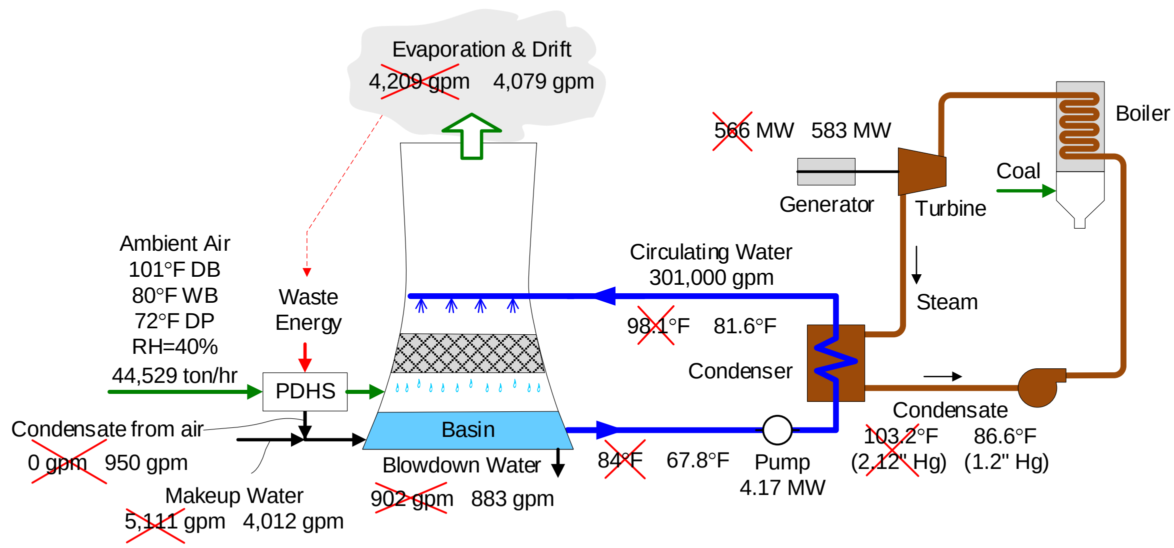


Figure 2. Sub-dew point cooling tower technology for a 550 MW coal power plant: Pressure Dehumidifying System (PDHS)

Figure 3 provides calculated net power gain as a function of ambient air temperature. The highest benefit to net power production is achieved at the highest ambient air temperature and lowest relative humidity.

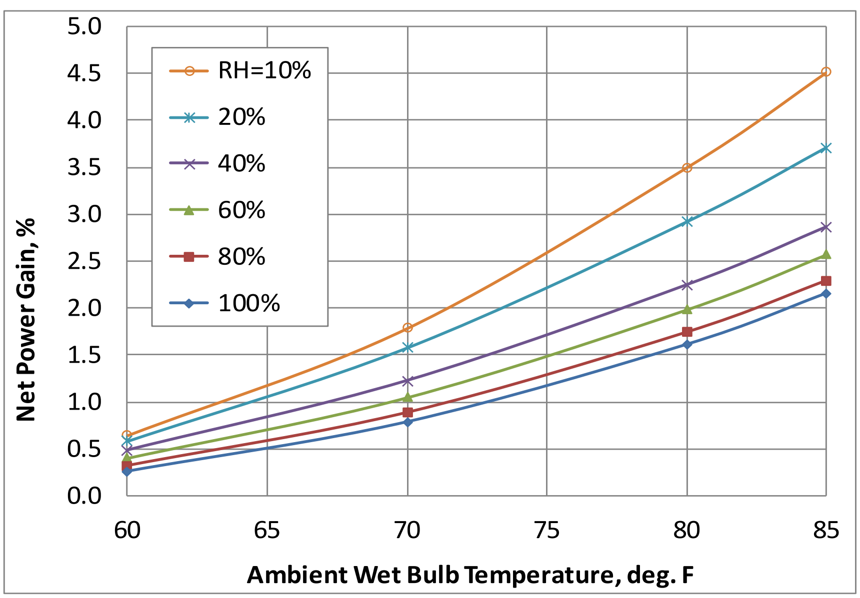


Figure 3. Net power gain by the new technology for a 550 MW coal power plant with 4-flow LP turbine (design conditions: TCW=83°F, TWB =78°F, 5°F approach, 14°F range, 301,000 gpm water flow rate)

The uniqueness of the SDPCT technology is using the reverse Brayton cycle (or Brayton refrigeration cycle) for deep cooling of ambient air prior to entering the cooling tower. The Brayton refrigeration cycle includes an air blower with a motor, turbo-expander, ambient air as a heat source, and cold water as a heat sink. Since the temperature difference between the heat source (ambient air) and heat sink (cold water) in the cycle is very low, the cycle efficiency can be extremely high. For example, if ambient air temperature is 86°F, cooling the air to ~69°F by the Brayton refrigeration cycle provides cooling with coefficient of performance COP~30, which means ultra-low energy consumption of the cooling system. Moreover, cooling the air after the blower allows for deeper air cooling due to the Brayton refrigeration cycle. Low-grade waste energy from the cooling tower can be used to cool the air.

Another unique SDPCT feature is simultaneous cooling and dehumidification of the ambient air. The air cooling and dehumidification is accomplished by a near-atmospheric pressure regeneration technique and efficient heat exchange components with ultra-low energy and water consumption. The blower in the system slightly pressurizes the incoming air and increases the air dew point making for easier moisture removal from the air. An expander offsets the power consumed by the blower thus enabling low energy consumption. Using the near-atmospheric pressure regeneration technique allows moisture removal from the incoming ambient air to lower absolute humidity. This is below the ambient level and it harvests the water from ambient air and using that water to partially compensate (up to 20% depending on the ambient humidity) makeup water.

Cooling the air downstream of the blower with ambient air or cold cooling tower exhaust (when the exhaust air temperature is below ambient temperature) will cool the air with the pressure regeneration technique resulting in deep cooling of water. Using the cooling tower exhaust is a possible scenario when the ambient air is efficiently cooled and dehumidified by the PDHS. Indirect cooling and dehumidifying of the air, by using heat-mass exchanger, further reduces the air temperature to below ambient air dew point temperature (T_{DP}), achieving cooled water temperature (T_{CW}) < ambient T_{DP}.

The proposed innovative waste heat assisted air cooling and dehumidification technique will significantly lower fresh water consumption in coal power plants due to lower water temperature. It is also expected not only to break the paradigm of a cooled water temperature limit of 5°F above the ambient wet bulb. Cooling the water below the ambient dew-point (TDP) leads to significantly increased power plant net efficiency through inlet air dehumidification and higher performance.

By splitting the intake ambient air between conventional and dew-point paths in the dew-point cooling tower (Figure 1) and controlling the air flow rates ratio, this will enhance the cooling tower flexibility and increase plant efficiency under cyclic and part-load operation.

SDPCT Advancement

Figure 4 shows the calculated psychrometric charts of air in a conventional cooling tower (line 1-7) and the proposed advanced cooling tower (line 1-6) for comparison. At this example condition, water is cooled to 84°F in a conventional cooling tower and to 67.8°F in the advanced cooling tower, below the dew point of the ambient air.

Figure 2 shows a simplified plant integration diagram for a 550 MWe power plant comparing SDPCT and conventional cooling tower estimated parameters. The same circulating water flow rate was chosen for SDPCT to compare with the conventional cooling tower. Lower circulating water temperature (by ~16°F) with SDPCT leads to ~3% turbine power gain. Approximately 20% less makeup water is used by SDPCT because condensate from air dehumidification in the PDHS.

The calculations show the PDHS provides more efficient air cooling and dehumidification at higher ambient air temperature and higher humidity. Extracting more water vapor from humid air and provides deeper cooling due to the reverse Brayton cycle features, while the energy consumption to blow the air and dehumidify it is low due to a high COP of the cycle.

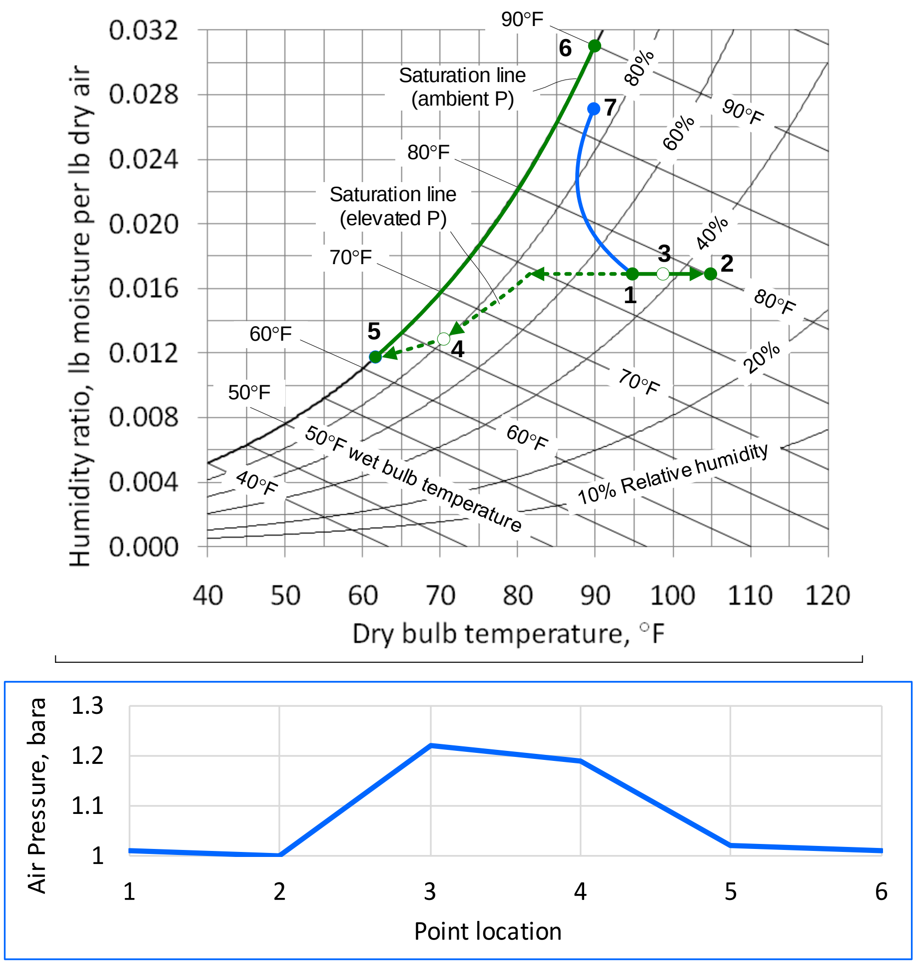


Figure 4. Example psychrometric chart of conventional cooling tower (line 1-7) and advanced cooling tower (1-2 – adiabatic compression by blower, 2-3 – sensible cooling, 3-4 – latent cooling and dehumidification, 4-5 – adiabatic expansion, 5-6 – direct atmospheric cooling). Solid and dashed lines are at different air pressure.

Figure 5 shows predictions of plant net power gain for SDPCT at 40% relative humidity compared to the typical conventional cooling tower at the same design conditions for a 550MWe capacity coal fired power plant. Based on the assumptions and assuming the research meets the goals previously described, the SDPCT is shown to be more efficient compared to the typical cooling tower. The SDPCT provides about 2.8% plant efficiency gain.

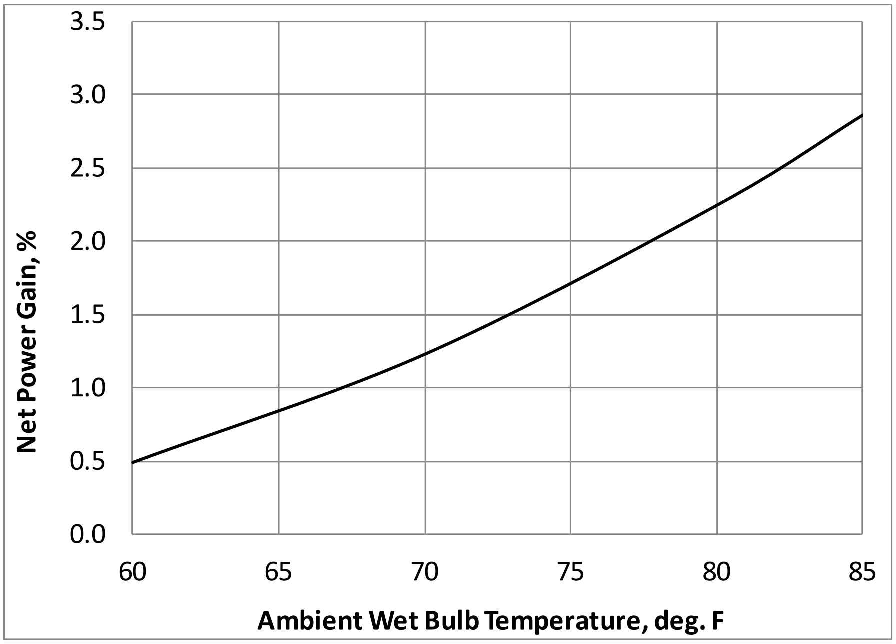


Figure 5. Net power gain by SDPCT for a 550MW coal power plant with 4-flow LP turbine (Design conditions: T_{CW}=83°F, T_{WB} =78°F, 5°F approach, 14°F range, and 301,000 GPM water flow rate)

Feasibility of Approach

The project advances the sub-dew point cooling tower technology from concept to readiness for demonstration. A highly qualified and experienced team (GTI Energy, Baltimore Aircoil Company (BAC), Worley/Advisian, Illinois Institute of Technology (IIT) with engagement of industrial consultants and utility partners) has been assembled to meet the project goals and objectives. This team has worked well together in the past and has the needed skills, specialists, and facilities needed to complete the Tasks on time and on budget. The Scope of Work is carefully constructed to provide a logical progression of activities and to utilize individuals and facilities at rational levels of commitment. The Project Management Plan (PMP) describes means included to identify and mitigate risks. Reports and go-no-go decision points are strategically placed to allow sound oversight of all project activities. This will help keep the project on track. No new materials or unproven equipment is required to complete testing and the team has access to all needed computer facilities.

Statement of Project Objectives

GTI Energy, BAC, Worley-Advisian (Worley), with academic support of IIT and engagement of industrial consultants and utility partners, carried out the Three Phase project to develop, and demonstrate the prototype (up to 100 kWth) of an economically viable "all weather" SDPCT with inlet air dehumidification including testing in laboratory-controlled environment for simulating various ambient conditions. The Phase One objective is to complete preliminary process design, modeling, and laboratory experimental equipment design. The objective of Phase Two is assemble the laboratory equipment and develop the project test plan. The Phase Three objective is to conduct laboratory testing, complete data analysis, and conduct a techno-economic analysis including scale-up recommendations.

To achieve the project objective, the following activities were performed:

- Refinement of the proposed concept as per typical coal-fueled power plant and preliminary performance characterization of the individual components and advanced techniques – Phase One
- Design, preliminary engineering and manufacturing review of SDPCT-prototype for efficient heat and mass transfer with minimized air pressure drop – Phase One
- Multi-level design optimizations at component, system and power plant levels through thermo-fluid and energy flow modeling – Phase Two
- Detailed engineering, fabrication, purchasing and testing of a SDPCT-prototype under various simulated ambient conditions at BAC test facility – Phase Two
- Techno-economic assessment (TEA) and sensitivity studies to guide design optimizations – Phase Three

The project was structured by 7 major tasks scheduled over 36-month period as illustrated by the Table 1 below. The timeline indicates a start date, and end date for each task as well as shows interdependencies between tasks. The project is divided into three 12-month long Phases and is arranged in a Task structure to facilitate organized and efficient project execution.

Table 1. Project tasks

Research Tasks	**Timeline**		**Interdependent Tasks**
	Start Date	**End Date**	
Task 1.0: Project management, communication and reporting	10/01/2019	09/30/2022	All tasks
Task 2.0: Concept refinement and performance characterization	10/01/2019	03/31/2020	1.0
Task 3.0: Components design and manufacturing review	03/01/2020	12/31/2020	2.0
Task 4.0: SDPCT prototype engineering and fabrication	01/01/2021	06/30/2021	3.0
Task 5.0: SDPCT prototype installation and calibration	07/01/2021	12/31/2021	4.0
Task 6.0: Performance data collection and processing	01/01/2022	06/30/2022	2.0 and 4.0
Task 7.0: Techno-economic assessment (TEA) and scale-up	07/01/2022	09/30/2022	All above tasks
Final reporting	10/01/2022	12/31/2022	All tasks

Phase One – Concept Characterization and Testing Design

Task 2 – Concept refinement and performance characterization

GTI Energy refines the conventional cooling tower layout with reliable and efficient dehumidification technique based on near-atmospheric pressure regulation followed by the SDPCT performance characterization for developing cost-effective design solution and follow-on manufacturing review. IIT assists GTI Energy with numerical simulation of heat transfer processes and smart fluid flow arrangement in associated heat and mass exchangers to develop optimized HX and HMX systems, and the overall Pressure Dehumidifying System.

Task 3 – Components design and manufacturing review

GTI Energy designs the key components of the proposed SDPCT based on the computer modeling at IIT and experimental evaluation. BAC will assess the preliminary design and assist GTI Energy with manufacturing review, while Worley and industrial partners perform preliminary TEA. The project team reviews and finalizes the drawings, sketches, and calculations for the follow-on detailed engineering and fabrication. Upon manufacturing review, the drawings, sketches, and necessary quantitative justifications via modeling will be completed for the follow-on detailed engineering and fabrication of the SDPCT-prototype.

Go/No Go annual review – GTI Energy along with the team presents the refined SDPCT concept characterization results and preliminary TEA findings referencing a coal power plant.

Phase Two – Test Equipment Assembly and Test Plan Development

Task 4 – Engineering, purchasing and fabrication of the prototype

The design team led by BAC engineers the SDPCT integrated prototype design and estimates the SDPCT-prototype size for performance evaluation in the framework of the proposed effort. BAC builds the SDPCT test unit as per GTI Energy preliminary design and specifies the vendors for the system components (off-shelf and fabricated). Overall system level performance with the chosen components is modeled by IIT and evaluated for different operating points.

Task 5 – Assembly, installation and test plan development

Once parts are available, assembly of the SDPCT takes place at BAC research and development (R&D) facility in Jessup, MD. BAC with GTI Energy support assembles the SDPCT-prototype in one of BAC's Cooling Tower test platforms (TBD during Task 3).

Go/No Go annual review – BAC along with the team presents the SDPCT-prototype installed at the R&D facility along with the developed test plan/matrix

Phase Three – Testing, Analysis, and Techno-Economic Analysis

Task 6 – Data collection and processing

SDPCT-prototype system performance is experimentally tested and modeled for a wide range of process temperatures and flow rates. Data are collected and compiled for the project team review, analysis, and modeling verifications. Interdisciplinary technical support is provided by all team members throughout the execution of the test plan.

Task 7 – Performance analysis, scale-up design recommendations and TEA

Project team works collaboratively on the TEA with the primary objective to perform cost-benefit analysis of SDPCT technology by developing the economic model including project specific inputs, preliminary performance and cost estimates, and sensitivities. The economic model should identify focus areas for the remainder of the project to optimize the design to achieve minimum lifecycle costs. The TEA should consider supercritical coal power plants located at three typical US locations and three ambient conditions at each location. This range should assess if the SDPCT technology provides substantial benefits for specific locations.

Relevance and Outcomes

The relevance of the effort to the objectives of the program and expected outcome/impacts of the project when successful at optimizing, designing, fabricating and testing are as follows:

- Up to 20°F steam condensation temperature reduction leading to a net plant efficiency gain, meeting the AOI 2 primary objective
- Up to 20% cooling tower water use reduction, meeting the other AOI 2 water management objective

The potential outstanding impact of the cost effective, durable SDPCT technology is that it has a cross-cutting nature, is retrofittable and it can benefit not only coal-fired power generation systems but also other industries and applications. Cooling towers are broadly used for a wide spectrum of other cooling applications, such as buildings, data centers, and numerous other industries needing facility cooling, including food processing, petroleum, and gas industries. The overall world-wide impact of this technology with dramatically reduced cooled water temperature would not only significantly improve power production efficiency, or reduce the power plant CO2 emission, but also reduce the power consumption for cooling drastically. Furthermore, the attractive potential of over 20% of cooling tower water consumption reduction, would profoundly help conserve the limited and precious fresh water resource.

Concept refinement and performance characterization

Preliminary concept of the Sub-Dew Point Cooling Tower (SDPCT) is shown in Figure 6. The SDPCT employs an innovative flow arrangement called a pressure dehumidifying system (PDHS) coupled with effective heat and mass transfer so air is cooled and dehumidified prior to entering the cooling tower fill. The air cooling and dehumidification is accomplished by a near-atmospheric pressure regeneration technique and efficient heat exchange components. The main components of the PDHS are a blower, an air heat exchanger (HX), heat-mass exchanger (HMX) and expander. The blower in the system slightly pressurizes the incoming air and increases the air dew point, thus making it easier to remove moisture from the air using the HMX. The air HX cools pressurized air after the blower, and the air is further cooled and dehumidified in the HMX. The expander is used to offset the power consumed by the blower, thus making this an ultra-low energy system.

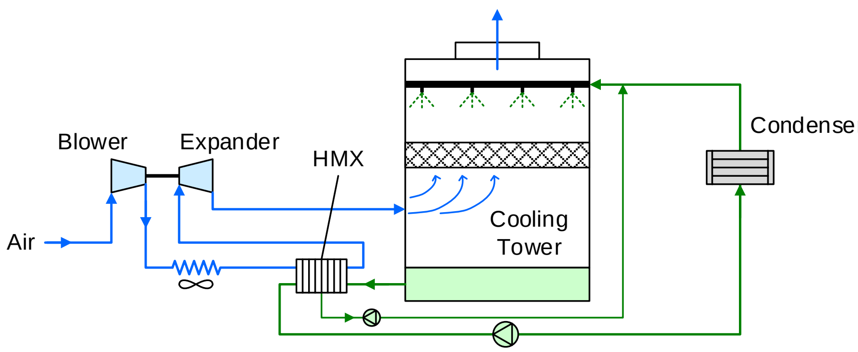


Figure 6 Preliminary concept of sub-dew point cooling tower with pressure dehumidifying system

The water cooled HMX scheme, HMX design parameters and simulation results are presented in Figure 7 and Table 2.

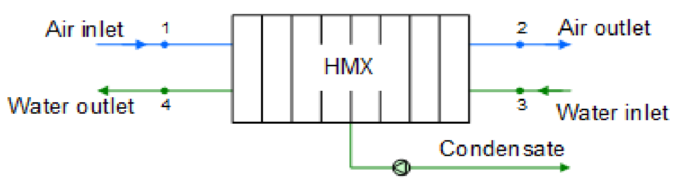


Figure 7 Water cooled Heat Mass Exchanger (HMX)

Preliminary PDHS estimates were made for a 100 kW cooling tower under the following cooling tower design conditions:

- ambient air temperature: 100°F (310.93°K)
- ambient relative humidity: 50%
- ambient wet bulb temperature: 78°F (298.71°K)
- ambient dew point temperature: 70°F (294.26°K)
- ambient pressure: 101.3 kPa
- entering water inlet temperature: 95°F (308.15°K)
- cooled water temperature: 65.7°F (291.7°K)

Operational ranges for water cooled air dehumidifier:

- air inlet temperature (point #1): 294°K … 321°K
- air inlet relative humidity (point #1): 30% … 85%
- air inlet mass flow rate (point #1): to be specified
- water inlet temperature (point #3): 276…300°K
- water mass flow rate: 1.6 kg/s (constant)
- air inlet pressure: 121,310 Pa (constant)
- water inlet pressure: 14.7 psi (constant)

Table 2. HMX (water cooled air dehumidifier) parameters at nominal design conditions

Stream # (see Figure 7)	1 (air)	2 (air)	3 (water)	4 (water)
Temperature	313.9°K (dry bulb) 296.9°K (dew point)	291.0°K	284°K°K	291.7°K - predicted
Air relative humidity	38.24%	100%	-	-
Absolute pressure	121,310 Pa	To be estimated	14.7 psi	To be estimated
Mass flow rate	2.433 kg/s	2.433 kg/s	1.6 kg/s	1.6 kg/s

Preliminary estimates have shown the possibility of more efficient dehumidification of air at a slightly higher pressure compared to ambient air. While the ambient dew point is 294.26°K at 1 atm, the dew point after the blower rises to 296.9°K at elevated pressure of 1.2 atm, allowing more water vapor to condense from the ambient air.

During the implementattion of the project, the peliminary concept was refined and the parameters of the pilot scale system were specified (Figure 8). Based on the refined concept, preliminary estimates of the performance of the pilot scale SDPCT were made. While the blower, HX and expander may be commercially available for pilot test setup, the HMX must be designed and built or specially ordered. The design and parameters of the tubular HMX were evaluated for the experimental setup. The maximum operating air pressure in HMX was set at 1.4 atm. The HMX configuration and preliminary design parameters for a 53 kW cooling tower are shown in Figure 9 and Table 4.

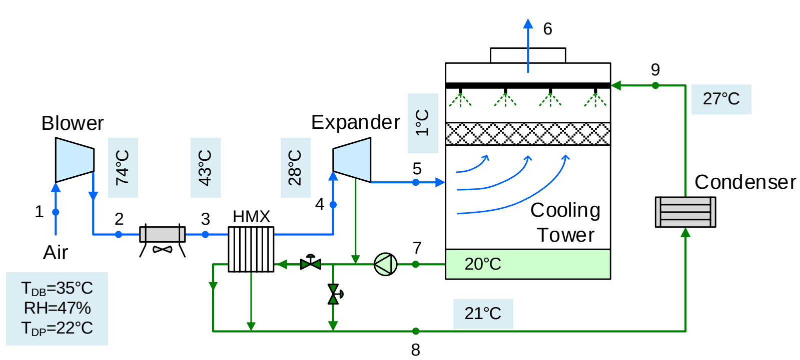
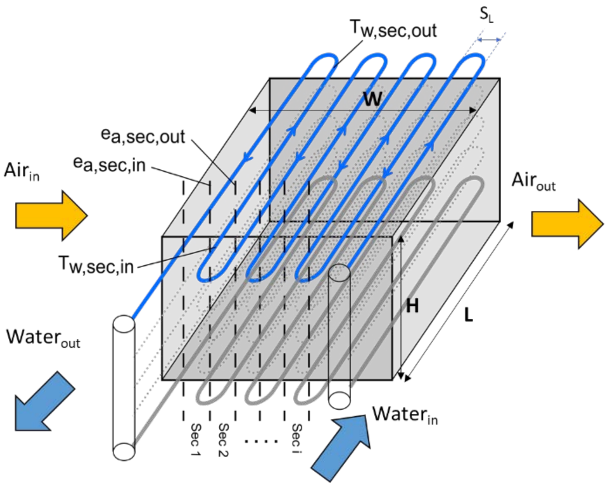


Figure 8 Refined concept of the SDPCT and parameters of the pilot scale system at nominal design conditions: HMX – heat and mass exchanger, T_{DB} – dry bulb temperature, T_{DP} – dew point temperature, RH – relative humidity



HMX Nominal Conditions		
Parameters	Inlet	Outlet
Air Temperature	42.8 °C	27.9 °C
Air Humid. Ratio	0.0167 kg/kg	0.0162 kg/kg
Air RH	44 %	100 %
Air Pressure	144640 Pa	T.B.E
Air Flow Rate	1.08 kg/s	1.08 kg/s
Water Temperature	19.7 °C	27.8 °C
Water Flow rate	0.50 kg/s	0.50 kg/s
Water Pressure	14.7 Psi	T.B.E

Figure 9 Heat-Mass Exchanger (HMX) design configuration

Table 3. Preliminary design parameters of HMX at nominal conditions

HMX Construction Parameters	
Physical quantity	**Values**
Tube arrangement	Staggered
Tube Material	Copper
Tube Outside Diameter* (d_o)	9.52 mm
Tube Inner Diameter* (d_i)	9.37 mm
Tube Thickness* (δ_t)	0.0762 mm
Longitudinal Tube Spacing (S_L)	21.10 mm
Transverse Tube Spacing (S_T)	20.06 mm
Total Outer Area (A_o)	62.2 m^2
Total Inner Area (A_i)	10.1 m^2
Fin Type	Al Plain Continuous
Fin Thickness (δ_f)	0.1 mm
Fin Density	428 /m
No. of Tubes in Height (N_c)	26
Number of Sections (N_{sec})	25
HMX Height (H)	0.52 m
HMX Width (W)	0.53 m
HMX Length (L)	0.53 m
Tube/Fin Material Volume	5.9×10^{-3} m^3

HMX Operating Data	
Physical quantity	**Values**
Operating Pressure (P)	**144640 Pa**
Design Cooling Load (Q_{design})	16.8 kW
Actual Cooling Load (Q_{actual})	17.4 kW
Air Inlet Dry Bulb Temperature ($T_{a,in}$)	42.85 °C
Air Inlet RH	44.0 %
Air Exit Temperature ($T_{a,out}$)	28.4 °C
Air Exit RH	95 %
Air Flowrate (m_a)	1.08 Kg/s
Air Heat Transfer Coefficient (h_o)	78 W/m^2-K
Air Mean Specific Heat (Cp_m)	1036 J/kg-K
Water Inlet Temperature ($T_{w,in}$)	19.42 °C
Water Exit Temperature ($T_{w,out}$)	27.8 °C
Water Flowrate (m_w)	**0.50 kg/s**
Water Heat transfer coefficient (h_i)	226.1 W/m^2-K
Water specific heat (Cp_w)	4140 J/kg-K
Water Condensate (m_{cond})	0.47 g/s
Fan Power Consumption	4.14 kW
COP	4.20

Taking into account the estimated parameters of the HMX, the performance of pilot scale SDPCT in nominal and off-design conditions was preliminary estimated. The results of the sensitivity analysis of the pilot scale SDPCT are presented in Table 4. As can be seen from the table, water cooling below the ambient dew point in the cooling tower can be achieved by using the PDHS at an ambient temperature of 20°C or higher and high relative humidity (≥40%).

Table 4. Preliminary estimates of the pilot scale SDPCT performance under off-design conditions: sub-dew point water cooling cases are highlighted in green, temperatures correspond to Figure 8, T - w/o PDHS,* $\Delta T_1=T_8\text{-}T_{8,\ w/PDHS}$, $\Delta T_2=T_8\text{-}T_{amb.\ wet\ bulb}$, $\Delta T_3=T_8\text{-}T_{amb.\ dew\ point}$

T_1, °C	RH_1, %	Tdp_1, °C	Twb_1, °C	T_8, °C	T_5, °C	T_8^*, °C	ΔT_1, °C	ΔT_2, °C	ΔT_3, °C
12.5	20.0	-10.1	4.3	19.0	0.1	18.9	0.1	14.7	29.1
12.5	30.0	-4.7	5.4	19.4	0.1	19.4	0.0	14.0	24.1
12.5	40.0	-0.8	6.5	20.8	0.7	19.9	0.9	14.3	21.6
12.5	50.0	2.4	7.6	21.0	0.0	20.5	0.5	13.4	18.6
12.5	60.0	5.0	8.7	20.8	0.1	21.0	-0.2	12.1	15.8
12.5	70.0	7.2	9.7	20.6	0.2	21.5	-0.9	10.9	13.4
12.5	80.0	9.2	10.6	21.3	0.0	22.0	-0.7	10.7	12.1
12.5	90.0	10.9	11.6	20.3	0.1	22.5	-2.1	8.7	9.4
12.5	100.0	12.5	12.5	20.2	0.2	22.9	-2.7	7.7	7.7
15.0	20.0	-8.0	6.0	19.1	0.1	19.7	-0.5	13.1	27.1
15.0	30.0	-2.5	7.2	19.7	0.2	20.3	-0.6	12.5	22.2
15.0	40.0	1.5	8.5	20.8	0.0	20.9	-0.1	12.3	19.3
15.0	50.0	4.7	9.7	21.0	0.0	21.5	-0.4	11.3	16.3
15.0	60.0	7.3	10.8	19.8	0.1	22.1	-2.2	9.0	12.5
15.0	70.0	9.6	11.9	20.4	0.2	22.6	-2.3	8.5	10.8
15.0	80.0	11.6	13.0	19.8	0.1	23.2	-3.3	6.8	8.2
15.0	90.0	13.4	14.0	21.3	0.0	23.7	-2.5	7.3	7.9
15.0	100.0	15.0	15.0	20.1	0.2	24.3	-4.2	5.1	5.1
20.0	20.0	-3.7	9.3	19.5	0.2	21.3	-1.7	10.2	23.2
20.0	30.0	1.9	10.9	20.9	0.0	22.1	-1.2	10.0	19.0
20.0	40.0	6.0	12.4	20.2	0.1	22.8	-2.6	7.8	14.2
20.0	50.0	9.3	13.8	20.7	0.2	23.6	-2.9	6.9	11.4
20.0	60.0	12.0	15.1	19.9	0.2	24.3	-4.4	4.8	7.9
20.0	70.0	14.4	16.4	19.9	0.2	25.0	-5.1	3.5	5.5
20.0	80.0	16.5	17.7	19.8	0.0	25.7	-5.9	2.1	3.3
20.0	90.0	18.3	18.9	20.0	0.1	26.4	-6.4	1.1	1.7
20.0	100.0	20.0	20.0	19.8	0.1	27.0	-7.2	-0.2	-0.2
25.0	20.0	0.5	12.5	19.9	0.1	22.9	-3.0	7.4	19.4
25.0	30.0	6.3	14.4	20.3	0.0	23.9	-3.6	5.9	14.0
25.0	40.0	10.5	16.2	20.5	0.0	24.9	-4.4	4.3	10.0
25.0	50.0	13.9	17.9	19.9	0.2	25.8	-5.9	2.0	6.0
25.0	60.0	16.7	19.5	19.9	0.2	26.7	-6.8	0.4	3.2
25.0	70.0	19.2	21.0	19.8	0.0	27.6	-7.8	-1.2	0.6
25.0	80.0	21.3	22.4	19.7	0.1	28.4	-8.7	-2.7	-1.6
25.0	90.0	23.2	23.7	19.6	0.4	29.2	-9.7	-4.1	-3.6
25.0	100.0	25.0	25.0	20.1	1.7	30.0	-9.9	-4.9	-4.9
30.0	20.0	4.6	15.7	20.1	0.4	24.6	-4.5	4.4	15.5
30.0	30.0	10.6	18.0	20.0	0.1	25.8	-5.8	2.0	9.4
30.0	40.0	15.0	20.1	20.1	0.5	27.0	-6.9	0.0	5.1
30.0	50.0	18.5	22.0	20.1	0.6	28.2	-8.1	-1.9	1.6
30.0	60.0	21.4	23.8	20.2	0.9	29.3	-9.0	-3.6	-1.2
30.0	70.0	23.9	25.5	20.7	2.1	30.3	-9.6	-4.8	-3.2
30.0	80.0	26.2	27.1	21.3	3.4	31.3	-10.0	-5.8	-4.9
30.0	90.0	28.2	28.6	21.9	4.9	32.3	-10.4	-6.7	-6.3
30.0	100.0	30.0	30.0	22.6	6.3	33.2	-10.6	-7.4	-7.4
35.0	20.0	8.7	18.9	21.7	3.3	26.3	-4.6	2.8	13.0
35.0	30.0	14.9	21.5	21.5	0.2	27.8	-6.4	0.0	6.6
35.0	40.0	19.4	23.9	21.8	3.5	29.3	-7.5	-2.1	2.4
35.0	50.0	23.0	26.1	21.8	3.0	30.7	-8.9	-4.3	-1.2
35.0	60.0	26.1	28.2	22.3	4.7	32.0	-9.7	-5.9	-3.8
35.0	70.0	28.7	30.1	22.9	6.3	33.2	-10.3	-7.2	-5.8
35.0	80.0	31.0	31.8	23.7	8.0	34.4	-10.8	-8.1	-7.3
35.0	90.0	33.1	33.5	24.5	9.6	35.6	-11.1	-9.0	-8.6
35.0	100.0	35.0	35.0	25.3	11.2	36.7	-11.4	-9.7	-9.7
40.0	20.0	12.8	22.0	23.0	2.6	28.1	-5.1	1.0	10.2
40.0	30.0	19.1	25.1	23.4	5.9	30.0	-6.6	-1.7	4.3
40.0	40.0	23.8	27.8	23.2	3.8	31.7	-8.5	-4.6	-0.6
40.0	50.0	27.6	30.3	23.7	7.1	33.4	-9.6	-6.6	-3.9

The optimization results of the pilot scale SDPCT and PDHS are presented in Table 5.

Table 5. Optimization results of pilot scale PDHS

Case #	1	2	3	4	5	6	7	8	9	10
1. **Ambient Conditions**										
Air Dry Bulb Temp., °C (T1)	35	35	35	35	35	35	35	35	35	35
Air Relative Humidity, %	47	47	47	47	47	47	47	47	47	47
Air Dew Point Temp., °C	22.01	22.01	22.01	22.01	22.01	22.01	22.01	22.01	22.01	22.01
Air Wet Bulb Temp., °C	25.5	25.5	25.5	25.5	25.5	25.5	25.5	25.5	25.5	25.5
Air Flow rate , kg/s	3.05	3.05	3.05	3.05	3.05	3.05	3.05	3.05	3.04	2.97
2. **Compressor Outlet Conditions**										
Air Dry Bulb Temp., °C (T2)	65.27	63.03	63.38	63.98	60.76	62.51	63.98	60.41	63.46	57.21
Air Relative Humidity, %	13.8	14.96	14.77	14.45	16.26	15.25	14.45	16.47	14.73	18.58
Pressure ratio	1.32	1.30	1.3	1.31	1.27	1.29	1.31	1.27	1.3	1.23
Power Consumed, kW	92.49	85.57	86.64	88.51	78.57	83.96	88.51	77.49	86.64	66.23
3. **Air Cooler Outlet Conditions**										
Air Dry Bulb Temp., °C (T3)	23.01	21.41	22.53	22.11	22.71	22.63	26.24	22.98	22.31	23.1
Air Relative Humidity, %	100	100	100	100	100	100	100	100	100	100
Efficiency/Effectiveness	0.63	0.67	0.69	0.69	0.71	0.7	0.62	0.72	0.68	0.68
Condensate, (g/s)	10.06	13.13	10.54	11.79	9.12	9.98	0.58	8.28	11.09	6.53
4. **HMX Outlet Conditions**										
Air Dry Bulb Temp., °C (T4)	N.A	N.A	21.62	N.A	19.75	21	22.05	19.5	N.A	N.A
Air Relative Humidity, %	N.A	N.A	100	N.A	100	100	100	100	N.A	N.A
Water Inlet Temp., °C (T7)	N.A	N.A	12.78	N.A	15	15	17.22	17.22	N.A	N.A
Water Outlet Temp., °C (T9)	N.A	N.A	20.13	N.A	21.53	18.3	25.22	20.63	N.A	N.A
Water Flow rate, kg/s (Mw7/Mw12: 0.1-1)	N.A	N.A	0.27	N.A	0.99	1.09	1.26	2.24	N.A	N.A
Condensate, g/s	N.A	N.A	2.13	N.A	6.93	3.84	11.27	8.2	N.A	N.A
Cooling load, kW	N.A	N.A	8.27	N.A	26.82	14.85	41.85	31.69	N.A	N.A
5. **Expander Outlet Conditions**										
Air Dry Bulb Temp., °C (T5)	1.67	1.67	1.67	1.67	1.67	1.67	1.67	1.67	2.22	7.22
Air Wet Bulb Temp., °C	1.67	1.67	1.67	1.67	1.67	1.67	1.67	1.67	2.22	7.22
Air Relative Humidity, %	100	100	100	100	100	100	100	100	100	100
Power Output, kW	65.11	60.15	60.92	62.26	55.14	59	62.26	54.36	61.05	47.25
Condensate, (g/s)	27.65	24.54	25.01	25.84	21.59	23.84	25.84	21.15	25.95	24.21
6. **Cooling Tower Conditions**										
Water Inlet Temp., °C (T12)	12.78	15.00	28.33	17.22	30.56	19.44	32.78	21.67	18.33	20.56
Water Outlet Temp., °C (T7 or T10)	10.56	10.56	12.78	12.78	15	15	17.22	17.22	12.78	16.11
Circ. water mass flow rate, kg/s (Inlet)	4.52	2.25	0.97	2.97	1.21	3.75	1.47	4.58	2.27	2.80
Air mass flow rate, kg/s	3.01	3.01	3.01	3.01	3.01	3.01	3.01	3.01	3.00	2.94
Evaporation (g/s)	8.13	8.07	12.69	10.96	16.27	14.23	20.46	17.88	10.65	11.77
Cooling load, kW	41.99	41.88	63.25	55.20	78.76	69.74	96.39	85.36	52.72	52.08
7. **Chiller Conditions**										
Water Inlet Temp., °C (T10)	10.56	10.56	12.78	12.78	15	15	17.22	17.22	12.78	16.11
Water Outlet Temp., °C (T11)	12.78	15.00	31.52	17.22	71.7	19.91	79.25	22.67	18.33	20.56
Cooling load, kW	41.54	41.36	54.18	54.60	51.12	54.08	52.85	52.69	52.20	51.58
Water mass flow rate (kg/s)	4.52	2.25	0.7	2.97	0.22	2.66	0.21	2.34	1.63	2.80
Sub dew point cooling, °C	-11.45	-11.45	-9.23	-9.23	-7.01	-7.01	-4.79	-4.79	-9.23	-5.9
COP*	1.52	1.63	2.11	2.08	2.18	2.17	2.01	2.28	2.04	2.72

Based on preliminary analysis of the SDPCT performance benefits and techno-economic assessment (TEA), the following conclusions were made:

- Water cooling temperature: goal – 1°F below dew point of ambient air; estimated potential – 1°F-20°F below dew point of ambient air
- Water savings: goal – 10-20%, estimated potential – >20%
- Net plant efficiency gain: goal – up to 3%, estimated potential – up to 1%

The refined SDPCT concept has been adapted for a 650 MW coal power plant with a retrofitied cooling tower using PDHS and a larger cooling tower fill surface area. The results of the calculations are shown in Figure 10. Parameters shown in the figure were calculated and optimized based on the minimum capital costs of the PDHS, maximum net power gain of the power plant, and the lowest cost of the electricity production. The economic justification for using a blower to slightly pressurize the incoming ambient air for cooling and dehumidification was done. The economic justification was done using the following elements: nominal power 650MW, the price of the energy produced at power plant 40 $/MWh, makeup water total cost 9.22 $/1,000 gal including cost of sewerage.

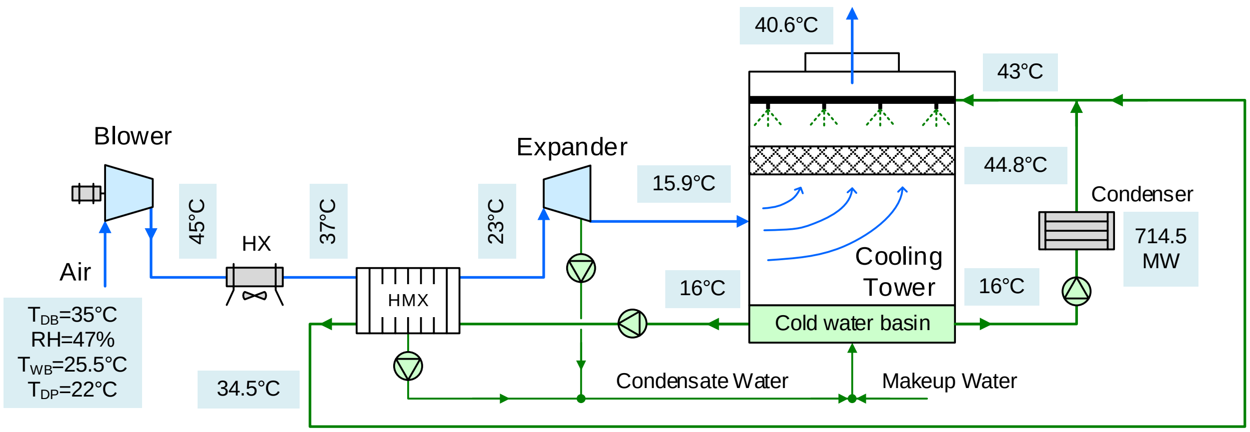


Figure 10 Further refined concept of the SDPCT adapted for a 650 MW coal power plant with PDHS display under nominal design conditions: HX – air cooled heat exchanger, HMX – heat and mass exchanger, T_{DB} – dry bulb temperature, T_{WB} – wet bulb temperature, T_{DP} – dew point temperature, RH – relative humidity

Preliminary analysis showed the following advantages of the SDPCT (estimated for baseline conditions):

- Achieving sub-dew point cooling of water: 6°C below ambient dew point temperature of 22°C or 9.5°C below ambient wet bulb temperature of 25.5°C, and 12°C below temperature of the cooling water compared to the standard cooling tower without PDHS
- Reduced makeup water consumption by 24% due to water harvesting from air and increased performance of the cooling tower
- Net power gain of the power plant by 1.4% due to lower temperature of the cooling water
- Reduction in the cost of electricity production by 4.3% due to increased performance of the power plant and reduced consumption of makeup water

SDPCT test prototype

The SDPCT testing and demonstration have been discussed with Baltimore Aircoil Company (BAC). The research and development facility at BAC is well known in the industry for its ability to support the demonstration testing and to serve as an independent testing lab for new developments.

BAC's VT0-12-E V-Series full size cooling tower (Figure 11) was considered for testing purposes to demonstrate the new SDPCT concept. The VT0-12-E is a forced draft, counterflow, centrifugal fan cooling tower with the following design conditions: 52.8 kW cooling capacity, 36 GPM water flowrate, 95°F ambient temperature, 78°F ambient wet bulb temperature.

Other major SDPCT components such as air fan or blower, turbo-expander, air-to-air heat exchanger, water-to-air heat exchanger and water pumps are off-the-shelf or bespoke straightforward modifications of existing designs. The air fan/blower and the turbo-expander are low-pressure units with a pressure ratio of up to 1.2. The nominal air flowrate through the air fan/blower and turbocharger is 5000 CFM.

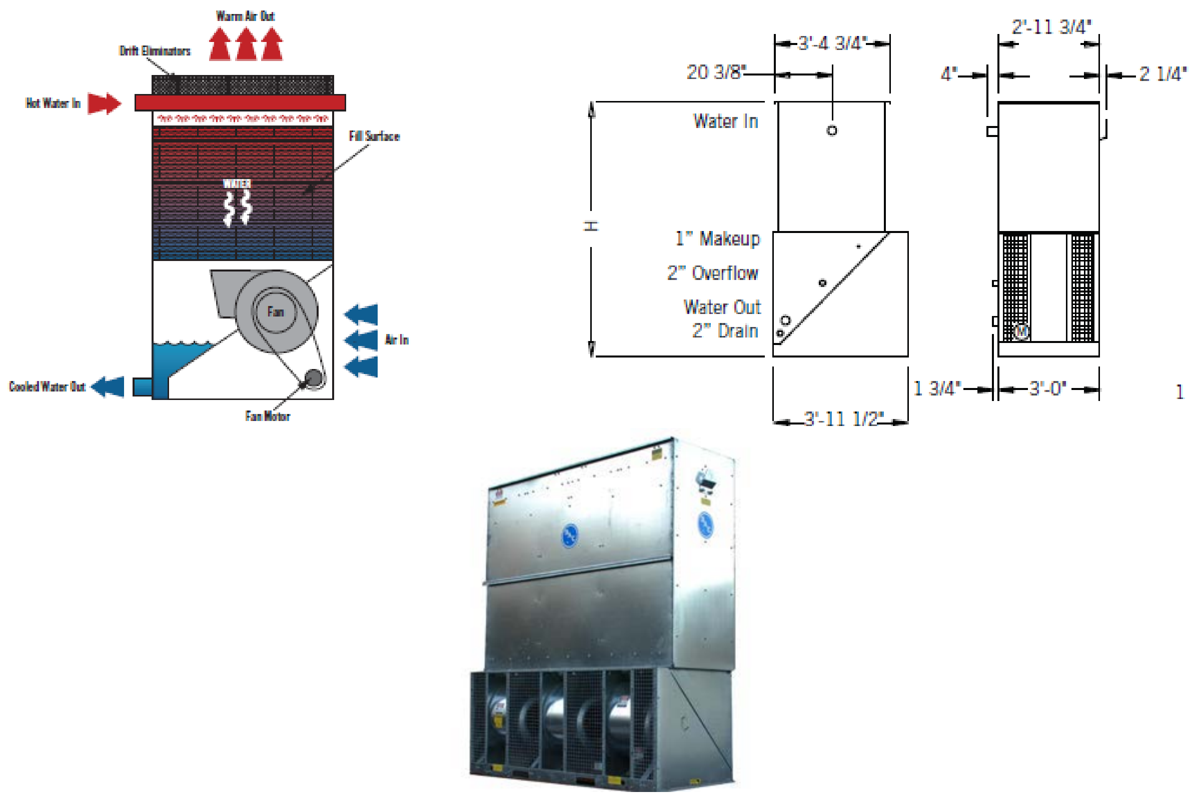


Figure 11 Series V cooling tower VTO-12-E by BAC: schematics, dimensions and photo

Figure 12 shows simplified process diagram of the SDPCT test unit with main components

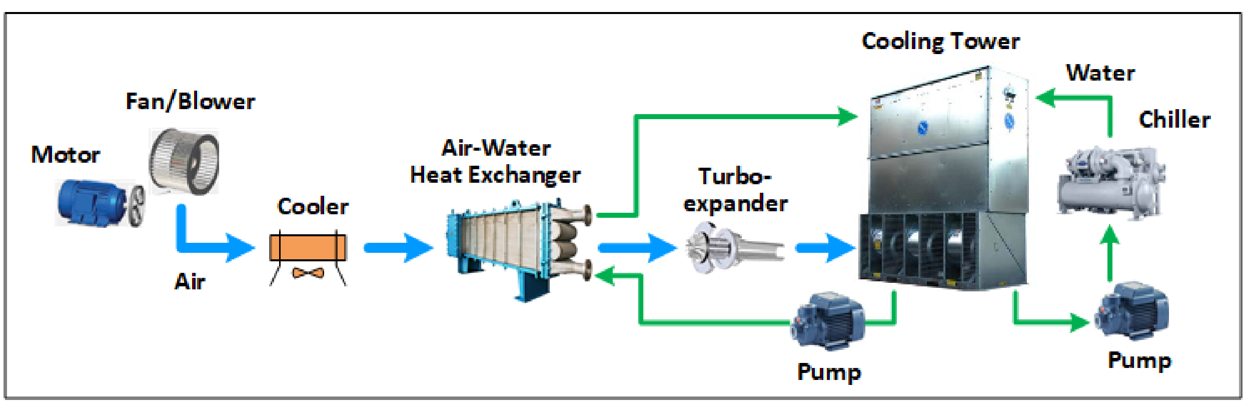


Figure 12 The SDPCT process flow diagram with main components of the test unit

Figure 13 shows key components of the PDHS and the nominal (design) conditions for testing a prototype PDHS for the VTO-12-E cooling tower (cooling capacity 52.8 kW). Table 6 lists the nominal values of the PDHS parameters.

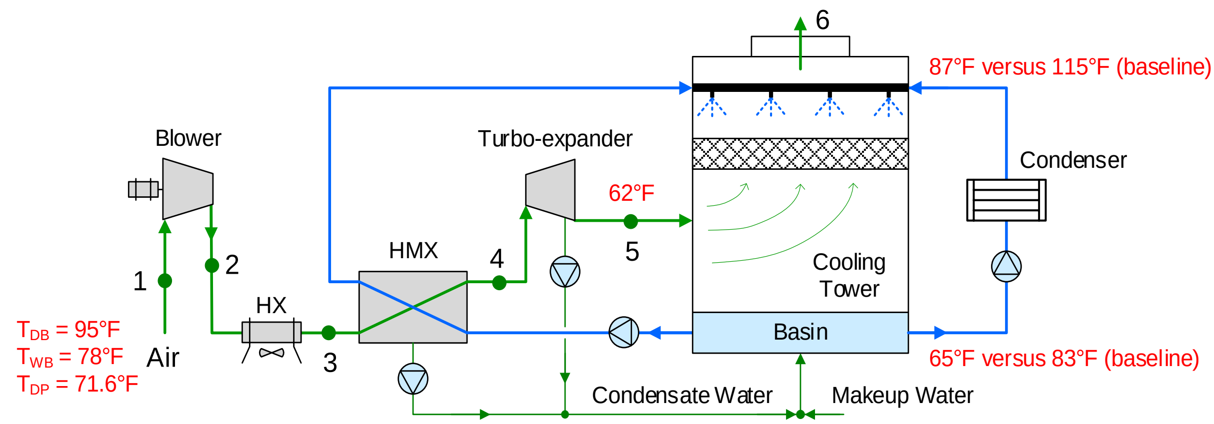


Figure 13 Key components of the PDHS for prototype testing and design conditions

Table 6. Nominal (design) test conditions for PDHS (locations correspond to Figure 13)

Component	Blower E2		HX-E3		HX-E4 (air stream)	HX-E4 (water stream)		Expander E5
Location	In	P1	P2	Ambient Air	P3	P8	P5	P16
Fluid	Air	Air	Air	Air	Air	Water	Water	Air
Temperature, °F	95	132	105	95	73.5	70.5	94.5	62
Pressure, psig	0	2.94	2.87	0	2.8	1.5	1.5	0
Flow, scfm/gpm	5,000 SCFM			5,000 SCFM	5,000 SCFM	14 GPM		5,000 SCFM

Piping and instrumentation diagram (P&ID) of SDPCT prototype

P&ID of the SDPCT prototype (52.8 kW) has been developed and finalized as shown in Figure 14. The prototype was planned to be tested at the experimental facility of Baltimore Aircoil Company in Jessup, MD. Nominal SDPCT component boundary conditions in terms of pressures, temperatures, volume and mass flowrates, that support the SDPCT conceptual design have been defined and are presented in Table 7 through Table 9.

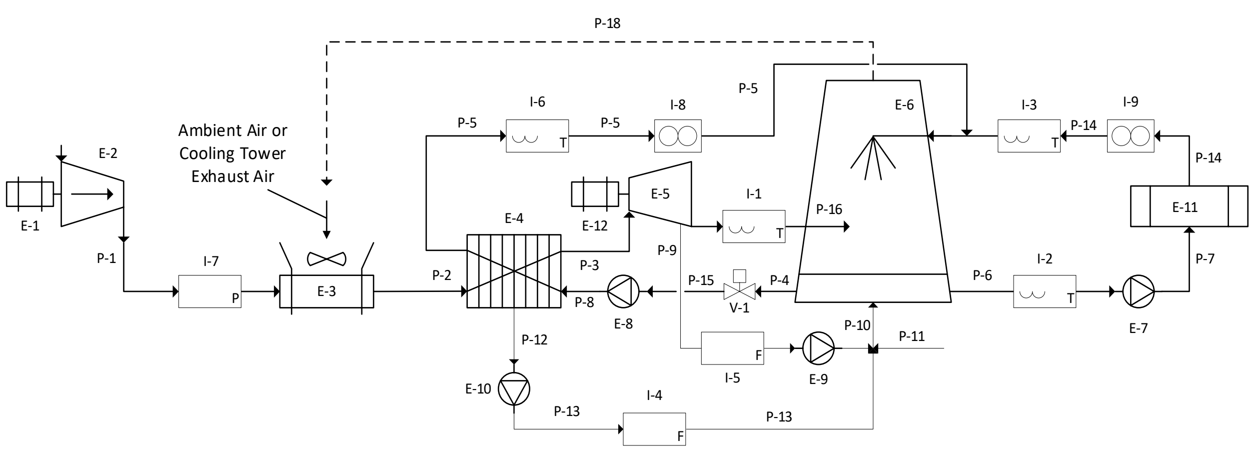


Figure 14 Piping and instrumentation diagram (P&ID) of the SDPCT prototype

Table 7. Process annotations for equipment list of the SDPCT P&ID (presented in Figure 14*)*

Equipment List				
Displayed Text	**Description**	**Manufacturer**	**Model**	**Note**
E-1	Elecitric motor	TBD	TBD	30 kW, standard, off-the-shelf
E-10	Water pump	TBD	TBD	0.1 gpm, standard, off-the-shelf
E-11	Water heater	TBD	TBD	53 kW, existing, BAC test unit
E-2	Blower	TBD	TBD	5000 CFM, standard, off-the-shelf
E-3	Air cooler	TBD	TBD	10 kW, availbale, custom made
E-4	Heat exchanger	TBD	TBD	20 kW, availbale, custom made
E-5	Turbo-expander	TBD	TBD	5000 CFM, standard, off-the-shelf
E-6	Cooling tower	Baltimore Aircoil (BAC)	VT0-12-E	52.8 kW, existing test unit at BAC
E-7	Water pump	TBD	TBD	36 gpm, existing, BAC test unit
E-8	Water pump	TBD	TBD	10 gpm, standard, off-the-shelf
E-9	Water pump	TBD	TBD	0.5 gpm, standard, off-the-shelf

Table 8. Process annotations for instrument list of the SDPCT P&ID (presented in Figure 14*)*

Instrument List				
Displayed Text	**Description**	**Manufacturer**	**Model**	**Note**
I-1	Thermometer	TBD	TBD	0-35°C, water, off-the-shelf
I-2	Thermometer	TBD	TBD	0-30°C, water, off-the-shelf
I-3	Thermometer	TBD	TBD	0-50°C, water, off-the-shelf
I-4	Water flowmeter	TBD	TBD	0.5 gpm, off-the-shelf
I-5	Water flowmeter	TBD	TBD	0.5 gpm, off-the-shelf
I-6	Thermometer	TBD	TBD	0-30°C, water, off-the-shelf
I-7	Pressure gauge	TBD	TBD	0-0.2 bar, off-the-shelf
I-8	Water flowmeter	TBD	TBD	0-10 gpm, off-the-shelf
I-9	Water flowmeter	TBD	TBD	0-36 gpm, existing, BAC test unit

Table 9. Process annotations for pipeline list of the SDPCT P&ID (presented in Figure 14*)*

Pipeline List						
Displayed Text	**Description**	**Fluid**	**Flowrate**	**Pressure**	**Temperature**	**Quantity**
P-1	Blower outlet line	Air	5000 cfm	1-1.2 bara	55°F-150°F	2
P-10	Water line	Water	0.3-10 gpm	1.1 bara	55°F-75°F	2
P-11	Make-up water line	Water	0.3-10 gpm	2 bara	45°F-75°F	1
P-12	Water condensate line	Water	0-0.1 gpm	1.1 bara	55°F-90°F	1
P-13	Water condensate line	Water	0-0.1 gpm	1.1 bara	55°F-90°F	2
P-14	Hot water line	Water	36 gpm	2 bara	60°F-110°F	3
P-15	Cold water line	Water	0-10 gpm	1.1 bara	55°F-75°F	1
P-16	Cooling tower air entering line	Air	5000 cfm	1 bara	50°F-70°F	1
P-2	Air cooler outlet line	Air	5000 cfm	1-1.2 bara	55°F-100°F	1
P-3	Turboexpander inlet line	Air	5000 cfm	1-1.2 bara	55°F-95°F	1
P-4	Cold water line	Water	0-10 gpm	1.1 bara	55°F-75°F	1
P-5	Hot water line	Water	0-10 gpm	2 bara	60°F-95°F	3
P-6	Cold water line	Water	36 gpm	1 bara	55°F-75°F	1
P-7	Water heater inlet pipe	Water	36 gpm	1 bara	55°F-75°F	1
P-8	Heat exchanger water inlet line	Water	0-10 gpm	1.1 bara	55°F-75°F	1
P-9	Water condensate line	Water	0-0.5 gpm	1 bara	50°F-90°F	1

Based on the refined SDPCT concept and the nominal parameters of the prototype test rig, the detailed parameters of the prototype SDPCT and PDHS were evaluated (Table 10). These parameters were used to define and order PDHS components for testing.

Table 10. Detailed parameters of the SDPCT prototype at nominal (design) conditions

Air Path				5.878208844	kW					3.840309704	kW			0	kW		
				heat added		heat removed		heat removed		heat removed		heat added		heat added		heat added	
Heat Transfer	Btu/min	-->		334.3721131		244.5859921		359.6970503		218.449617		97.55238887		0		244.5859921	
Material	Unit	-->	Amb Air	E2: Blower	P-1	E3: Air Cooler HX_3	P-2	E4: Air Cooler HMX_4	P-3	E5: Turbo Expander	P-16	E6: Cooling Tower	E-6	E6: Cooling Tower Fan	P-18	E3: Air Cooler HX_3	Air Out
Temperature	°F	-->	95		132		105		73.49999992		61.99999907		75.18337424		75.18910987		102.1211299
Specific Humidity	lbmw/lbmda	-->	0.016857143		0.016857143		0.016857143		0.0149129		0.011918655		0.018708169		0.018708169		0.018708169
Moisture Content	lbmw/lbm	-->	0.01657769		0.01657769		0.01657769		0.014693774		0.011778274		0.018364601		0.018364601		0.018364601
Enthalpy Prior Comp	Btu/lbm	-->	#N/A		157.8377578		151.2402463		143.5487376		138.9491407		139.3427724		145.7085772		152.2941007
Enthalpy Current Comp	Btu/lbm	-->	148.8183375		157.8377578		151.2402463		141.7288594		136.1392464		145.7085772		145.7085772		152.2941007
Enthalpy Next Comp	Btu/lbm	-->	148.8183375		157.8377578		149.4079203		138.9123955		142.4869		145.7085772		145.7085772		#N/A
Enthalpy H2O vap	Btu/lbm	-->	1103.295219		1119.00655		1107.577907		1094.016681		1089.021297		1094.74619		1094.748674		1106.347404
Enthalpy H2O liq	Btu/lbm	-->	63.08339885		100.0585972		#N/A		41.60011733		30.09816712		43.28286139		43.28859456		70.19739259
Density	lbm/ft3	-->	0.070803075		0.079640046		0.08324565		0.088053259		0.075511523		0.073355984		0.074256036		0.069828803
Flow Rate	ACFM	-->	523.5995998		465.500258		445.3381263		420.2181639		488.566693		506.2974332		500.1606377		531.8714439
Flow Rate	SCFM	-->	500		500.065621		499.9995069		498.3890199		496.0590426		501.3908005		501.3883172		501.4610105
Flow Rate	lbm/min	-->	37.07246185		37.07246185		37.07246185		37.00157888		36.89241502		37.13994636		37.13994636		37.13994636
Flow Rate Dry	lbm/min	-->	36.45788606		36.45788606		36.45788606		36.45788606		36.45788606		36.45788606		36.45788606		36.45788606
Flow Rate H2O	lbm/min	-->	0.614575794		0.614575794		0.614575794		0.543692821		0.434528966		0.682060301		0.682060301		0.682060301
Pressure	psia	-->	14.69595		17.63514		17.590035		17.54493		14.69595		14.69595		14.87635		14.69595
Water Out/In	lbm/min	-->		1.2				0.070882972	lbm/min	0.109163855	lbm/min	0.247531335	lbm/min				
				press ratio								62.62096806	Btu/lbm	liq in			
												1094.74619	Btu/lbm	vap out			
Ref Temperature	°F	70						water out		water out		water in					
Ref Pressure	psia	14.69595						73.49999992	°F	61.99999907	°F	94.5	°F				
Water STP Density	lbm/gal	8.328474309										101.3729006	°F				
												69.48607364	Btu/lbm				
Wet Bulb Temp	°F	78										99.25171762	Btu/min				
Dew Point	°F	71.8															
Relative Humidity	%	47.5															
	MW	dry	mass fraq		mass fraq		mass fraq		mass fraq		mass fraq		mass fraq		mass fraq		mass fraq
Nitrogen	28.01348	0.755704008	0.743176181		0.743176181		0.743176181		0.744599864		0.746803119		0.741825805		0.741825805		0.741825805
Oxygen	31.9988	0.231604716	0.227765245		0.227765245		0.227765245		0.228201569		0.228876812		0.227351388		0.227351388		0.227351388
Argon	39.948	0.012691276	0.012480884		0.012480884		0.012480884		0.012504793		0.012541795		0.012458206		0.012458206		0.012458206
Water	18.01527	0	0.01657769		0.01657769		0.01657769		0.014693774		0.011778274		0.018364601		0.018364601		0.018364601
		Air	Nitrogen;Oxy		Nitrogen;Oxy		Nitrogen;Oxy		Nitrogen;Oxy		Nitrogen;Oxy		Nitrogen;Oxy		Nitrogen;Oxy		Nitrogen;Oxy
	1/MW	dry	mole fraq		mole fraq		mole fraq		mole fraq		mole fraq		mole fraq				
Nitrogen	0.0356971	0.7812	0.760590306		0.760590306		0.760590306		0.762911706		0.766514643		0.758393294				
Oxygen	0.03125117	0.2096	0.204070312		0.204070312		0.204070312		0.204693156		0.205659843		0.203480843				
Argon	0.02503254	0.0092	0.008957285		0.008957285		0.008957285		0.008984623		0.009027054		0.008931411				
Water	0.05550847	0	0.026382097		0.026382097		0.026382097		0.023410514		0.01879846		0.029194452				
							part press		part press		part press		part press				
Nitrogen	psia						13.3788101		13.38523248		11.26466087		11.14530992				
Oxygen	psia						3.589603938		3.591327097		3.022366767		2.990344291				
Argon	psia						0.157558951		0.157634586		0.132661137		0.13125557				
Water	psia						0.464062012		0.410735836		0.276261224		0.429040214				

Water Loop										
				heat added				heat removed		
Heat Transfer	Btu/min	-->		279.6402674				17.49560614		
Material	Unit	-->	P-8	E4: Air Cooler	P-5	P-5+P-10	P-10	E6: Cooling Tower	P-4	
Temperature	°F	-->	70.5		94.5	94.07439797	66.5260576		70.5	
Enthalpy H2O vap	Btu/lbm	-->	1092.715466		1103.080483	1102.897656	1090.989672		1092.715466	
Enthalpy H2O liq	Btu/lbm	-->	38.6007522		62.58390214	62.15875865	34.6263914		38.6007522	
Density	lbm/ft3	-->	62.29469691		62.05877289	62.06379807	62.32352513		62.29469691	
Flow Rate	Agpm	-->	1.400148718		1.405471555	1.427058714	0.02161052		1.400148718	
Flow Rate	Sgpm	-->	1.4		1.4	1.421618225	0.021618225		1.4	
Flow Rate	lbm/min	-->	11.65986403		11.65986403	11.83991086	0.180046828		11.65986403	
Pressure	psia	-->	16.19595		16.19595		14.69595		14.69595	
Water Out/In	lbm/min	-->		0	lbm/min		evaporation	0.247531335	lbm/min	P-18
								75.18337424	°F	
								1094.74619	Btu/lbm	
							drain	-0.06748451	lbm/min	-P-11

Water Loop										
				heat added				heat removed		
Heat Transfer	Btu/min	-->		359.6970503				97.55238901		
Material	Unit	-->	P-8	E4: Air Cooler	P-5	P-5+P-10	P-10	E6: Cooling Tower	P-4	
Temperature	°F	-->	70.5		101.3729006	100.8428217	66.5260576		70.5	
Enthalpy H2O vap	Btu/lbm	-->	1092.715466		1106.027261	1105.800376	1090.989672		1092.715466	
Enthalpy H2O liq	Btu/lbm	-->	38.6007522		69.44991556	68.92036216	34.6263914		38.6007522	
Density	lbm/ft3	-->	62.29469691		61.9737997	61.98060478	62.32352513		62.29469691	
Flow Rate	Agpm	-->	1.400148718		1.407398618	1.428974179	0.02161052		1.400148718	
Flow Rate	Sgpm	-->	1.4		1.4	1.421618225	0.021618225		1.4	
Flow Rate	lbm/min	-->	11.65986403		11.65986403	11.83991086	0.180046828		11.65986403	
Pressure	psia	-->	16.19595		16.19595		14.69595		14.69595	
Water Out/In	lbm/min	-->		0	lbm/min		evaporation	0.247531335	lbm/min	P-18
								75.18337424	°F	
								1094.74619	Btu/lbm	
							drain	-0.06748451	lbm/min	-P-11

Water Loop										
				heat added				heat removed		
Heat Transfer	Btu/min	-->		359.6970503				97.55238901		
Material	Unit	-->	P-8	E4: Air Cooler HMX_4	P-5	P-5+P-10	P-10	E6: Cooling Tower	P-4	
Temperature	°F	-->	70.5		94.5	94.16799279	66.5260576		70.5	
Enthalpy H2O vap	Btu/lbm	-->	1092.715466		1103.080483	1102.937865	1090.989672		1092.715466	
Enthalpy H2O liq	Btu/lbm	-->	38.6007522		62.58390214	62.25225921	34.6263914		38.6007522	
Density	lbm/ft3	-->	62.29469691		62.05877289	62.06269539	62.32352513		62.29469691	
Flow Rate	Agpm	-->	1.800990138		1.807836823	1.829423906	0.02161052		1.800990138	
Flow Rate	Sgpm	-->	1.800798844		1.800798844	1.822417069	0.021618225		1.800798844	
Flow Rate	lbm/min	-->	14.99790691		14.99790691	15.17795374	0.180046828		14.99790691	
Pressure	psia	-->	16.19595		16.19595		14.69595		14.69595	
Water Out/In	lbm/min	-->		0	lbm/min		evaporation	0.247531335	lbm/min	P-18
								75.18337424	°F	
								1094.74619	Btu/lbm	
							drain	-0.06748451	lbm/min	-P-11

Engineering, Purchasing and Fabrication of the Prototype

Figure 15 shows the revised PDHS prototype with components ordered/purchased for testing at GTI Energy laboratory. The nominal values of the design parameters of the flows are shown in the figure.

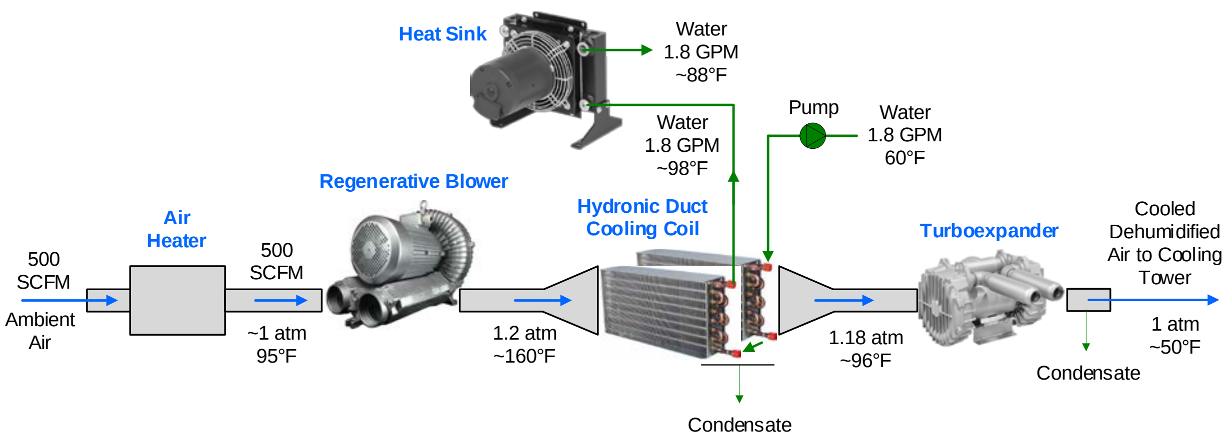


Figure 15. Revised PDHS prototype for testing at GTI Energy laboratory

The following table shows description of materials for the test PDHS. GTI Energy specified vendors, purchased the equipment, assembled the PDHS and tested the PDHS in GTI Energy laboratory. PDHS test plan and data collection protocols were developed for testing.

Table 11. Test PDHS components for purchasing

Item Name	Supplier	Web Link	Item #	Parameters
Air Heater	McMaster-Carr	Insertion Heater with Standard Wire Lead Covering, for 3/4" Hole, 240V AC, 10" Long Heating Element, 2000W | McMaster-Carr	35025K544	Ten Insertion Heaters for 3/4" Hole, 240V AC, 10" Long Heating Element, 2000W
Regenerative Blower	Atlantic Blowers	Atlantic Blowers | Regenerative Blowers || AB-1300	AB-1300	Max. Flow: 791 CFM Max. Pressure: 169” H_2O
Hydronic Duct Cooling Coil	Granger	PRECISION COILS Hydronic Duct Heating and Cooling Coil, Slip & Drive, 1.8 gpm, Required Air Flow 600 cfm - 5GEN8|SP1011212N - Grainger	5GEN8	Four units in series 12 1/8” x 12 3/8” x 6” Pressure drop (one unit): 2.7” H_2O

Heat Sink	McMaster-Carr	https://www.mcmaster.com/heat-sinks/	3579K21	Cooling Capacity: 20,000 Btu/hr 14 5/16" x 13 13/16" x 13 13/16
High-Flow Low-Pressure Air Blower (as turboexpander)	McMaster-Carr	https://www.mcmaster.com/catalog/9960k66	9960K66	Max Flow: 795 CFM Max. Pressure: 105" H_2O
Ducting, tubing, piping	Grainger, McMaster-Carr	https://www.grainger.com/ https://www.mcmaster.com/		-
Controls, measurement and data acquisition components	Omega, McMaster-Carr	https://www.grainger.com/ https://www.mcmaster.com/		-

Figure 16 shows the process flow diagram of the PDHS prototype and the arrangement of instruments for measuring temperature (T), pressure (P) and relative humidity (RH). An air heater and a steam generator were installed at the blower inlet to simulate different ambient conditions at different temperature and air humidity.

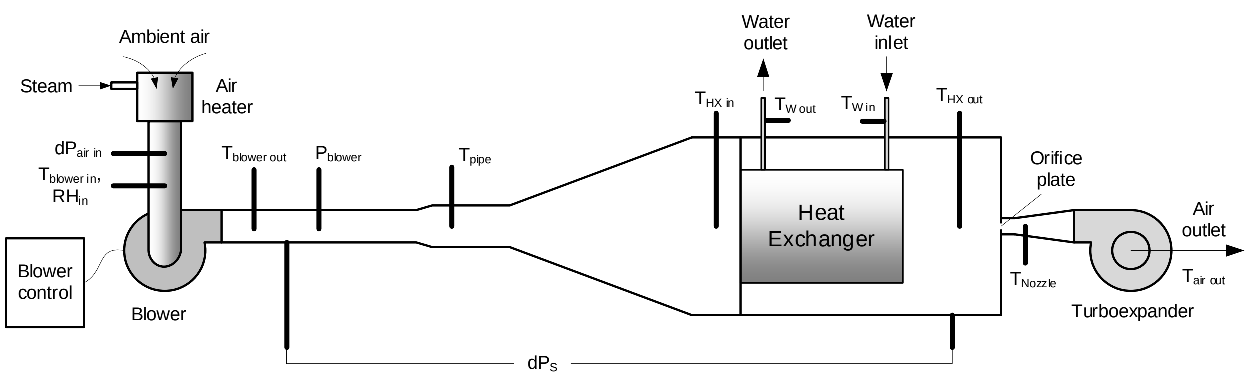


Figure 16. Process flow diagram of the PDHS prototype

Figure 17 illustrates PDHS prototype that has been assembled and installed at the GTI Energy Applied R&D Test Facility. Upon performance evaluation of PDHS at GTI Energy the further integration of SDPCT was planned to discussed and decided jointly by GTI Energy, BAC and NETL.

Figure 17. PDHS prototype during assembly and installation

Figure 18 shows the assembled PDHS prototype test bench at the GTI Energy Applied R&D test facility.

Figure 18. Assembled PDHS prototype test bench

Test Plan Development

The test plan was developed to accommodate typical ambient conditions for the U.S. climate zone with coal power plants. Figure 19, Figure 20 and Figure 21 show a U.S. coal-fired power plants map, major designated climate zones and ambient wet bulb temperatures in three selected representative cities in the U.S. major climate zones for coal-fired power plants. This information is used in the test plan of the experiments with the PDHS prototype.

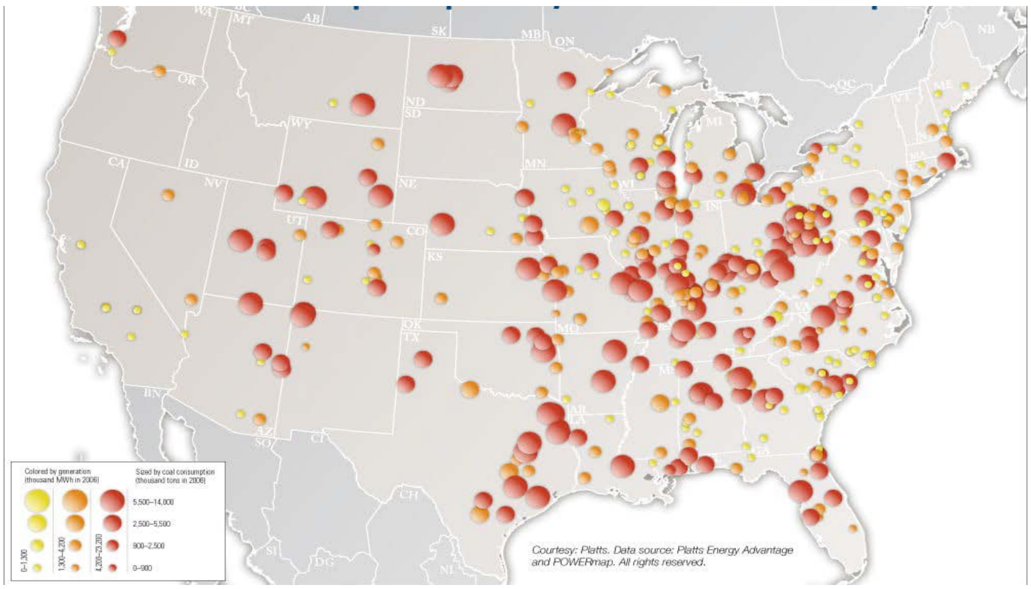


Figure 19. U.S. coal-fired power plants map by MWh and fuel consumption

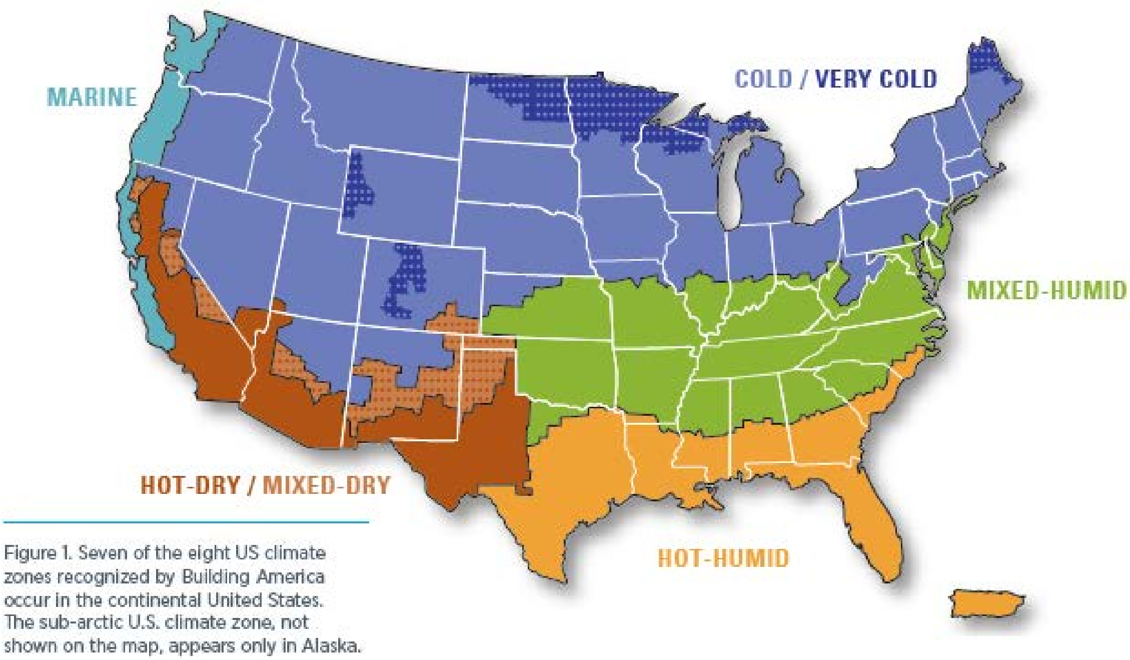


Figure 20. U.S. climate designations used by the International Energy Conservation Code (IECC) (https://www.energy.gov/eere/buildings/climate-zones)

Figure 21 shows three ambient wet bulb temperature T_{WB} occurrence in three selected representative cities of the U.S. major climate zones for coal power plants and the range of T_{WB} used in the experiments. Higher ambient wet bulb temperatures (above 48°F) were of interest in the experiments because cooling tower performance and power plant efficiency are generally negatively affected by higher ambient temperature and humidity.

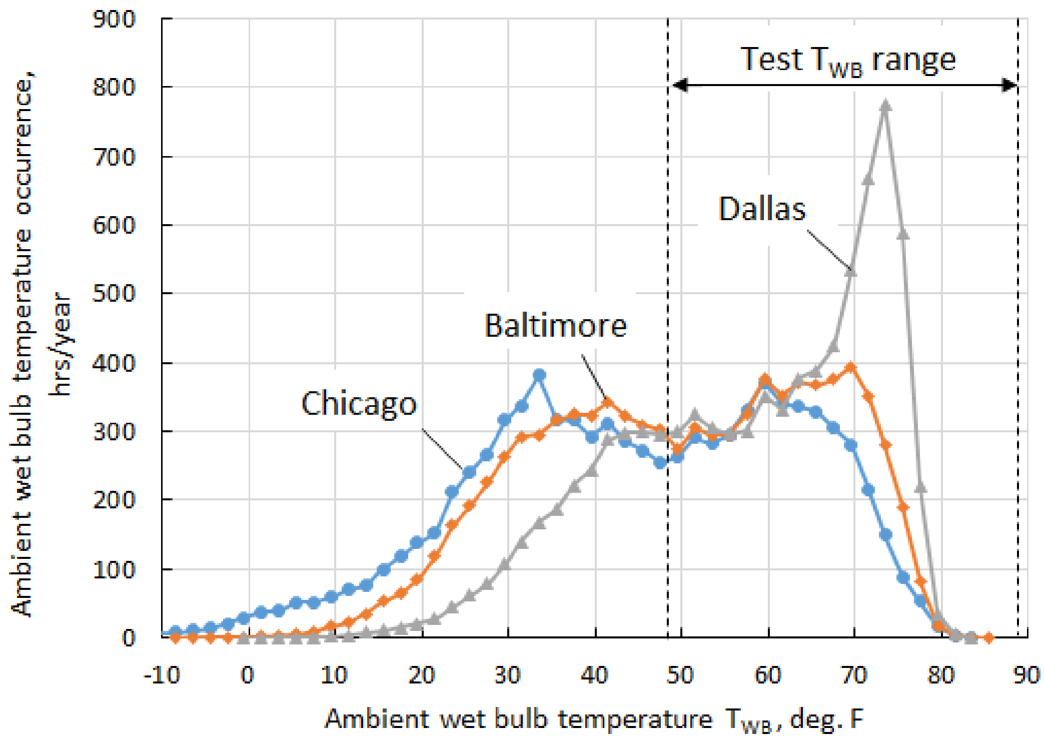


Figure 21. Ambient wet bulb temperatures in three representative cities in the U.S. major climate zones for coal power plants (courtesy by BAC) and test west bulb temperature T_{WB} range

The design parameters of the PDHS prototype were as follows:

- PDHS cooling capacity: up to 16 kW
- blower pressure ratio: 1.01-1.2
- air flow rate: 200-700 cfm
- cooling water inlet temperature: 43°F-65°F

During the experiments, ambient air conditions were simulated using an electric heater and an air humidifier (steam generator) installed in front of the blower. This made it possible to change the temperature and humidity of the air at the inlet to the blower, and the parameters of the incoming air were:

- dry bulb: 66°F-107°F
- relative humidity: 16%-78%
- wet bulb: 49°F-89°F
- dew point: 24°F-87°F

PDHS prototype test results

Figure 22 shows air wet bulb temperature versus air dry bulb temperature at the blower inlet measured during the experiments.

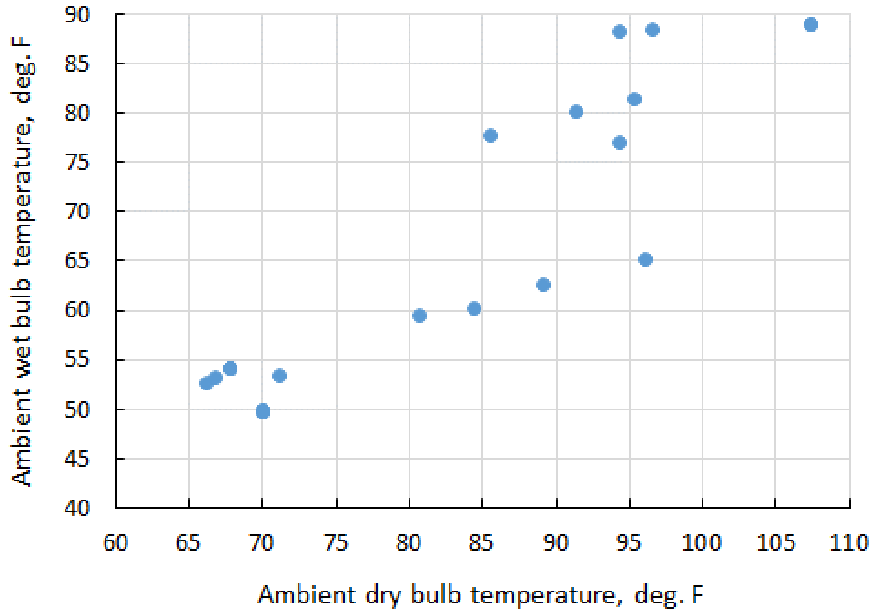


Figure 22. Tested ambient wet bulb temperature versus dry bulb temperature

Table 12 summarizes the PDHS test results, including measured condensate collection and water savings with PDHS. Some of the test data has been evaluated based on the experimental results.

Table 12. Summary of the pilot-scale PDHS prototype test results

Test point	Blower inlet air					Pressure ratio	Blower power		Expander power gen. (eff. 85%)	Water flow rate	Water inlet temp.	Cooled air (orifice, meas.)	PDHS cooling capacity	COP	Wet bulb approach	Dew point approach	Water saving
	T_{WB}	T_{DB}	RH	T_{DP}	Flow rate		(meas.)	(eff. 85%)									
	°F	°F	%	°F	CFM	-	kW	kW	kW	GPM	°F	°F	kW	-	°F	°F	%
1	49.6	70.1	17.9	24.5	428.4	1.227	9.177	5.079	3.451	2.4	43.4	63.8	3.677	2.26	14.2	39.3	-
2	49.6	70.1	17.9	24.5	415.8	1.225	9.177	4.892	3.354	2.3	43.8	70.2	2.967	1.93	20.6	45.7	-
3	49.7	70.1	18.1	24.8	439.1	1.242	9.616	5.517	3.717	2.0	43.6	61.6	4.409	2.45	11.9	36.8	-
4	49.7	70.1	18.1	24.8	428.4	1.238	9.616	5.308	3.577	2.0	43.3	60.1	4.198	2.42	10.4	35.3	-
5	52.5	66.2	38.3	40.1	329.0	1.102	4.061	1.826	1.278	0.9	51.1	60.0	1.520	2.77	7.5	19.9	-
6	53.0	66.9	38.2	40.6	208.9	1.102	3.110	1.164	0.820	0.9	64.5	66.9	0.726	2.11	13.9	26.3	-
7	53.4	71.2	27.9	36.3	344.4	1.157	5.583	2.889	1.958	1.9	45.1	54.3	3.232	3.47	0.9	18.0	-
8	54.0	67.8	39.3	42.2	209.0	1.101	3.110	1.155	0.793	2.0	51.9	55.0	1.625	4.49	1.0	12.8	-
9	54.0	67.8	39.3	42.2	209.0	1.101	3.110	1.155	0.787	2.0	47.7	56.2	1.706	4.63	2.2	14.0	-
10	54.0	67.8	39.3	42.2	209.0	1.101	3.110	1.155	0.792	1.0	48.0	59.0	1.658	4.57	5.0	16.8	-
11	54.0	67.8	39.3	42.2	209.0	1.101	3.110	1.155	0.805	0.5	49.1	63.9	1.088	3.11	9.9	21.7	-
12	54.0	67.8	39.3	42.2	209.0	1.101	3.110	1.155	0.810	0.5	49.8	66.2	0.904	2.62	12.2	24.0	-
13	59.3	80.7	25.9	42.6	211.6	1.100	3.110	1.152	0.791	1.0	63.7	67.3	1.623	4.50	8.0	24.7	-
14	62.5	89.2	20.1	43.0	213.2	1.099	3.110	1.150	0.778	1.0	63.6	67.5	2.114	5.69	5.0	24.5	-
15	65.1	96.1	16.5	43.5	214.6	1.097	3.110	1.141	0.764	0.9	63.6	67.9	2.551	6.75	2.8	24.4	-
16	60.0	84.5	21.3	40.7	334.7	1.099	4.033	1.807	1.230	0.9	50.5	47.6	3.024	5.24	-12.4	6.9	-
17	76.8	94.4	45.2	69.9	527.4	1.104	6.107	3.019	2.036	1.9	48.6	72.4	9.027	9.18	-4.4	2.5	0.4
18	77.7	85.6	70.3	74.8	328.8	1.097	4.047	1.769	1.222	0.8	50.3	72.3	4.870	8.90	-5.4	-2.5	3.2
19	80.0	91.4	61	76.0	336.8	1.097	4.018	1.809	1.236	0.8	53.0	72.0	5.854	10.21	-8.0	-4.0	5.6
20	81.2	95.4	54.7	76.5	331.8	1.097	4.018	1.783	1.209	0.8	50.1	73.6	6.246	10.88	-7.6	-2.9	7.0
21	88.2	94.4	78.2	86.5	214.2	1.095	3.053	1.139	0.775	0.8	55.1	72.3	5.643	15.49	-15.9	-14.2	28.2
22	88.4	96.7	72.1	86.2	214.7	1.095	3.053	1.143	0.771	0.8	53.5	72.4	6.167	16.54	-16.0	-13.8	32.5
23	88.8	107.4	48.5	83.8	430.2	1.223	8.428	5.173	3.349	2.3	43.6	77.9	15.821	8.67	-10.9	-5.9	28.7
24	88.8	107.4	48.5	83.8	426.9	1.229	9.360	5.212	3.371	2.3	43.6	77.9	15.814	8.59	-10.9	-5.9	28.5

Figure 23, Figure 24 and Figure 25 are graphical representations of the test results taken from Table 12. Figure 23 shows how much the air temperature before cooling tower can be reduced with PDHS compared to the ambient dew point temperature. The negative dew point approach (the difference between the air temperature at the outlet of PDHS and the ambient dew point) in PDHS means that the air is cooled below the ambient dew point and the air is dehumidified. The higher the ambient wet bulb temperature, the lower the dew point approach. Based on PDHS performance data analysis supported by engineering simulations, the ability to provide targeted cooled water temperatures at least 1°F below ambient dew point has been confirmed.

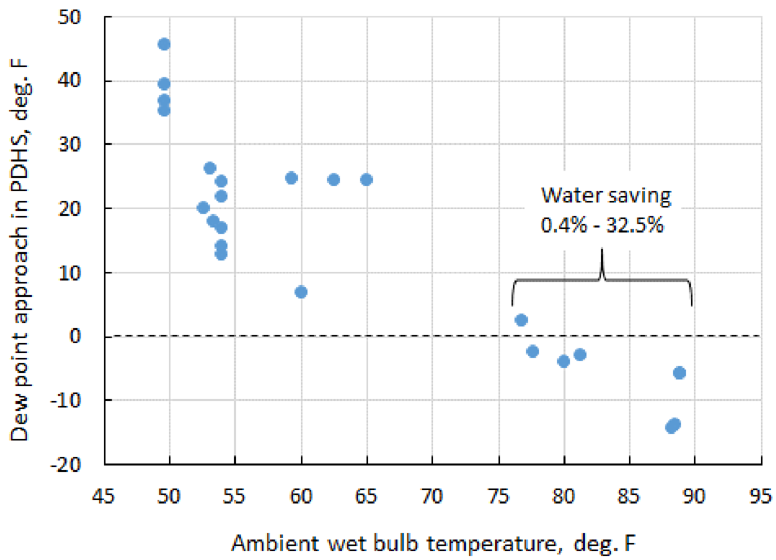


Figure 23. Measured dew point approach in PDHS

Figure 24 shows water savings versus ambient wet bulb temperature. Water savings were estimated as the amount of measured condensate relative to the calculated amount of make-up water in the cooling tower. As can be seen from the figure, the estimated water savings were more than 20%, which is higher than originally predicted at the concept stage.

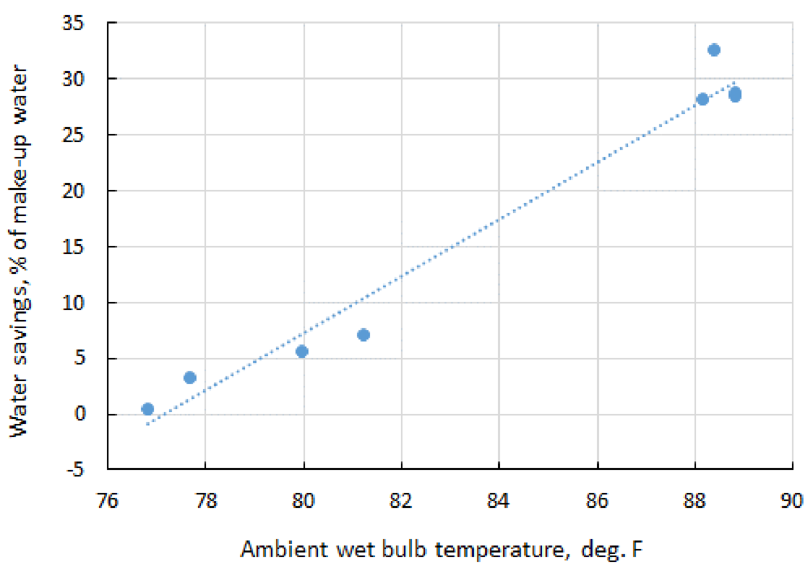


Figure 24. Water savings by using PDHS to cool and dehumidify air in a cooling tower

Figure 25 shows coefficient of performance (COP) of the PDHS estimated as

$$COP = Q_c / (P_b - P_e).$$

Here Q_c is the cooling capacity of the PDHS, P_b is the electrical power consumed by the blower, P_e is the electric power generated by the turbo-expander. As can be seen from the figure, the higher the ambient wet bulb temperature, the higher the COP of PDHS, achieving the COP above 16.

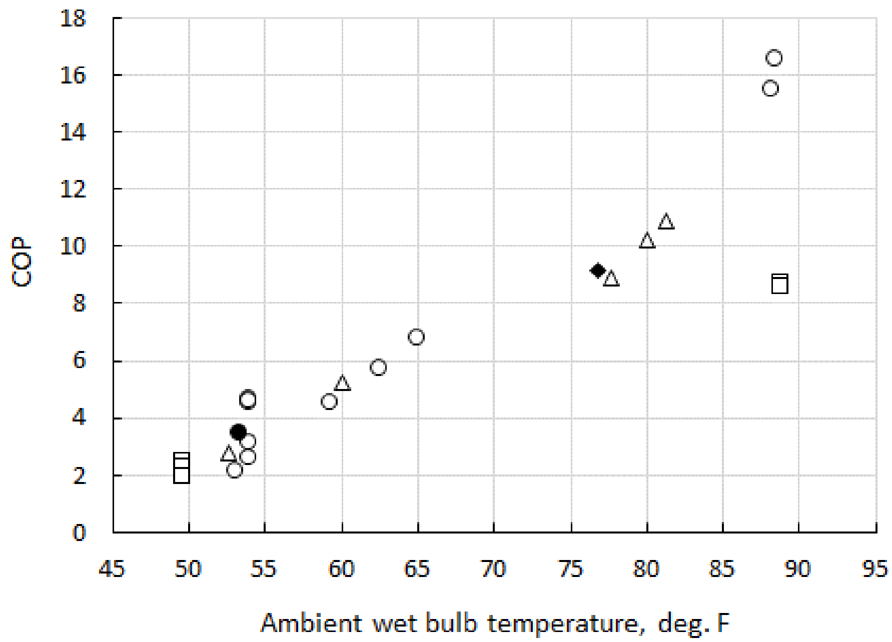


Figure 25. Coefficient of Performance (COP) of the PDHS

PDHS and SDPCT Integration

GTI Energy met with BAC at GTI Energy test site to observe the PDHS test facility, discuss the results of the PDHS prototype testing, and discuss the possibility of SDPCT simulation instead of PDHS testing with the BAC cooling tower at BAC's facility. During the meeting, it was decided that PDHS would not be tested at BAC's facility, but instead BAC would use GTI Energy's experimental data in its in-house modelling software to simulate SDPCT. GTI Energy would provide BAC with a detailed description of the test conditions and test data, and will provide BAC with a PDHS mathematical model for inclusion in BAC's in-house software to further simulate the enhanced PDHS cooling tower. A joint meeting of representatives of GTI Energy, NETL and BAC was held to confirm SDPCT integration approach and performance simulations.

PDHS model and modeling results

Figure 26 and Figure 27 show two possible layouts for the PDHS system. In the first layout, sensible cooling of air is used after the blower. In the second layout, water cooling is used after the Heat Exchanger (HX)-Condenser. Cooling tower exhaust air (dashed line in the diagrams) can also be used to cool pressurized air or water when the cooling tower exhaust air is colder than the ambient air.

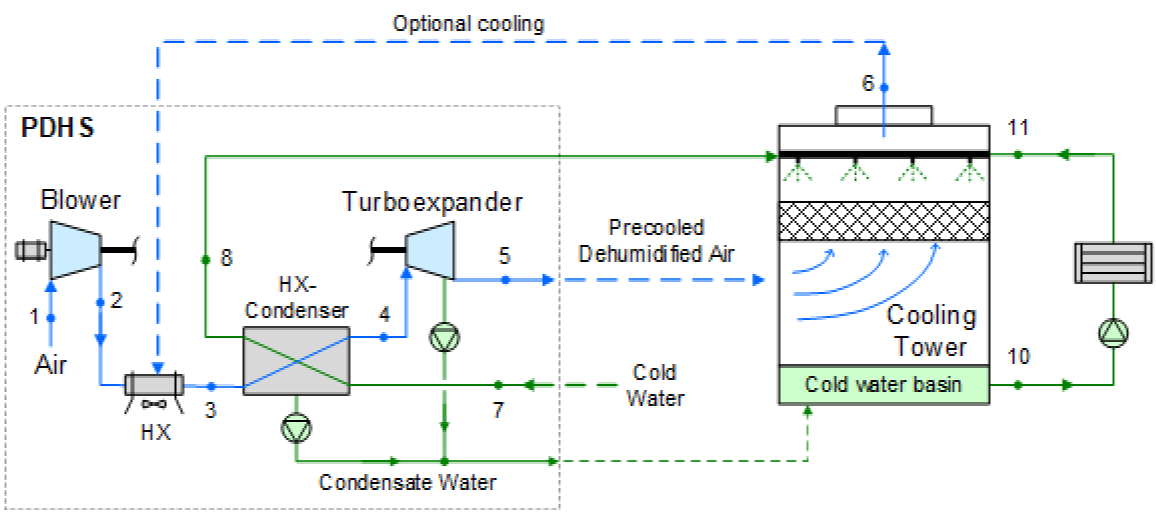


Figure 26. PDHS with sensible cooling of compressed air

A blower in the PDHS pressurizes incoming air and increases the dew point of the air. This makes it easier for the system to remove moisture from the air. The air is cooled and the moisture is partially removed in the HX-Condenser. An expander is used to compensate for the power consumed by the blower. The turboexpander also removes additional moisture from the air as the air temperature in the turboexpander decreases. This allows the system to reduce power consumption, achieve air temperatures below the ambient dew point, and dehumidify the air before air enters the cooling tower.

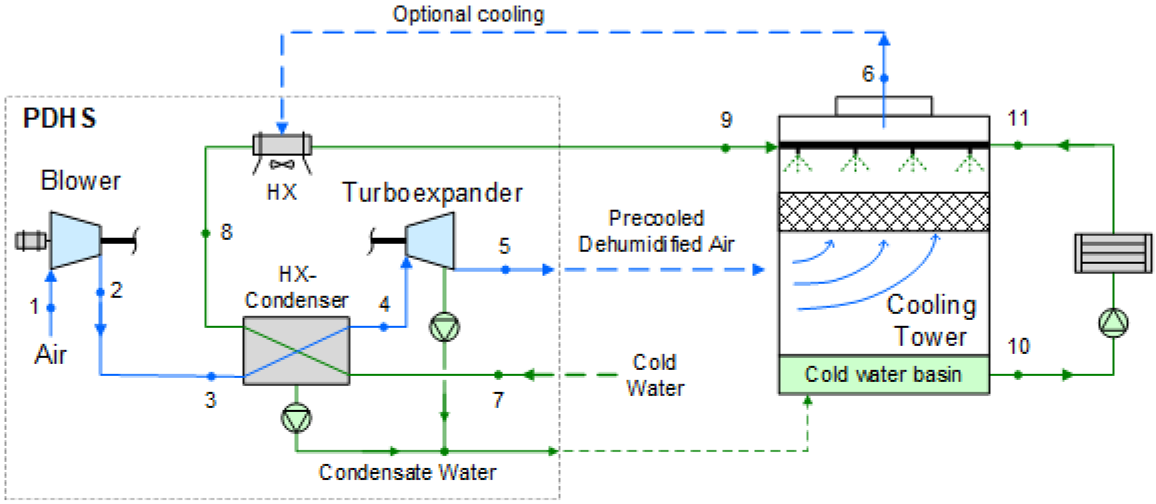


Figure 27. PDHS with sensible cooling of water after HX-condenser

The following equations were used to simulate the PDHS.

<u>Nomenclature</u>:
c_p – specific heat capacity at constant pressure, J/kg-K
c_v – specific heat capacity at constant volume, J/kg-K
h – enthalpy, J/kg
$k = c_p/c_v$ = specific heat ratio
m_a – dry air mass flow rate, kg/s
P – air pressure, Pa
PR – pressure ratio
RH – relative humidity (0 - 1.0)
T – temperature, K
W – power, W
w – humidity ratio, kg water per kg dry air
η – efficiency

1. Blower inlet (ambient air): m_1, P_1, T_1, h_1, RH_1, w_1

   $m_1 = m_a*(1+w_1)$

2. Blower outlet (pressurized air): m_2, P_2, T_2, h_2, RH_2, w_2, η_{blower}

   $m_2 = m_1$

   $w_2 = w_1$

   RH_2 depends on T_2 and P_2

   $P_2 = P_1*PR_{blower}$

   PR_{blower} = 1.05-1.3

   $T_2 = T_1 + T_1*(PR_{blower}^{(k_1-1)/k_1} - 1) / \eta_{blower}$

   $\eta_{blower} = 0.93$

$W_{blower} = m_1*[cp_1*T_1/\eta_{blower}*(PR_{blower}^{(k_1-1)/k_1} - 1) + w_1 * (h_2 - h_1)]$

3. Air Heat Exchanger (HX) outlet (layout I): m_3, P_3, T_3, h_3, RH_3, w_3

   $m_3 = m_2$

   $w_3 = w_2$

   RH_3 depends on T_3 and P_3

   $P_3 = P_2 - dP_{HX}$, dP_{HX} = 250 Pa

   $T_3 = T_{amb}$ + 2°K or $T_3 = T_6$ + 2°K if cooling tower exhaust air is used for cooling

   $Q_{HX} = m_2*(h_2 - h_3)$

4. HX-Condenser outlet: m_4, P_4, T_4, h_4, RH_4, w_4

   $m_4 = m_a*(1+w_4)$

   $w_4 \leq w_3$, $w_4 = w_3$ if $T_4 > (T_{DP})_4$

   RH_4 = 1.0 if $T_4 = (T_{DP})_4$

   $P_4 = P_3 - dP_{HXC}$, dP_{HXC} = 250 Pa

   $T_4 = T_7$ + 2°K

   $Q_{HXC} = m_3 * (h_4 - h_3)$

   Condensate mass flow rate: $m_{cond4} = m_a*(w_4 - w_3)$

5. Turboexpander outlet: m_5, P_5, T_5, h_5, RH_5, w_5

   $m_5 = m_a*(1+w_5)$

   $w_5 \leq w_4$, $w_5 = w_4$ if $T_5 > (T_{DP})_5$

   RH_5 = 1.0 if $T_5 = (T_{DP})_5$

   $P_5 = P_{amb} + dP_{CT}$, where dP_{CT} is cooling tower air pressure drop

   $PR_t = P_4/P_5$

   Temperature T_5 with no condensation in turboexpander:

   $T_{5,\, ideal} = T_4/(PR_t^{(k4-1)/k4})$

   $T_5 = T_4 - \eta_t*(T_4 - T_{5,\, ideal})$

   Temperature $T_{5,\, cor}$ with condensation in tuboexpander can be found from the following equations:

   $T_{5,\, cor} = T_4 - (Q_5 - m_{cond5}*H_{vap})/(m_4*c_{p4})$,

   where $H_{vap} = 2453*10^3$ J/kg

   $Q_5 = m_4*(h_5-h_4)$, here h_5 is calculated based on T_5, P_5 and RH_5

   Condensate mass flow rate: $m_{cond5} = m_a*(w_5 - w_4)$

   η_t = 0.9

$W_t = m_4*(1+w_1)*cp_4*\eta_t*T_4*(1-1/PR_t^{(k_4-1)/k_4})$

Condensate mass flow rate: $m_{cond5} = m_a*(w_5 - w_4)$

6. Cooling tower exhaust air: m_6, P_{amb}, T_6, h_6
7. Cold water inlet: m_7, T_7, h_7

   The cold water for cooling in the HX-Condenser is taken from the cooling tower water basin and it can also be mixed with the make-up water if it is colder than the water in the basin.

   $m_7 = m_8$ (see the following how to calculate m_8)
8. Cold water outlet: m_8, T_8, h_8

   $T_8 = T_3 - 2°K$

   $m_8 = Q_{HXC} /(h_8 - h_7)$
9. Water Heat Exchanger outlet (layout II): m_9, T_9, h_9

   $T_9 = T_{amb} + 2°K$ or $T_9 = T_6 + 2°K$ if cooling tower exhaust air is used for cooling

   $m_9 = m_8$

PDHS power consumption: $W_{PDHS} = W_{blower} - W_t$

Based on this system of equations, a computational model was developed in BAC to simulate the performance of the integrated system, including a blower, an air-cooled heat exchanger, a heat and mass exchanger, a turbo-expander and a cooling tower. This model made it possible to investigate the system performance under various ambient conditions, for various system layouts, and for various capacities of each component.

The PDHS model was validated using the test data from GTI Energy (Figure 28). The predicted air temperatures at the blower outlet and at the turbo-expander outlet match very well with those from the testing.

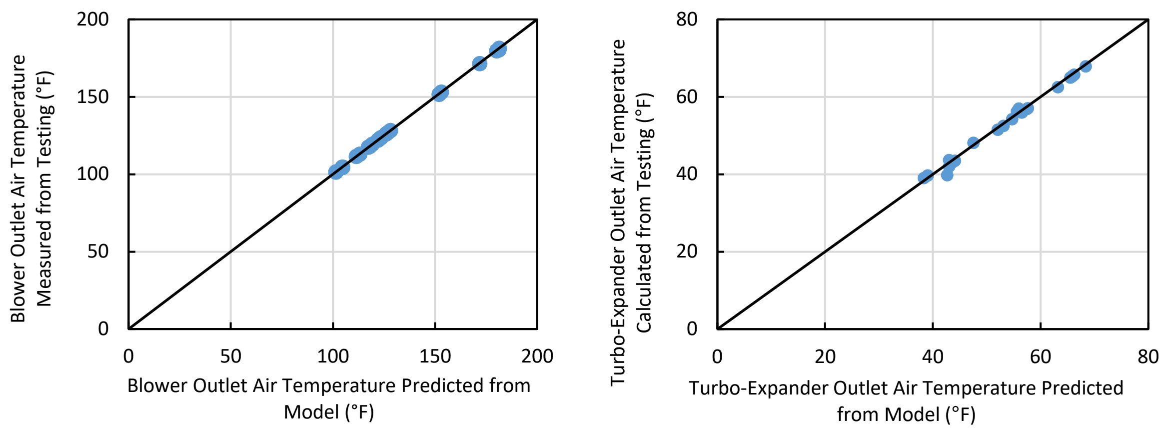


Figure 28. Comparison of the PDHS model simulations with experimental data

As the next step, the PDHS model was integrated into BAC's proprietary cooling tower model to investigate the cooling capacity and energy consumption of the system. The cooling tower model includes modules for calculating air flow and heat and mass transfer during evaporation over a discrete heat transfer surface area. It is able to predict the cooling capacity of the cooling towers at various ambient conditions, at different box sizes, at different fan horsepower, and at different thermal duties. The model has been validated by full-scale product testing, and the predicted cooling tower capacity has been certified by Cooing Technology Institute (CTI) with an error tolerance less than or equal to 5%.

Based on the model described above, BAC performed calculations for each PDHS test point and estimated SDPCT performance. The calculations gave the following results.

- Confirmed water savings (reduction in make-up water consumption) up to 20% - 33% for all selected climate conditions.
- Up to 100% water savings (reduction in make-up water consumption) is achievable at favorable conditions.
- A higher dehumidification rate achieved with a higher blower pressure ratio.
- Confirmed sub-dew point cooling: 2.5°F - 4.2°F below ambient dew point was achieved.
- SDPCT coefficient of performance: up to 4.35.
- Cooling tower with the PDHS can run at much lower temperature of cold water (45°F - 70°F) compared to the cooling tower without the PDHS (80°F - 91°F).

PDHS layout for a 650MW coal power plant

Figure 29 shows a PDHS design for a 650 MW power plant cooling tower. The dimensions of the PDHS layout and the number of PDHS units are mainly dependent on the capacity of the blower and expander, as well as the allowable system footprint. As an example, Figure 30 shows a blower with a pressure of 30kPa (0.3bar) and an air flow rate of up to 1,000 m3/s. The blower tip diameter is 3.5m. The PDHS requires 6 to 13 blowers (depending on the cooling tower design) for a 650MW coal power plant. The turboexpander may be similar in design to the blower blade and is directly coupled to the blower via a central shaft.

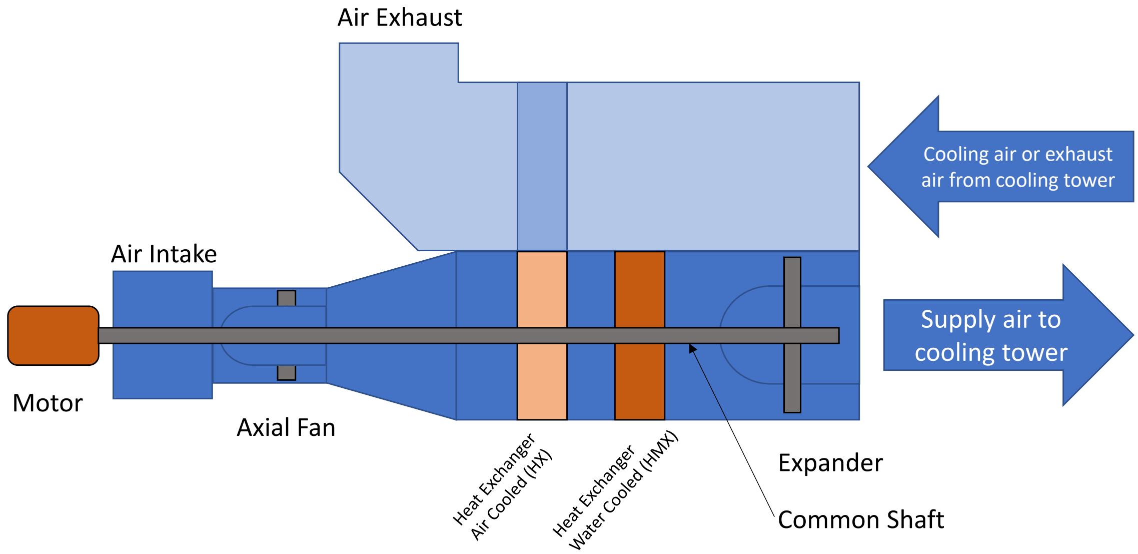


Figure 29. PDHS scheme for a cooling tower for a 650MW coal power plant

Figure 30. Adjustable pitch axial blower

For air-to-air heat exchange, a heat pipe based heat recovery unit (Figure 31) would result in the most compact heat exchanger design. Dimensions of the unit: 5m×5m×1.5m.

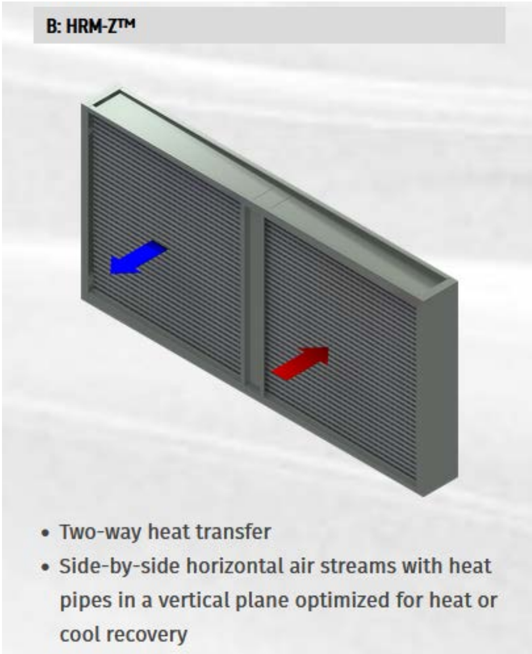


Figure 31. A heat pipe based heat recovery unit (HX)

For air-to-water heat exchange, a fined tube heat exchanger unit (Figure 32) would result in the most compact heat exchanger design. Dimensions of the unit: 5m×5m×0.75m.

Figure 32. An air-to-water heat exchanger unit (HMX)

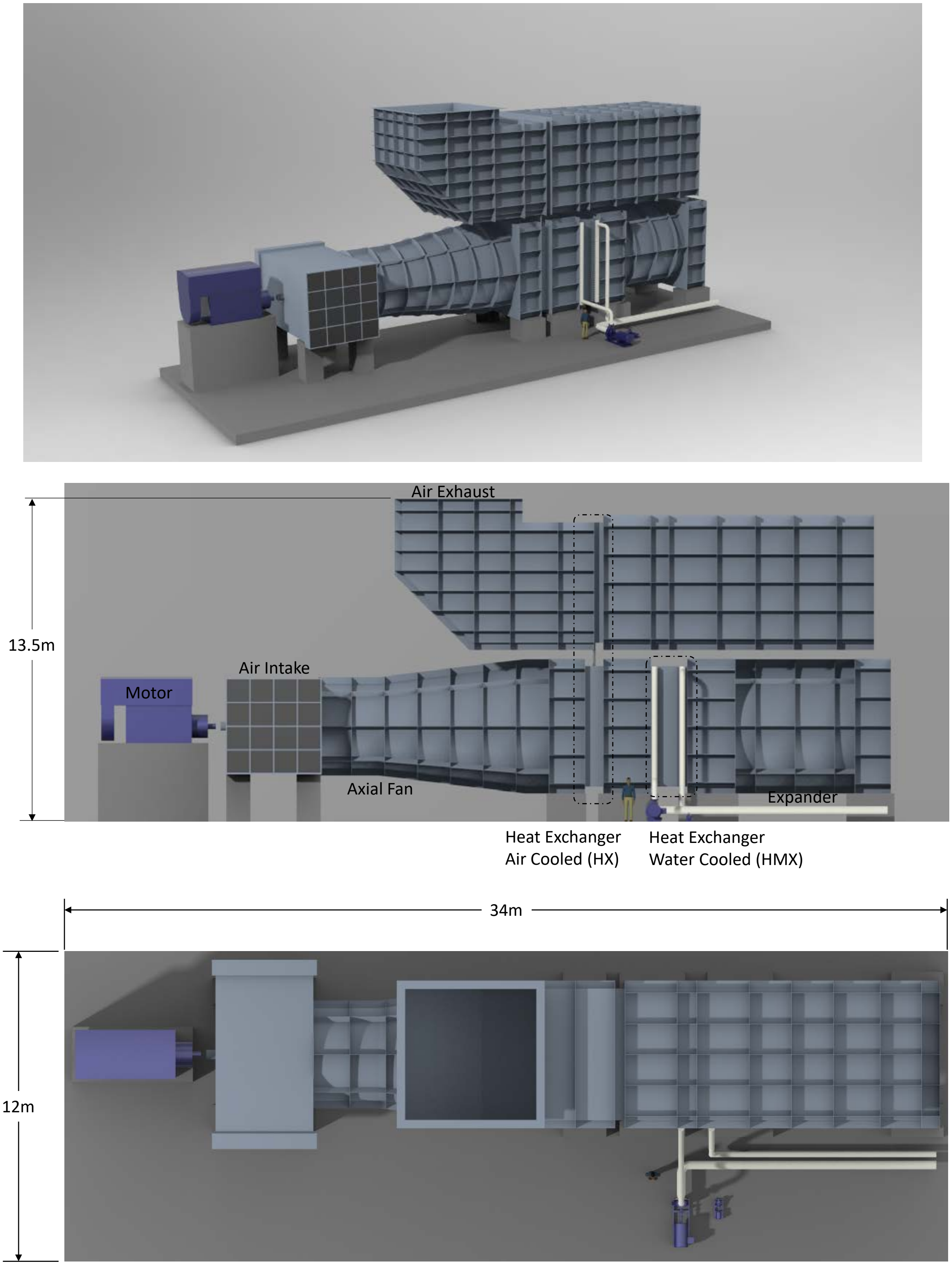


Figure 33. PDHS Layout for a 650MW coal power plant cooling tower

Basis of assessment

The Techno-Economic Assessment (TEA) includes the evaluation of integrating Sub-Dew Point Cooling Tower (SDPCT) into a coal-fired power plant, the associated capital costs, and life cycle costs for such retrofit. A hypothetical supercritical power plant based on DOE/NETL Fossil Fuel Baseline Case 12A[1] is used as a basis for these investigations. A thermal model was developed to replicate the steam turbine cycle performance by using the same steam parameters as the DOE Baseline Case 12A. The thermal model was used to predict the power plant performance with Pressure Dehumidifying System (PDHS) integration. The ambient conditions used in DOE Baseline Case 12A are at ISO conditions (59°F/15°C Dry Bulb temperature and 60% relative humidity). Since the proposed PDSH retrofit is mainly applicable for high ambient temperature operation, the following site conditions are used in the TEA to evaluate the potential benefit of PDSH system (Table 13).

Table 13. Site characteristics

Dry Bulb Temperature, °C (°F)	**35 (95)**
Wet Bulb Temperature, °C (°F)	25.5 (77.9)
Relative Humidity, %	47.0

The thermal model developed based on DOE Baseline Case 12A is used to predict the power plant off-design performance at the assumed site ambient conditions. The predicted results are used as a baseline to compare the performance impacts with PDSH retrofit.

The expected performance and design parameters of the PDSH system were simulated by GTI Energy based on test data gathered from the prototype test setup. The simulation results were provided to Worley/Advisian as the basis for performing the TEA.

Power plant integration evaluation

The PDHS system can be integrated into a power plant as shown in Figure 34. The integration is mainly applied to the existing cooling tower. The remaining equipment of the existing cooling water system, including the steam turbine condenser and circulating pumps can be reused without modification. However, the turbine condenser in DOE Case 12A has a relatively high design Terminal Temperature Differences (TTD). To maximize the benefit of lower cooling water temperature with PDSH retrofit, an additional case to replace the existing condenser with a lower TTD is also included in the TEA.

1 Cost and Performance Baseline for Fossil Energy Plants, Volume 1: Bituminous Coal and Natural Gas to Electricity, Rev. 4, NETL-PUB-22638, September 24, 2019.

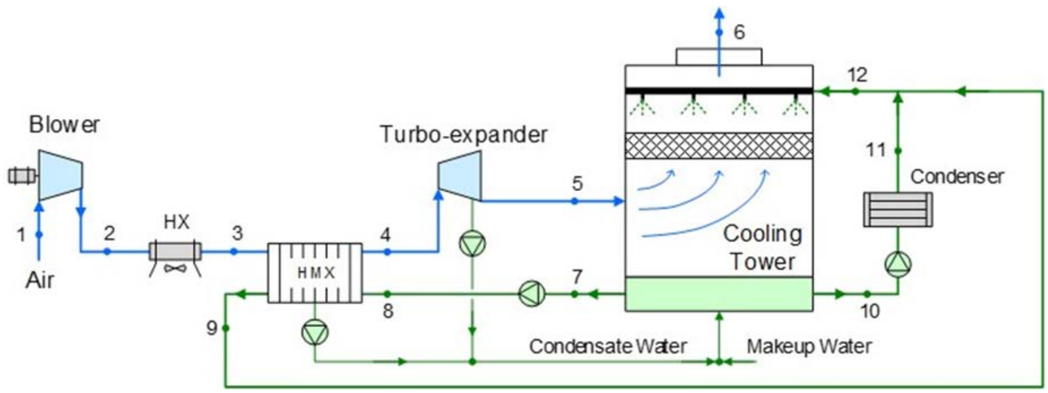


Figure 34. Potential integration of power plant and PDHS

GTI Energy provided the following design parameters for the analysis of the PDHS (Table 14).

Table 14. PDSH System Design Parameters

Stream # (see Figure)	**Unit**	**Air**				
		1	2	3	4	5
Flowrate	kg/s	12259	12259	12259	12226	12142
Temperature	°C	35	44.2	37.0	23.1	13.2
Relative humidity	%	47.3	32.2	47.20	100	100
Absolute pressure	kPa	101.3	111.4	111.2	110.9	101.4

Stream # (see Figure)	**Unit**	**Water**					
		7	8	9	10	11	12
Flowrate	kg/s	3673	3673	3673	14467	14467	18140
Temperature	°C	16.0	16.0	35.0	16.0	28.2	29.6
Relative humidity	%	-	-	-	-	-	-
Absolute pressure	kPa	-	-	-	-	-	-

Stream # (see Figure)	**Unit**	**Condensate**			Evaporated water	Makeup water
		HMX	Expander	Total		
Flowrate	kg/s	33.2	83.5	116.7	235.82	254.1
Temperature	°C	23.3	18.2	19.6	-	-

There are four major pieces of equipment in the PDHS.

- Air Blower: to supply the ambient air to the PDHS.
- Air Cooled Heat Exchanger (HX): to slightly cool the air from the air blower discharge.
- Heat & Mass Exchanger (HMX): to further cool air temperature and condense some moisture from the air by using a portion of cooling water from the cooling tower.
- Turbo Expander: to recover mechanical power while reduce the air temperature below dew point.

The expected overall power plant performance with PDSH retrofit comparing to the baseline cases are summarized in Table 15. The comparisons are based on the PDHS operating parameters predicted by GTI Energy. The results show about 30% reduction of cooling tower make-up water consumption at the evaluated ambient conditions for the PDHS retrofit in comparison with the baseline case. The steam turbine gross outputs also increase by about 23-27MW (3.4%-4.1%) due to lower steam turbine back pressure for the PDHS cases. The overall plant net outputs also increase slightly by 1.6MW for the case of reutilizing the existing condenser and 6 MW for the case with condenser upgrade. However, the plant net output impact for the PDHS system will greatly depend on the air blower and expander efficiencies.

Table 15. Predicted power plant performance with PDSH Retrofit

Case Description	**Baseline**		**PDHS Retrofit w/o Condenser Upgrade**		**PDHS Retrofit w/ Condenser Upgrade**	
1. Ambient Conditions						
Dry Bulb Temperature, °C (°F)	35 (95)					
Wet Bulb Temperature, °C (°F)	25.5 (77.9)					
Relative Humidity, %	47.0					
2. Cooling System Operating Parameters						
Condenser Pressure, bar (in Hg)	0.1041	3.07	0.0548	1.62	0.0457	1.35
Condenser Circulating Water Flow, kg/s (lb/s)	14382	31706	14382	31706	14328	31587
Condenser Cold Inlet Temperature, °C (°F)	28.3	82.9	16.0	60.8	29.6	85.2
Condenser Hot Outlet Temperature, °C (°F)	40.8	105.5	28.1	82.7	28.1	82.6
Condenser Duty, MWth (MMBtu/hr)	753.4	2572.9	730.0	2492.9	-86.9	-296.6
Condenser TTD, °C (°F)	6.9	12.4	6.4	11.5	3.2	5.7
Cooling Tower Circulating Water Flow, kg/s (lb/s)	14802	32632	18140	39991	18086	39872
Cooling Tower Inlet Air WB Temperature, °C (°F)	**25.5**	**77.9**	**13.2**	**55.8**	**13.2**	**55.8**
Cooling Tower Water Inlet Temperature, °C (°F)	40.2	104.4	29.6	85.2	29.6	85.2
Cooling Tower Water Outlet Temperature, °C (°F)	28.3	82.9	16.0	60.8	16.0	60.8
Cooling Tower Approach Temperature, °C (°F)	2.8	5.0	2.8	5.0	2.8	5.0

Case Description	Baseline		PDHS Retrofit w/o Condenser Upgrade		PDHS Retrofit w/ Condenser Upgrade	
Cooling Tower Range Temperature, °C (°F)	11.9	21.5	13.6	24.4	13.6	24.4
Cooling Tower Duty, MWth (MMBtu/hr)	738.3	2521.4	1029.4	3515.6	1029.4	3505.1
Cooling Tower Makeup Rate, kg/s (lb/s)	369.3	814.2	254.1	560.2	254.1	558.5
Cooling Tower Fan Power, MW	3.0		1.8		1.79	
Circulating Water Pump Power, MW	5.3		5.3		5.3	
PDHS Air Cooler Fan Power*, MW	0.0		4.7		4.7	
PDHS Air Blower Power*, MW	0.0		113.2		112.9	
PDHS Water Booster Pump Power, MW	0.0		0.3		0.3	
Air Expander Shaft Power Output*, MW	0.0		95.7		95.4	
Total Cooling System Aux Power Consumption, MW (Net of power consumed and generated)	8.3		29.6		29.5	
3. Plant Performance						
Steam Turbine Output, MW	666.5		689.4		693.7	
Auxiliary Power Excluding Cooling System, MW	27.0		27.0		27.0	
Plant Total Auxiliary Power, MW	35.3		56.6		56.5	
Plant Net Output, MW	631.2		632.8		637.2	
Plant Net Output Gain, MW	Base		1.6		6.0	
Plant Net HHV Efficiency, %	39.01%		39.15%		39.42%	
Plant Net HHV Efficiency Gain, %	Base		0.14%		0.41%	
4. Cooling Tower Water Consumption						
Makeup Water Consumption, gpm	5862		4034		4022	
Makeup Water Consumption Reduction, %	Base		31.2%		31.4%	

**: Based on data provided by GTI Energy.*

It is currently assumed that the isentropic efficiency of both blower and expander is 90%. It shall be noted that the high-speed blowers and expanders may not be suitable for the PDHS application due to relatively larger diameter of blower and expander stages, as well as due to their increased noise levels requiring expensive noise reduction measures at the proposed retrofit project sites. To counter the noise levels, a low speed design may be employed but such approach will further limit the maximum achievable efficiencies for the blower and expander.

An approximate calculation indicates that there will be no overall plant net power gain if the efficiency of the blower and expander is reduced by about 4-5% compared to the currently assumed values in the TEA. The low-pressure expanders most likely need to be custom designed. The availability and achievable performance of such expanders should be investigated with the potential blower & expander OEMs in the next phase of commercialization assessment.

The air-cooled HX and water-cooled HMX will be of conventional coil type design. There are numerous vendors in the market who could supply these types of heat exchanger. It is assumed that the major components of the PDHS system, including blower, HX, HMX and expander, will be assembled in a single module in the manufacturer's shop and shipped to the project site. The number of required PDHS modules will depend upon the total cooling duty of the cooling tower and with modularization to facilitate road transportation.

Capital and operating cost estimate

The integration of the PDHS into a power plant is assumed to be a retrofit project for this TEA as per definition given in the DOE/NETL's FOA. Therefore, the capital expenditure (CAPEX) estimate only covers the cost of a new PDHS system and associated retrofit to the existing plant cooling tower system. The following assumptions and basis are considered for the order of magnitude cost estimate.

- Equipment costs are assumed to be commercially available, and no additional allowances are included for the R&D of the first kind of technology.
- Assuming there is adequate space for the installation of new PDHS at the existing plant site.
- Assuming the underground is clear without the need of modifying existing buried underground piping and cables.
- Assuming no excessive noise attenuation is required at the project site.
- Assuming the main component of the PDHS will be modular design and shop fabricated/assembled. The modularization will be maximized to minimize the field installation.
- The equipment costs are mainly estimated based on Worley's in-house data by applying scaling factors for the capacities and sizes required by the retrofit project.

The order of magnitude capital costs of the PDHS retrofit are summarized in Table 16.

The operational expenditure (OPEX) includes fixed and variable operation & maintenance (O&M) costs. The following O&M costs are assumed and used in the economic analysis.

- Fixed O&M Costs: Assumed 1% of CAPEX annually to cover the operation and maintenance labor costs, external contractor's service charge, spare parts, etc.

- Variable O&M Costs: $50 per operating hours for the consumables, including lube oil, grease, compressed air, cleaning chemicals, washing water, etc. Electricity consumption is included in the new power generation thus excludes from the variable O&M costs.

Table 16. Cost estimate summary – PDHS retrofit

PDHS Capital Cost, Million USD			
Case Description	**w/o Condenser Upgrade**	**w/ Condenser Upgrade**	**Remarks**
Air Blowers	21.6	21.6	
Air Cooler	26.8	26.8	
HMX	13	13.0	
Turbo Expander	18	18.0	
Misc Water Booster Pumps	1.2	1.2	
Piping	1	1.0	
Existing Cooling Tower Modifications	5.4	5.4	
Electrical System Modification	4.5	4.5	
Existing Condenser Retrofit	0	10.1	New condenser with lower TTD
Total Retrofit CAPEX	91.5	101.6	Note
Total Retrofit CAPEX, $/kW	57,188	16,933	Incremental net output to Baseline

Note: CAPEX excludes Owner Cost. A 10% of estimated retrofit CAPEX is included in LCOE analysis separately.

Economic analysis

The economic analysis was performed to determine the levelized cost of electricity (LCOE) for the PDHS retrofit project. A Microsoft Excel-based model was developed to address costs on an annual basis and take into account the performance impact with retrofit. Additionally, a breakdown of the LCOE, by CAPEX, water saving, and operating and maintenance costs to provide understanding of the cost benefits and deficiencies. LCOE is represented by the cost at which the present value (PV) of all revenues from electricity generation is equal to the present value of all expenditure for its production (including construction and operation). Since the operation of boiler and steam turbine plant will not be changed, there will be no additional fuel consumption and reduction with PDHS retrofit. Furthermore, there will be not any change on the operating cost of the boiler and steam turbine plant due to PDHS retrofit. Therefore, the LCOE calculation is only affected by the following cost factors:

- CAPEX of PDHS retrofit
- Fixed and variable O&M Costs of PDHS system
- Saving on water cost (duction from the cost) with PDHS retrofit
- Incremental net power generation

The following is the assumed economic criteria used in the LCOE analysis (Table 17).

The results of the TEA are summarized in Table 18 and chart in Figure 35.

The PV and LCOE analysis results show that the potential cost saving due to reduced cooling tower water consumption is the main benefit of PDHS retrofit. The water cost saving can offset about 65-72% of the initial CAPEX on both PV and LCOE values. The LCOE for the option without condenser upgrade is relatively high due to small incremental net power output gains.

Table 17. Assumed economic criteria for LCOE analysis

Financing	
Escalation Rate	3.0%
Discount Rate	7.0%
Percent Financing Debt	50.0%
Plant Life, Years	25
Water Cost [(1)]	\$8.0/1000gal
Other Criteria	
Plant Capacity Factor	65%
Owner's Cost	10%

(1) Including waste water treatment and discharge fee.

The calculated LCOE for the option with condenser upgrade is within more reasonable range, especially the increased net power output will not produce any additional carbon deoxidate and other emissions. However, the economic viability of the retrofit depends on the specific market conditions, especially the electricity selling prices during the hot days. This market based economic analysis is beyond the scope of this study and should be conducted in the next phase of commercialization evaluation when the candidate project site for retrofit can be identified.

Table 18. Economic analysis for PDHS retrofit

Case Description	**w/o Condenser Upgrade**	**w/ Condenser Upgrade**
1. Present Value of Costs, MM\$		
CAPEX	104.1	115.6
Fixed O&M	13.7	15.2
Variable O&M	4.3	4.3
Water Cost Saving	-75.0	-75.4
Total PV of Cost	**47.1**	**59.6**
2. LCOE, \$/MWh		
CAPEX	980.5	290.3
Fixed O&M	129.3	38.3
Variable O&M	40.2	10.7
Water Cost Saving	-706.0	-189.5
Total LCOE	**443.9**	**149.8**

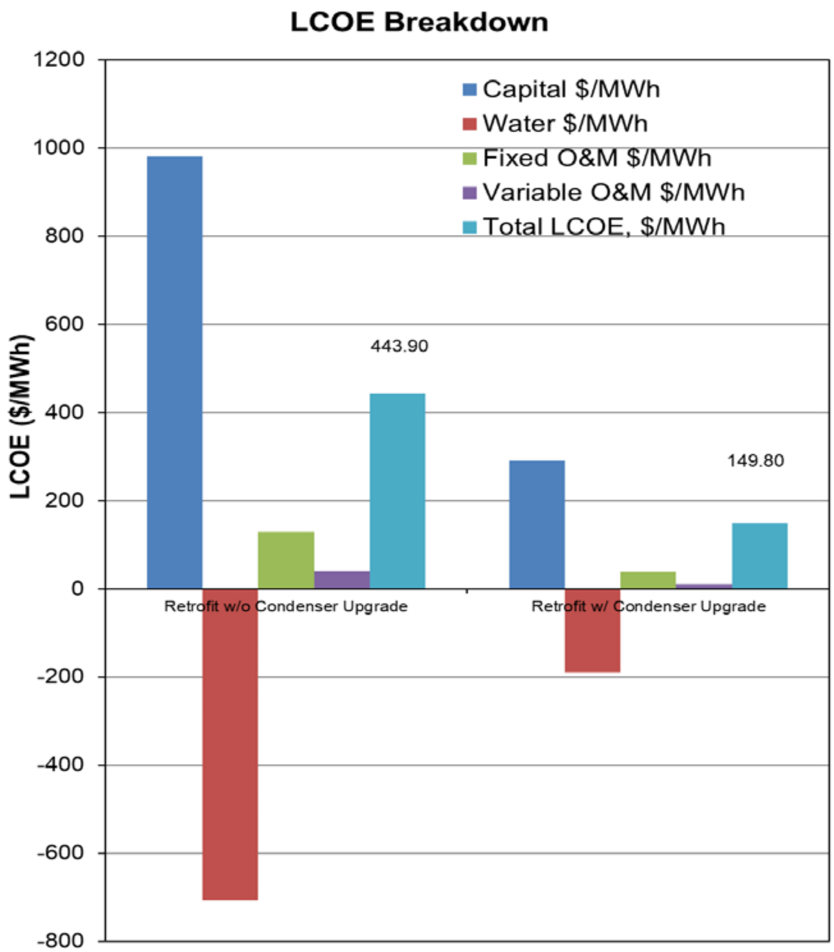


Figure 35. LCOE breakdown

Sensitivity analysis

Sensitivity studies around water cost, capacity factor and capital costs were performed to help identify scenarios where the PDHS retrofit would be economically favorable over the base case. All these three parameters significantly impact on the LCOE results and would be the main consideration factors for implementation of PDHS retrofit. The impact of these parameters are illustrated in Figure 36, Figure 37 and Figure 38.

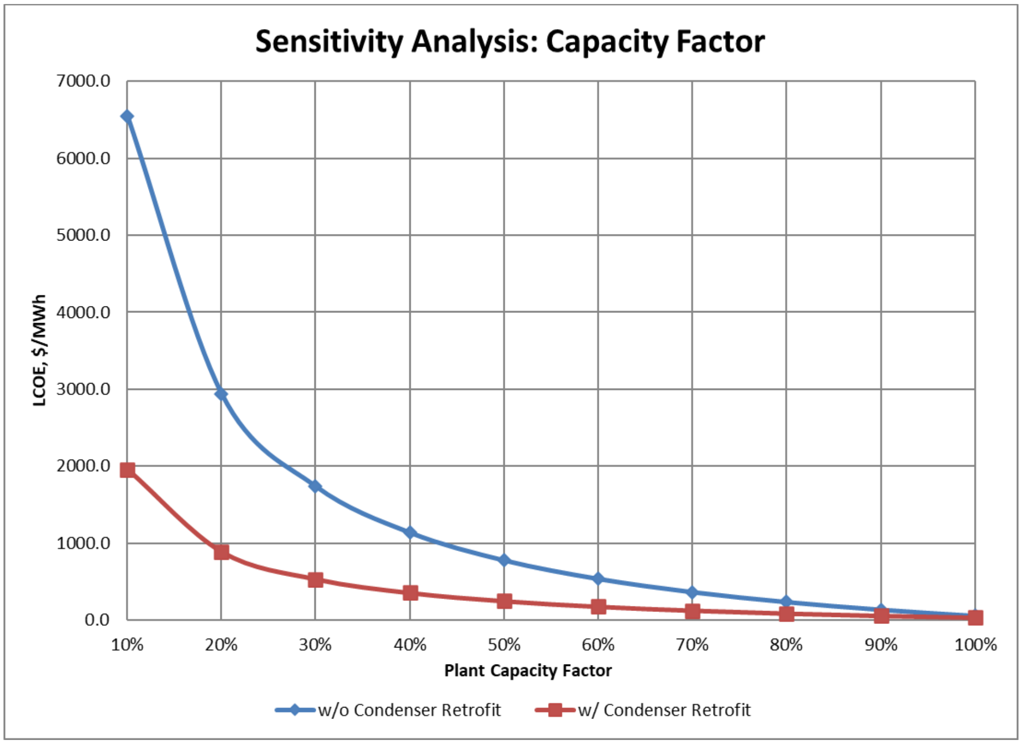


Figure 36. Sensitivity analysis – capital cost reduction

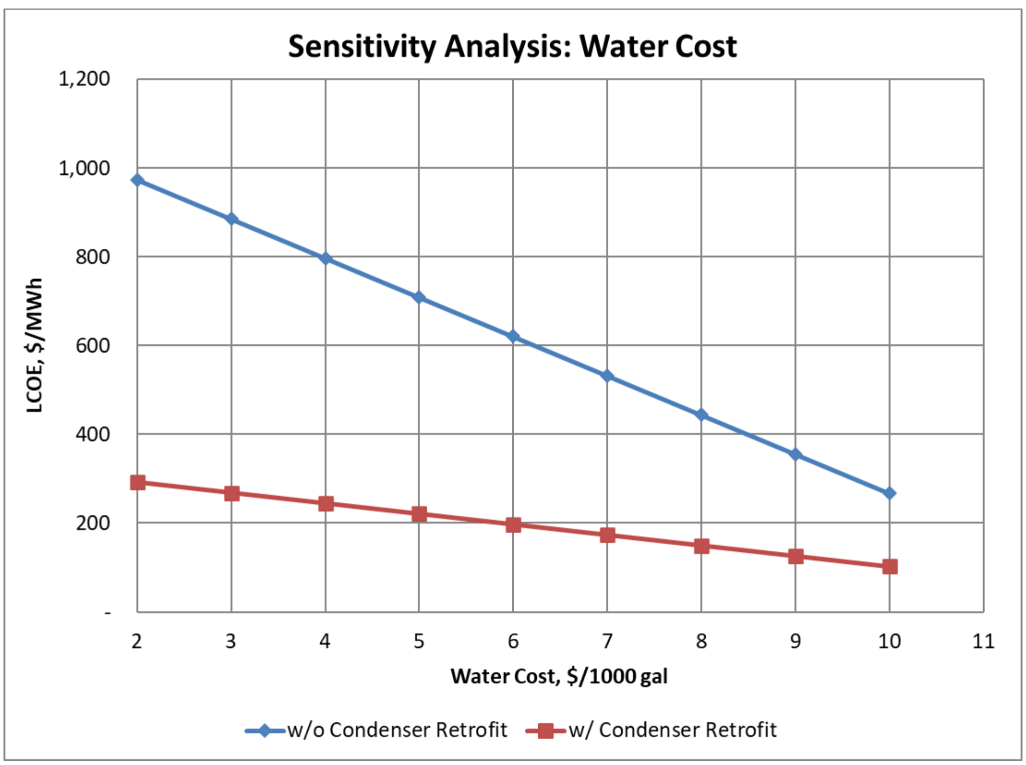


Figure 37. Sensitivity analysis – water cost

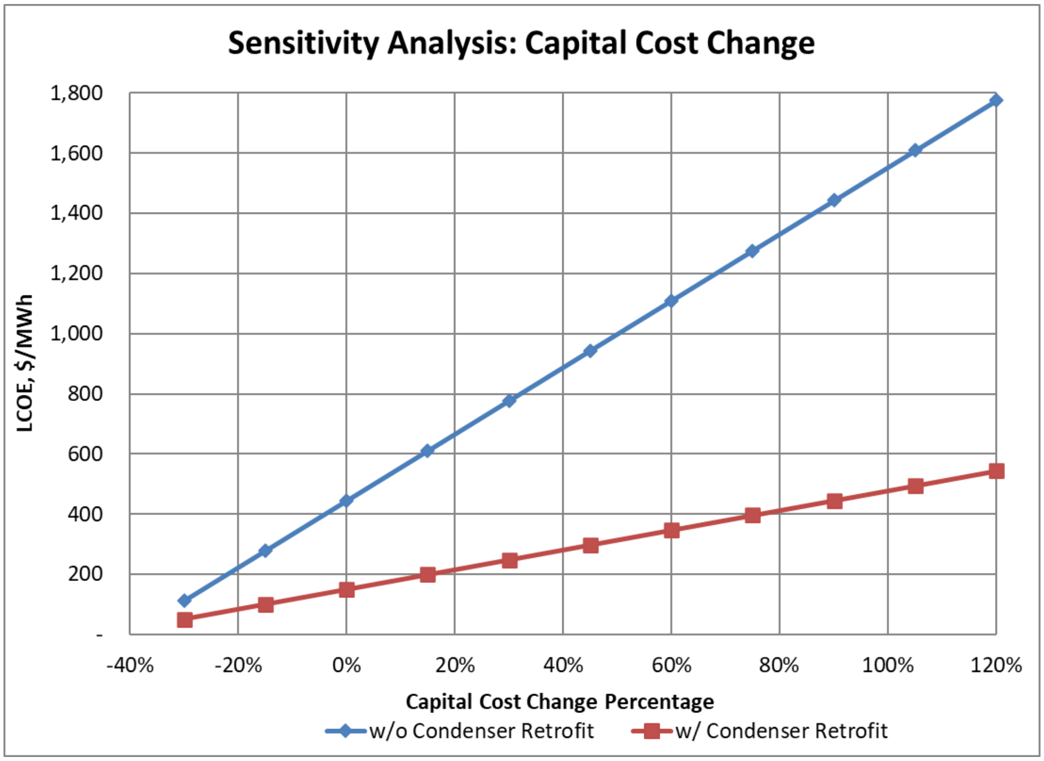


Figure 38. Sensitivity Analysis – Capital Cost Reduction

Technical Challenges and Market Barriers

When commercializing the SDPCT technology, the following technical challenges and market barriers should be considered if the SDPCT development continues:

- Optimization of the control system should be used to ensure the reliability and synchronization of the operation of blower and turboexpander in the PDHS.
- For a full-scale PDHS design, noise reduction should be considered
- Cold weather operation will require bypassing the PDHS.
- Appropriate turbo-expanders are not available off-the-shelf for low pressure systems, so the technology transfer plan should include special ordering from a qualified OEM.
- Increased cooling tower footprint by adding some PDHS components to a cooling tower.

Commercialization path

The following commercialization path has been proposed for next phase development of the SDPCT.

- GTI Energy to evaluate the spectrum of the industrial applications for the subject technology beyond CFPP
- Identify the commercialization team and key stakeholders
- Technology transfer activities and commercialization plan development

- PDHS design refinement and SDPCT integration engineering for follow-on field demonstration
- IP licensing and marketing support

Conclusions and recommendations of TEA

Based on the results of the TEA for the PDHS cooling tower retrofit concept, following conclusions could be drawn:

- Though the PDHS retrofit can lower the steam turbine back pressure notably, resulting in an increase of steam turbine gross output during the hot day operation. The overall plant net output may not increase at the same degree due to the additional parasitic power required for the PDHS, particularly the air blowers.
- Compared to the assumed existing plant baseline, plant net output increases by about 0.3% for the option without condenser upgrade and about 1% for the case with condenser upgrade at 95°F/35°C ambient.
- The most advantage of PDHS retrofit is drastically water reduction for the cooling tower. The cooling tower make-up water can be reduced by 31% at the evaluated ambient conditions. This not only results in direct operation cost saving, but also bring additional environmental benefit to reduce water draw and waste water disposal costs and wastewater discharge environmental issues.
- The calculated LCOE for the option with condenser upgrade is about $150/kWth, which is considered to be within a reasonable range for such retrofit. The actual economic viability of the retrofit needs to be assessed via a market based economic analysis when the candidate project site for retrofit can be identified.
- The air blower and expander efficiencies significantly impact on the overall plant net output and should be further investigated with the potential OEMs of these two pieces of equipment.

Chapter 7 Outreach to Utility Advisors

To improve the feasibility and position the SDPCT technology for the potential follow-on field demonstration, technology transfer, and commercialization, Jessica Shi Consulting, assisted by the project team, reached out to the cooling technology experts in several major utility companies. The valuable inputs from the following contacts in four companies (Table 19) are reported in this Chapter.

Table 19. Cooling Expert Contact Information.

Company	Expert Name	Title	Email Address	Phone No.
American Electric Power (AEP)	Matthew T. Usher	Director, New Technology Development & Policy Support	mtusher@aep.com	(614) 716-1000
Electric De France (EDF) -EDF R&D	Franck DAVID	Chercheur Senior	franck.david@edf.fr	(+33) 1 30 87 78 20
Electric Power Research Institute (EPRI)	Andrew Howell	Technical Executive	ahowell@epri.com	(980) 215-1805
Southern Company Services (SO)	Rebecca D. Osteen	Research Engineer	RDOSTEEN@SOUTHERNCO.COM	(205) 257-5951

The comments and advice by the cooling experts are categorized and documented in the tables below.

Table 20. Cooling experts comments about their interest in this technology.

Company	Is this technology of interest to you and/or to your company?	Would this technology be a possible option for your company if it is commercialized in the future?
AEP	AEP remains engaged with EPRI and other industry partners to investigate technologies that improve efficiency, operational flexibility and reduce water usage. This particular technology does not hit the mark for us because we do not believe that the benefit, justifies the high cost of implementation.	Likely not.

EDF R&D	On principal, the technology can be of interest and is well explained in the documents. The experiments and demo give a first good assessment of the potential. Beyond the potential, I can see some weakness: • The technology will be mostly applicable when the ambient wet bulb is above 75 F or 22 °C. Those temperature are not very frequent. • Some technology developments need to be confirmed (i.e., turbo expender). • Additional systems like IHX is required which will probably need space and additional foot print. • A need to imagine a double air inlet for the two cases: cold / hot conditions (note that we are usually using natural draft cooling tower for our power plants)	Yes, in the future if the ambient conditions (temperature, water scarcity) and / or regulation require a strong effort.
EPRI	It certainly could be if it works as described and the large-scale configuration is practical and not excessively costly.	Power generating companies are reluctant to adopt new technology but can become interested if the science and pilot demonstrations are convincing.
SO	This technology is not of great interest at this time.	Probably not as it does not make sense as a retrofit option and the cost would most likely not make sense for new projects.

Table 21. Cooling experts comments about practicality of this technology.

Company	What do you think about the practicality of this technology for the retrofit and new project applications?	What are your concerns, the risks we should address, and mitigation strategies you would recommend?
AEP	It seems to be technically feasible and plausible for its intended application.	No comments
EDF R&D	The technology seems a little bit far from an adequate TRL to be accepted and need some steps like demo coupled with real cooling system (first on medium size cooling system).	Range of applicability; component availabilities; foot print and economy

EPRI	Retrofit on a limited basis (sort of semi-pilot) seems best to me initially, as it could confirm the effectiveness of the technology and lead to expansion. As a focus area, retrofits may be limited economically if remaining life of a plant is short.	It seems to me the biggest risk is the cost of developing and constructing suitable equipment not otherwise available, with no guarantee that it will prove marketable. Obviously, this would initiate on a limited basis and increase in size and scope of the initial efforts were successful.
SO	This would not make sense for a retrofit application. There may be some application is a cooling tower were undersized, but we design our cooling systems to handle the ambient conditions they operate under. The cost of retrofitting would not make sense for us. There might be some application for new projects if you could reduce the size of your condenser or circ water pump to offset the cost. In regions where water scarcity is an issue or there is a high price of water, there could be benefits if a meaningful amount of water were recovered. There would need to be a cost benefit analysis for those scenarios to see if it makes financial sense.	• There could be issues with reaching acid dew point when recovering heat from flue gas. • How does this system ensure the cold, dehumidified air is distributed evenly across the tower? It seems this will be a challenge on full-scale power plant cooling towers. • Slide 13 states the “cold weather operation will require bypassing the PDHS.” I’m concerned that the equipment will impede air flow to the tower when the system is not in service. • The footprint of the system is also a concern. Many of our units are space constrained around the cooling towers, so depending on how large the system footprint is, this could be an issue.

Table 22. Cooling experts comments about economics of this technology

Company	Is this technology economically viable?	Which is more important on the economic consideration: power generation improvement or saving on cooling tower makeup water?	What cost targets should we strive for?

AEP	We do not believe that it is. The $/kW and LCOE are just too high for the incremental efficiency benefit. No regulated utility commission would support us implementing this technology and passing that high cost onto our customers.	Neither matters when the costs are this high.	Less than $1000/kW
EDF R&D	The numbers need to be checked internally according to our conditions of application (We do need to make our internal assessments which can require some working effort.).	Both are important but power improvement (or no power degradation) is the first concern.	I don't have the information.
EPRI	I'm not sure I can sort this out definitively.	This is situation-specific but probably more plants would find the efficiency improvement more important. 31% makeup water reduction could be very significant in some situations, possibly overly optimistic but I don't know the details used to arrive at that figure.	Return on investment is key, particularly with a time frame in the 3–5-year range.
SO	No, it looks like the saving water is a large part of the cost analysis. The lower the cost of water, the less this makes economic sense. Most of our power plants do not have a cost for water, so there would not be financial savings there.	For Southern Company, improved power generation and heat rate is more important.	

Table 23. Cooling experts comments about demonstration of this technology

Company	Would your company be interested in hosting or recommend a host site for a possible DOE funded demonstration project?
AEP	No thank you. Not at this time.
EDF R&D	We would need to make a deeper evaluation of the potential in order to convince my colleagues from engineering division for a possible demo.
EPRI	Possibly recommending if the technology and next steps fit together.
SO	No.

Table 24. Cooling experts comments about any additional comments/advice

Company	Any additional comments/advice
AEP	
EDF R&D	As suggested above, there is a first step (at R&D level) for evaluation of the process. The time I spent on the subject was too short. We need at first reconsider the thermodynamics combining our input data (usual ambient condition, power plant) and some information you mentioned derived from the demo – It will probably need several days /weeks next year.
EPRI	I would think BAC would be in position to identify customers who might do a pilot level test. I would be interested in having a Webex meeting to gain a better understanding of the technical basis for the technology.
SO	

The inputs above are included and discussed in other parts of this report as well. The team is very appreciative to the above valuable input and guidance from our advisors and will strive to improve the techno-economic feasibility.

Conclusions

The enhanced cooling tower technology for power plant efficiency increase and operating flexibility has been developed and demonstrated. A pressure dehumidifying system (PDHS) installed at the cooling tower inlet is used to cool and dehumidify the inlet air. The PDHS prototype has been tested at GTI industrial laboratory. Various process temperatures and flow rates were simulated in the pilot-scale PDHS test. Three selected representative cities of the U.S. major climate zones for coal power plants and the range of the wet bulb temperature were used in the experiments. Experiments have confirmed that the cooled air is dehumidified when the air temperature falls below the local dew point. The most advantage of PDHS retrofit is drastically water reduction for the cooling tower. Estimated water savings in the cooling tower, estimated from the results of experiments, amounted to more than 20%. The reduction in make-up water results in water cost savings of 47.1-59.6 $/MWh. Achieved water cooling 2.5°F - 4.2°F below ambient dew point resulting in a net increased in plant efficiency of 0.36-1.06%. The calculated LCOE for the option with condenser upgrade is about $150/kWth, which is considered to be within a reasonable range for such retrofit. The PDHS retrofit can lower the steam turbine back pressure notably, resulting in an increase of steam turbine gross output during the hot day operation. The potential outstanding impact of the cost effective, durable sub-dew point cooling tower technology is that it has a cross-cutting nature, is retrofittable and it can benefit not only coal-fired power generation systems but also other industries and applications. The overall world-wide impact of this technology with dramatically reduced cooled water temperature would not only significantly improve power production efficiency, or reduce the power plant CO_2 emission, but also reduce the power consumption for cooling drastically. Furthermore, the attractive potential of over 20% of cooling tower water consumption reduction, would profoundly help conserve the limited and precious fresh water resource.

References

[1] "DOE-FOA-0001816".

[2] Y.Chudnovsky, A.Kozlov, P.Glanville, "Development of an advanced dew point cooling fill concept for power plants," *EPRI Technical Report*, 2016.

List of Acronyms

Acronym	Description
SDPCT	Sub-Dew Point Cooling Tower
PDHS	Pressure Dehumidifying System
TEA	Techno-economic assessment
HX	Heat Exchanger
HMX	Heat and Mass Exchanger
COP	Coefficient of Performance
TTD	Terminal Temperature Difference
CAPEX	Capital Expenditure
LCOE	Levelized Cost of Electricity
OPEX	Operational Expenditure
PV	Present Value
O&M	Operation & Maintenance
P&ID	Piping and Instrumentation Diagram
OEM	Original Equipment Manufacturer