ASTFE
American Society
of Thermal and Fluids Engineers

8th Thermal and Fluids Engineering Conference (TFEC)
March, 2023
Partially Online Virtual and at University of Maryland, MD

TFEC-2023

THERMAL INVESTIGATION OF A CHILLER - COOLING TOWER CONFIGURATION WITH PRECOOLING PROVIDED BY A BRAYTON-CYCLE-BASED-SYSTEM

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ABSTRACT

A system-level study has been conducted of a cooling system consisting of a chiller, a cooling tower and a Brayton-cycle-based-system to precool the air entering the cooling tower. The novelty of this system is the air precooling system that is expected to enhance the cooling load of the chiller. The simulations consist of thermodynamic relations for compression, expansion and phase change processes, and heat and mass transfer balances in the heat exchange components. Simulation results are compared against measured data collected in a laboratory set up consisting of a blower, a water-cooled heat exchanger and an expander, and the simulation results agree well with the measured data validating the various elements of the model.
Simulations are performed for a warm summer day with 35 degrees C inlet air and 47% relative humidity. Three different configurations are considered: (a) a closed loop water flow system (Case A) where the water flow between the cooling tower, chiller, and the air-water heat exchangers form a closed loop system (b) open loop water flow system (Case B) where the water flow through the air to water heat exchangers is provided by municipal water and the cooling tower and chiller operate in closed loop (c) same as the (b) configuration but with an air to air heat exchanger placed downstream of the blower to precool the air entering the air-water heat exchanger.

Results are presented utilizing a few empirical models for the cooling tower that assume a certain performance across the cooling tower. These results demonstrate that Cases B and C provide improvements in the chiller load and COP of the system compared to case A, but no substantial differences between Case B and Case C is observed.

KEY WORDS: Heat Transfer, Cooling Tower, Chiller, Precooling, Reversed Brayton Cycle

1. INTRODUCTION

Cooling systems such as cooling towers are required and deployed in many practical applications. A very common example is power plant cooling where cooling towers are used to cool the circulating water in the condenser. Another potential application could include cooling systems where a non-refrigerant based chiller is used to provide the cooling load for a building system. In these cases, chillers may be coupled with external cooling towers that cool down the cooling water (or working medium) in the chiller. In both cases above, it is beneficial for the cooling tower to provide the lowest discharge temperature of the cooling water since this effects the steam condensation temperature (in a power plant condenser) or the cooling load in the case

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of a chiller. A cooling tower is generally an air/water direct contact heat exchanger whose primary task is to transfer the heat from the water to the counterflow air stream. A small fraction of water evaporation and makeup water is needed to maintain a constant water flow rate in the chiller [1, 2]. The major drawback of wet cooling towers is that they are limited by the wet bulb temperature of the air entering the tower. Therefore there is a strong deterioration in cooling tower performances under summer conditions when hot air is used for cooling the water [3]. In order to solve such issues, several air-precooling approaches have been studied such as staged evaporative pre-cooling, energy recovery from exhaust air [4], and mechanical vapor compression [5].

This study explores a new system of air precooling. Instead of using ambient air at the inlet of the cooling tower as is usually done, the air is precooled and dehumidified by a reverse Brayton-cycle-based-system before entering the tower. This air-precooling system composed of a compressor, a heat and mass exchanger and an expander, is expected to enhance the cooling load of the chiller at the expense of some work done by the reverse Brayton cycle system. For power plant applications, the enhanced cooling tower performance is expected to lower the steam condensation temperature and therefore contribute to increased power-plant efficiency. Under some limited scenarios, this air precooling system combined with the cooling tower can achieve enough cooling of the air to enable a portion of it to be used to provide building cooling.

This paper utilizes a systems approach and thermodynamic relations and energy balances to evaluate the system performance. The system model results are evaluated against laboratory data for the Brayton-Cycle system (without the cooling tower) in order to validate the system model. The overall model is then utilized to explore different model configurations and the effect of parametric variations to explore system performance.

2. SYSTEM MODELING

A schematic of the entire system is shown in Figure 1 with the various components shown. As noted earlier, the primary components of the system are: a compressor or a blower (C), a heat/mass transfer exchanger (HMX), a cooling tower (CT), a turboexpander/turbine (EXP), and a chiller (C). In a closed-loop configuration described later, an additional heat exchanger (HX) is employed between the HMX and the CT. In this section, a description of the the models used for the individual components in the system are given. Thermodynamic relations are used for the compression and expansion processes, whereas standard heat and mass transfer balance equations are utilized in the heat and mass exchanger. The Cooling Tower (CT) model is based on empirical models that assume a certain performance across the CT.

A lab-scale facility consisting of the blower, the HMX and the turboexpander has been assembled at the Gas Technology Institute (GTI). In this setup, the cooling water to the HMX is provided in an open-loop from the city-water supply. Data for a range of flow rates and test conditions have been collected to demonstrate the viability of reverse-Brayton cycle system concept. This data also serves to validate the system model constructed as described below.

2.1 Compressor/Blower Model

The compression ratio CR of the compressor or blower is defined as the ratio between the outlet and inlet pressure of air:

$$CR = \frac{P_{ao}}{P_{ai}} \tag{1}$$

For a defined CR, the exit air pressure from the compressor can be calculated using the compression ratio.

$$P_{AB,ao} = CR * P_{AB,ai} \tag{2}$$

According to the Gas Processors Suppliers Association (GPSA) Engineering Databook, the temperature of air at the compressor outlet is given by the following thermodynamic relation:

$$T_{AB,ao,db} = T_{AB,ai,db} + \Delta T_{AB,is} \tag{3}$$

$$\Delta T_{AB,a,is} = T_{AB,ai,db} * \frac{CR^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{C,is}} \tag{4}$$

Once the pressure and the dry bulb temperature of the air coming out of the compressor have been calculated, the psychometric relations are used to determine all the remaining properties of the outlet air.

The work done by the compressor is defined as the air mass flow rate times the variation in enthalpy between the inlet and the outlet of the compressor.

$$\dot{W}_C = \dot{m}_a(e_{AB,ao} - e_{AB,ai}) \tag{5}$$

2.2 Heat and Mass Exchanger Model

The HMX calculations are done essentially following the NTU model. This model requires information about the surface geometry and area, and the overall heat transfer coefficient. The Hydronic Duct Heating and Cooling Coil - the HMX used here- is a cross flow finned tube heat exchanger. It basically consists of air and water streams that exchange heat with one another. In the case considered the air is the hot fluid and the water is the cold fluid.

On the test setup based at GTI facilities the overall heat exchanger consists of a series of four individual HMX units connected in series. For the calculations, the system of 4 HMX units in series is divided into 4 sections - each section corresponding to 1 unit. The output of the unit i therefore corresponds to the input of the unit $i + 1$. The inlet air conditions at the first unit corresponds to the outlet air conditions from the compressor.

The manufacturer specifications for the HMX units, as used in the GTI test set up, have been used here to define the different geometry parameters needed in the model built. These are used for calculating the inner and outer surface areas for the heat exchanger surfaces.

For the heat transfer coefficient on the air side - h_a - choice has been made to use the Colburn j-factor analogy. The Colburn j-factor is defined as follows:

$$j = St.Pr^{2/3} \tag{6}$$

Where Pr is the Prandtl Number and St the Stanton Number.

For the HMX setup used the following correlation for the Colburn j-factor may be used:

$$j = 0.163 Re^{-0.369} \left(\frac{S_T}{S_L}\right)^{0.106} \left(\frac{F_p}{D_c}\right)^{0.0138} \left(\frac{S_T}{D_c}\right)^{0.13} \tag{7}$$

Once the Colburn j-factor has been computed, the Stanton number and the convective heat tranfert coefficients are calculated using the following formulas:

$$St = j.Pr^{-2/3} \tag{8}$$

$$h_a = St.G.c_{p,a} \tag{9}$$

where:

$$G = \frac{V_{max,a}}{V_{ai}} \tag{10}$$

and

$$V_{max,a} = \frac{\bar{V}_a.S_T}{min((2(S_d - d_o)),\ (S_T - d_o))} \tag{11}$$

with:

$$\bar{V} = \frac{\dot{m}_a.V_{ai}}{H_{HMX}.L_{HMX}} \tag{12}$$

and

$$S_d = \sqrt{S_L^2 + \left(\frac{S_T}{2}\right)^2} \tag{13}$$

To get the convective heat transfer coefficient on the water side - h_w - correlations for internal flow in a circular pipe are used. These correlations are based on the value of the Reynolds number of the internal flow.The Reynolds number is therefore first calculated and a deterministic choice for the correlation to be used is made based on the value of the aforesaid Reynolds number.

$$Re_D = \frac{4\dot{m}}{\pi D \mu} \tag{14}$$

Choice has therefore been made to work with the Gnielinski correlation as it has the widest range of validity:

$$Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}\left(Pr^{2/3} - 1\right)} \tag{15}$$

where:

$$f = \frac{1}{(0.79ln(Re_D) - 1.64)^2} \tag{16}$$

Once the Nusselt Number has been calculated using the appropriate correlation the convective heat transfer coefficient can be obtained.

$$h_w = \frac{Nu_D.k_w}{D} \tag{17}$$

The diameter D in the above formula is equal to the inner diameter of the tubes and all fluid properties used are those of the water.

After having obtained the different heat transfer coefficients on both air and water side the calculations for each section can be started with the NTU method.

For each section i the following steps need to be taken:

- Calculation of the NTU on the air side NTU_a (Eq. 18)
- Calculation of the R coefficient (Eq. 19)
- Calculation of the surface temperature of the pipe T_s (Eq. 20)

The air side Number of Unit Transfer - NTU_a - is obtained using the following expression:

$$NTU_a = \frac{h_a.A_{ex,o,HMX}}{\dot{m}_a c_{p,a}} \tag{18}$$

The R coefficient is the fraction ratio of the air enthalpy difference on the air side over the temperature difference on the water side and can be written as follows:

$$R = \left(\frac{c_{pa}.h_w}{h_a}\right).\left(\frac{A_{ex,i,t}}{A_{ex,o,HMX}}\right).\left[\frac{1+\frac{NTU_a}{2}}{1+\frac{NTU_w}{2}}\right] \tag{19}$$

Once R is known the temperature at the surface of the pipe T_s can be computed using the following expression:

$$T_s = \frac{-(R+1.4).\sqrt{(R+1.4)^2+0.184.(e_{ai}+R.T_{wi}-10.76)}}{0.092} \tag{20}$$

This temperature is then compared to the air dew point temperature. If this temperature T_s is below the dew point temperature of the air in the i section - i.e. if $T_s \leq T_{a,dp,i}$ - condensation happens in the HMX. If T_s is above the dew point temperature there is no condensation in the section.

If condensation happens in the i section of the HMX setup, the HMX calculations are initiated by getting the heat load of the section i:

$$\dot{Q}_{HMX,i} = \dot{m}_w c_{pw}(T_{s,i} - T_{w,i})(1 - e^{-NTU_w}) \tag{21}$$

Once the heat load of the section is known, the outlet air and water temperatures can be found using:

$$T_{w,i+1} = T_{w,i} + \frac{\dot{Q}_{HMX,i}}{\dot{m}_w c_{p,w}} \tag{22}$$

$$e_{a,i+1} = e_{a,i} - \frac{\dot{Q}_{HMX,i}}{\dot{m}_a} \tag{23}$$

$$T_{db,a,i+1} = \left[\frac{1-\frac{NTU_a}{2}}{1+\frac{NTU_a}{2}}\right].T_{db,a,i} + \left[\frac{NTU_a}{1-\frac{NTU_a}{2}}\right].T_{s,i} \tag{24}$$

The absolute humidity can be calculated as:

$$w_{a,i+1} = \frac{e_{a,i+1} - c_{p,a}T_{db,a,i+1}}{h_{fg,w} + 1.8e_{a,i+1}T_{db,a,i+1}} \tag{25}$$

The remaining air properties such as relative humidity, pressure, etc. are determined using the psychometric code.

When there is no condensation involved the following steps the temperature of the air leaving the section can be found using the simple and usual relation:

$$T_{db,a,i+1} = T_{db,a,i} - \frac{\dot{Q}_{HMX,i}}{\dot{m}_a c_{p,a}} \tag{26}$$

Since there is no condensation in the section the absolute humidity of the air doesn't change:

$$w_{a,i+1} = w_{a,i} \tag{27}$$

The remaining air properties such as relative humidity, pressure, etc. are determined using the psychometric charts.

Once every section has been solved the properties of the air coming out of the HMX system can be calculated.

The total load of the HMX Setup is obtained by summation of the load of each section.

$$\dot{Q}_{HMX} = \sum_{i=1}^{nbSec} \dot{Q}_{HMX,i} \tag{28}$$

If condensation happens in one of the section, the amount of condensed water is calculated as follows:

$$\dot{m}_{Condensate\,HMX} = \dot{m}_a.(w_{ai} - w_{ao}) \tag{29}$$

The heat transfer from air and water streams must balance, so the HMX must satisfy the following condition on heat load:

$$\dot{Q}_{HMX,a} = \dot{Q}_{HMX,w} \tag{30}$$

Where:

- $\dot{Q}_{HMX,a}$ is the heat load lost by the air flow while cooling
- $\dot{Q}_{HMX,w}$ is the heat load gained by the water flow while heating

Due to a non insulation of the HMX, an additional heat loss has to be considered. This extra heat loss has been calculated as the combination of heat loss by conduction, radiation and convection due some recirculation of the air around the HMX. These estimated heat losses are calculated using a 1-D heat conduction loss model (for conduction losses), and assumed emissivity of the surface for radiation losses and an assumed heat transfer coefficient for recirculating air around the heat exchanger containment system. The appropriateness of these loss calculations are verified during the validation phase with the experiments.

2.3 Heat Exchanger

For the heat exchanger (HX), the HMX model was used without considering any condensation in it. As noted earlier, in the closed loop configuration a HX is placed between the exit of the HMX and the inlet to the CT (shown in Fig. 1)

2.4 Expander

The expansion - taken to be adiabatic - leads to a decrease in the fluid pressure and temperature. The Expander Ratio, ER defines the amount of fluid expansion achieved in the expander. This ratio ER corresponds to the ratio between the outlet and inlet pressure of fluid.

$$ER = \frac{P_{ai}}{P_{ao}} \tag{31}$$

As the temperature drops during the adiabatic expansion condensation may take place in the expander. For condensation to happen the temperature of the air needs to go below the dew point. The outlet pressure from the expander corresponds to the pressure of the ambient air.

$$P_{ao,EXP} = P_{amb} = 101.321\,kPa$$

The isentropic efficiency equations are used to get the temperature of the air coming out of the expander.

$$T_{EXP,ao,db} = T_{EXP,ai,db} + \Delta T_{EXP,is} \tag{32}$$

$$\Delta T_{EXP,is} = \eta_{EXP,is}.T_{EXP,ai,db}\left(\frac{1}{ER^{\frac{\gamma-1}{\gamma}}} - 1\right) \tag{33}$$

The power output from the expander can be accessed using the following expression:

$$\dot{W}_{EXP} = \dot{m}_a(e_{ai} - e_{ao}) \tag{34}$$

If condensation takes place in the expander the rate of condensation is calculated by:

$$\dot{m}_{Condensate\,EXP} = \dot{m}_a.(w_{ai} - w_{ao}) \tag{35}$$

2.5 Cooling Tower

In this study an empirical model for the cooling tower that assume a certain performance across the cooling tower have been used. That empirical model lies on a given approach temperature for the cooling tower:

$$T_{wo,CT} - T_{ai,wb} = \Delta T \tag{36}$$

The air outlet temperature and water inlet temperature to the cooling tower are calculated using the following formulas:

$$\dot{Q}_{CT} = \dot{m}_{w,CT} * c_w * (T_{wi,CT} - T_{wo,CT}) \tag{37}$$

$$\dot{Q}_{CT} = \dot{m}_a * (e_{ao,CT} - e_{ai,CT}) \tag{38}$$

2.6 Chiller

In the chiller the water absorbs the heat from the working fluid that provides the cooling air or load to the building. The water temperature at the outlet of the chiller can be obtained using the following expression:

$$T_{wo,Chiller} = \frac{T_{wi,CT} * \dot{m}_{w,CT} - T_{wo,WC} * \dot{m}_{HMX}}{\dot{m}_{Chiller}} \tag{39}$$

The inlet water temperature is known from the cooling tower model.

The cooling load of the chiller can then be calculated as follows:

$$\dot{Q}_{Chiller} = \dot{m}_{w,Chiller} * c_w * (T_{wo,Chiller} - T_{wi,Chiller}) \tag{40}$$

3. CONFIGURATIONS STUDIED

In this section, the various configurations studied are described. For all configurations that include a cooling tower the following the constraint was set:

$$T_{db,ai,CT} < T_{wi,CT} \tag{41}$$

3.1 Baseline PDHS

The baseline case corresponds to the Precooling and Dehumidifying System (PDHS) alone. This PDHS is made of a compressor, a HMX and an expander using external water supply for the HMX. An experimental setup of this system has been built as noted earlier and used to validate the numerical models.

The experiments at GTI were conducted for different inlet air conditions, air and water mass flow rates and different pressure ratio. To vary the inlet air conditions, the team at GTI used a heater and humidifier that are not being accounted for in the numerical model of the test setup.

The air and water temperature was being measured at the inlet and outlet of each component using thermocouples.

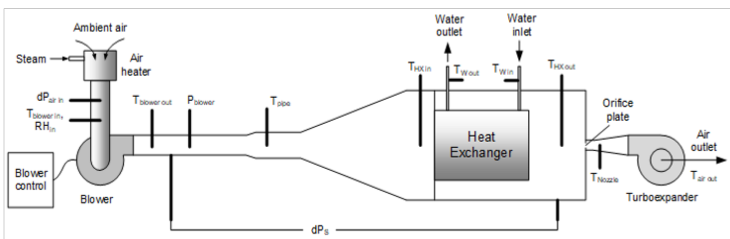

Fig. 1 *Schematic of the Test Setup build at GTI Energy for the PDHS. Note that the Heat and Mass Exchanger noted in the setup is the HMX referenced in the text to allow for condensation*

3.2 Baseline PDHS - Closed Loop Water (PDHS-CLW)

This configuration corresponds to the PDHS system coupled to a cooling tower and a chiller. The PDHS is providing air to the cooling tower at a lower temperature than the ambient air. The outlet water to the CT is here also linked to the HMX and is used as water supply to the HMX.

Since the water coming out of the CT is used to cool the HMX and its temperature is a function of the expander outlet air wet bulb temperature, an iteration process is used to solve as follows:

- Calculate the exit air conditions to the air blower knowing the air inlet conditions and blower efficiency and pressure ratio
- Calculate the air inlet conditions to the HMX knowing the air outlet conditions to the air blower and taking into account heat and pressure loss in the pipe.
- Solving for the HMX and expander knowing the air inlet conditions to the HMX. To do so, a first water inlet temperature is guessed and then it is being updated at each iteration until convergence using the following relation:
$$T_{wi,HMX} = T_{wo,CT} = T_{wb,ao,EXP} + \Delta T \tag{42}$$
- Solving for the CT and Chiller knowing the expander outlet air and CT outlet water. A first load is guessed and updated at each iteration until the constraint is met on the air and water temperature is met.

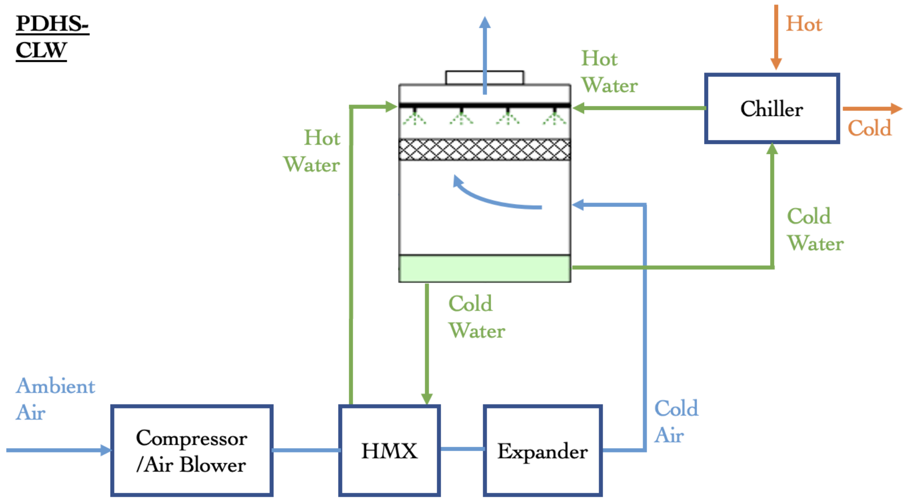

Fig. 2 *Schematic of the Precooling and Dehumidifying System integrated in the Cooling-Tower/Chiller System with a Closed Loop Water. An additional HX (water cooler) not shown in the figure is incrporated in the closed loop system between the HMX and the cooling tower and is included in the calculations*

An additional water cooler (also called HX) has been added here between the HMX and the CT to cool down the water coming out of the HMX. This water cooler or HX is being bypassed when the ambient air reaches a higher temperature than the water.

3.3 Baseline PDHS - Open Loop Water (PDHS-OLW)

In this configuration the PDHS is linked to the CT to provide air. However, an open loop is now considered for the HMX water. The CT does not provide water to the HMX as in the previous section. In this configuration, running water at constant temperature is used. The system has been tested with the following range of water T in order to cover all seasons:

- Water Inlet T HMX (°C) = [15, 18, 20, 23, 25, 28 , 30]

This time, since water is being provided at a constant temperature to the HMX, there is no need to loop to solve for the PDHS. The only loop needed is the one to solve for the coupled system CT-Chiller.
The following steps are followed to solve:

- Solve for the PDHS components after components knowing the inlet air conditions and all the component models.
- In that case, the water goes into the HMX at a constant temperature and is then dumped.
- Once the air exiting the expander is known, the CT and Chiller can be solved using the same process as before.

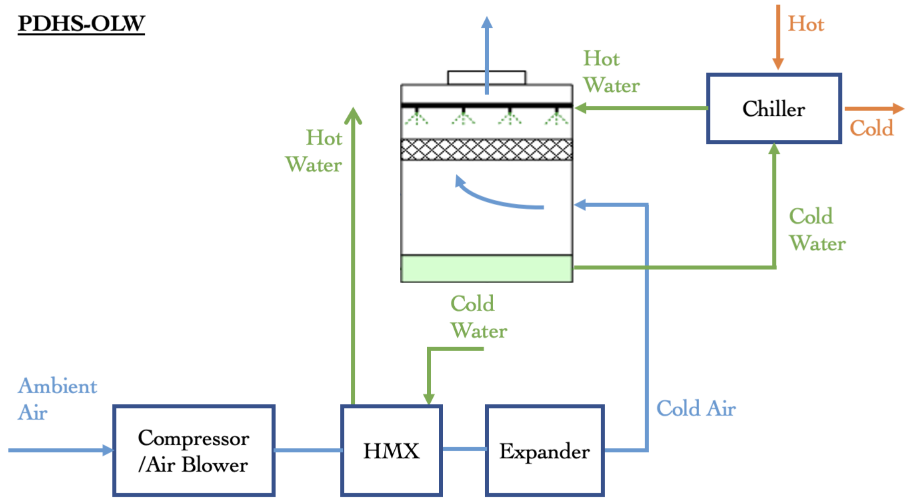

Fig. 3 *Schematic of the Precooling and Dehumidifying System integrated in the Cooling-Tower/Chiller System with an Open Loop Water*

4. RESULTS AND DISCUSSION

This section presents the results for the three configurations considered.

4.1 Baseline PDHS

For the baseline PDHS system a test setup has been built to conduct experiments and validate the results obtained numerically.

The experimental setup has been used to validate the numerical models. In the table below are presented the results of the numerical HMX model to compare with the measurements used in the GTI setup. These results corresponds to three different air inlet conditions.

Table 1 Baseline

Inlet Condition	EXP1	NUM1	EXP2	NUM2	EXP3	NUM3
Air Dry Bulb Temp., °C	34.67	34.67	29.78	29.78	21.17	21.17
Air Relative Humidity, %	45.20	45.20	70.30	70.30	17.9	17.9
Air Wet Bulb Temp., °C	24.90	24.90	25.38	25.38	9.79	9.79
Air Dew Point Temp., °C	21.06	21.06	23.78	23.78	-4.16	-4.16
Air Flow Rate, m^3/s	0.249	0.249	0.155	0.155	0.196	0.196
Air Pressure, kPa	101.3	101.3	101.3	101.3	101.3	101.3
Compressor Outlet Conditions						
Air Dry Bulb Temp., °C	53.40	53.50	47.50	47.50	82.40	82.40
Air Pressure, kPa	111.9	111.9	111.1	111.1	123.9	123.9
Pressure Ratio	1.104	1.104	1.097	1.097	1.224	1.224
Efficiency, %	47.12	47.12	45.95	45.95	28.80	28.80
HMX Air Conditions						
Air In Dry Bulb Temp., °C	48.78	51.53	42.83	44.89	73.67	78.56
Air Out Dry Bulb Temp., °C	22.56	23.78	24.56	25.14	22.89	24.39
Air Heat Flow, kW	7.590	7.919	3.671	3.550	12.009	12.789

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Table 1 – *Continued from previous page*

HMX Water Conditions						
Water Inlet Temp., °C	9.22	9.22	10.17	10.17	6.56	6.56
Water Outlet Temp., °C	24.39	24.41	29.11	26.22	25.94	26.62
Water Mass Flow Rate, kg/s	0.120	0.120	0.050	0.050	0.145	0.145
Water Heat Flow, kW	7.403	7.403	3.999	3.320	11.766	12.040

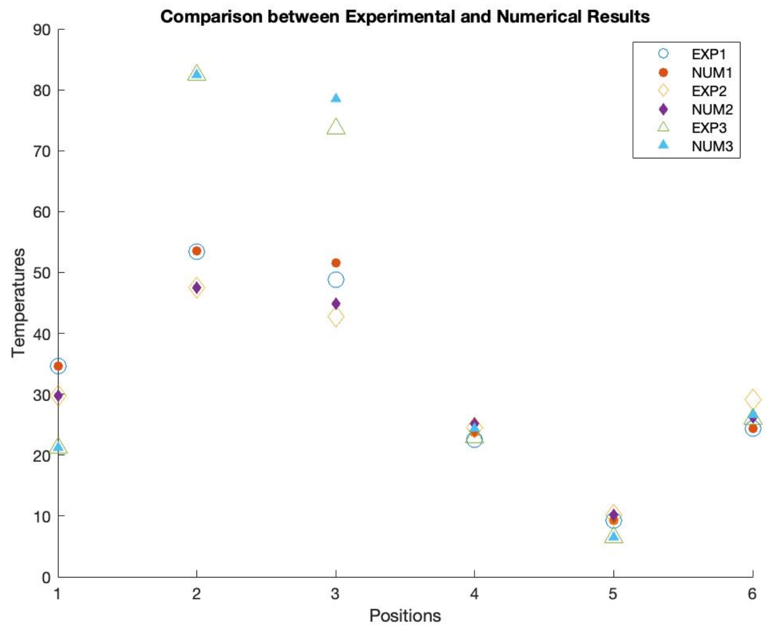

Fig. 4 *Comparison between Experimental and Numerical Results for the HMX*

Results show that the model created produces predictions that agree well with the experimental results obtained. The computation and experiments are also shown graphically in Figure 3. The small difference in the results shown is likely due to the HMX body not being insulated and some heat losses from its outer surfaces. While these losses were estimated in the calculations, the estimation is likely to differ slightly from the experimental situation.

4.2 Baseline PDHS - Closed Loop Water (PDHS-CLW)

Focus is now on utilizing the validated model in evaluating the closed loop PDHS-CT-Chiller system. Since the objective here was to achieve more than 1 ton of cooling (more than 3.52 kW of cooling as needed for a small building) in the chiller, different air and water flow rates and different CT ΔT were tested.

Ambient summer condition were used as inlet conditions to the compressor:

- $T_{db} = 35\ °C$
- $\phi = 47\%$
- $P = 101.3\ kPa$

In this configuration, a compressor and expander efficiency of 90 % was assumed.

For this configuration, for all the CT model tested, only one case has achieved the set goal for the cooling load.

- CT $\Delta T = 14\,°C$
- $\dot{m}_a = 0.996\,m^3/s$
- $\dot{m}_{w,HMX} = 0.0852\,kg/s$
- $\dot{m}_{w,Chiller} = 0.7668\,kg/s$

The detailed results for each component for this case are shown in Table 2. As can be seen, a chiller cooling load of nearly 1.8 ton (6.3KW) was achieved.

Table 2 Baseline PDHS - Closed Loop Water (PDHS-CLW)

Inlet Condition	Case 1
Air Dry Bulb Temp., °C	35.00
Air Relative Humidity, %	47.00
Air Wet Bulb Temp., °C	25.50
Air Flow Rate, m^3/s	0.996
Air Pressure, kPa	101.3
Compressor Outlet Conditions	
Air Dry Bulb Temp., °C	44.87
Air Relative Humidity, %	30.64
Air Pressure, kPa	111.8
Pressure Ratio	1.100
Efficiency, %	90.00
Power, kW	11.562
HMX Air Conditions	
Air In Dry Bulb Temp., °C	44.47
Air Out Dry Bulb Temp., °C	43.31
HMX Water Conditions	
Water Inlet Temp., °C	39.58
Water Outlet Temp., °C	43.24
Water Mass Flow Rate, kg/s	0.0852
Water Heat Flow, kW	1.290
Water Cooler Conditions	
Water Inlet Temp., °C	43.24
Water Outlet Temp., °C	39.30
Water Heat Flow, kW	1.389
Expander Outlet Conditions	
Air Dry Bulb Temp., °C	34.87
Air Relative Humidity, %	44.14
Air Pressure, kPa	101.3
Efficiency, %	90.00
Power, kW	8.720
Chiller Conditions	
Water Inlet Temp., °C	39.58
Water Outlet Temp., °C	41.58
Water Mass Flow Rate, kg/s	0.7668

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Table 2 – *Continued from previous page*

Load, kW	6.332
Load, Ton	1.799
Cooling Tower Conditions	
Air Inlet Temp., °C	34.87
Air Outlet Temp., °C	41.35
Water Inlet Temp., °C	41.36
Water Outlet Temp., °C	39.58
Water Mass Flow Rate, kg/s	0.852
Load, kW	6.233
System Performance	
COP	2.23

4.3 Baseline PDHS - Open Loop Water (PDHS-OLW)

In this section is presented the results for the open loop configuration. Again, the goal is to achieve the highest possible chiller load.

For this configuration, a cooling tower model with a lower ΔT equal to $8\ °C$ was used and different air flow rate and the water inlet temperature to the HMX were considered keeping the water flow rates in the HMX and in the Chiller constant. The lower cooling tower performance nevertheless enabled the desired cooling load as can be seen in the results below.

Here again, a compressor and expander efficiencies of 90% were considered and a pressure ratio of 1.1 was used.

The system has been tested for summer ambient conditions:

- $T_{db} = 35\ °C$
- $\phi = 47\%$
- $P = 101.3\ kPa$

Results graphs are presented below. For this case, we tested the performance for varying air flow rate and water temperature, and are presenting the detailed results and plots for this case. The different parameters below are plotted and calculated:

- Chiller Load in Tons
- Overall System COP:

$$COP = \frac{Chiller\ Load}{Comp\,Work\ -\ Exp\,Work\ +\ Pump\,Work} \tag{43}$$

- PDHS COP:

$$COP_{PDHS} = \frac{PDHS\ Cooling\ Capacity}{Comp\,Work\ -\ Exp\,Work} \tag{44}$$

Results (figures 6 and 7) show that the chiller load and overall system COP increase with the air flow rate and decrease when the water temperature increases. The 2^{nd} observation is logical since hotter water temperature results in less cooling in the HMX. As shown in Fig. 6, clearly COPs in the range of 2-10 can be achieved.

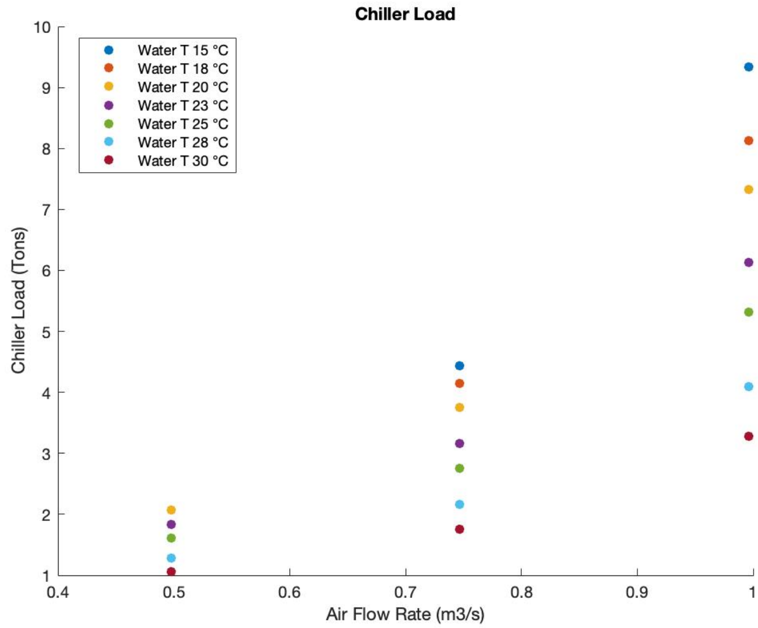

Fig. 5 *Evolution of the chiller load in tons with the air flow rates at different water temperatures for the PDHS-OLW*

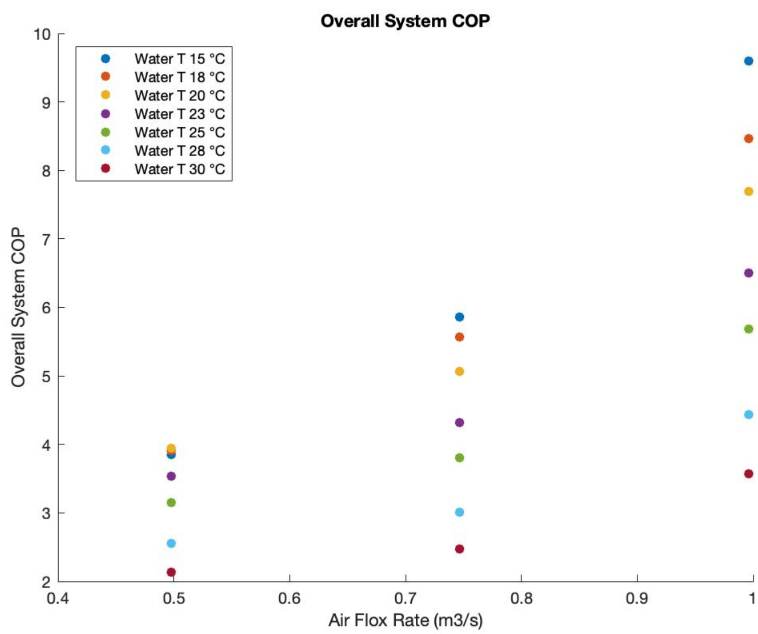

Fig. 6 *Evolution of the overall system COP with the air flow rates at different water temperatures for the PDHS-OLW*

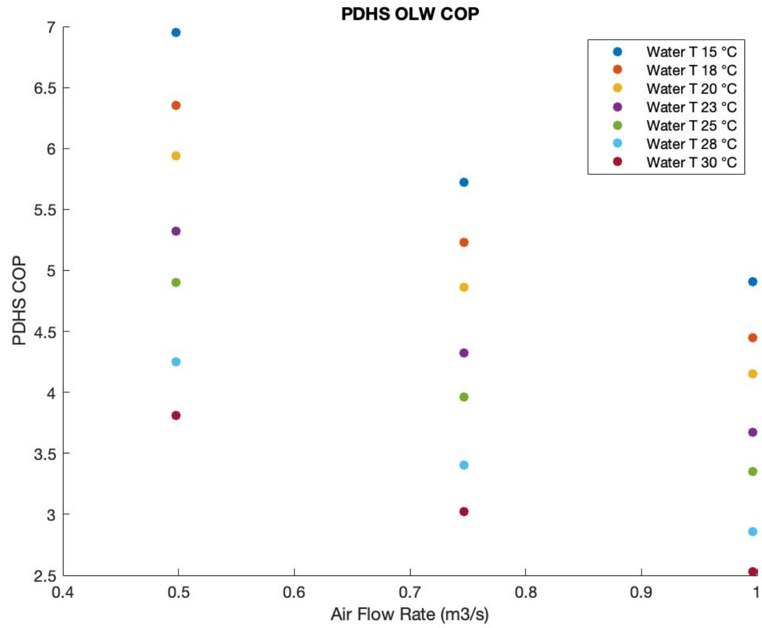

Fig. 7 *Evolution of the PDHS COP with the air flow rates at different water temperatures for the PDHS-OLW*

Figure 7 presents the PDHS COP alone and is particularly relevant in the case a configuration without a chiller and a CT was to be considered to provide cooling using the PDHS system alone. This can generally be done if the air at the exit of the expander reaches a low enough temperature, say below 20°C. Taking a look at the exit air temperature to the expander show that this can only be done for low water temperature and low air flow rate i.e. water temperature of 15 °C and below and air flow rate of 0.498 m^3/s. This corresponds to the cases in the lower left cases in Figures 7 (shows COP) and Figure 8 (shows expander outlet temperature which is the source of the cooling water).

To achieve more cooling, placing an additional HX between the compressor and the HMX was considered. However, this additional HX did not result in the extra cooling expected as one can see looking at the results below in Table 3.

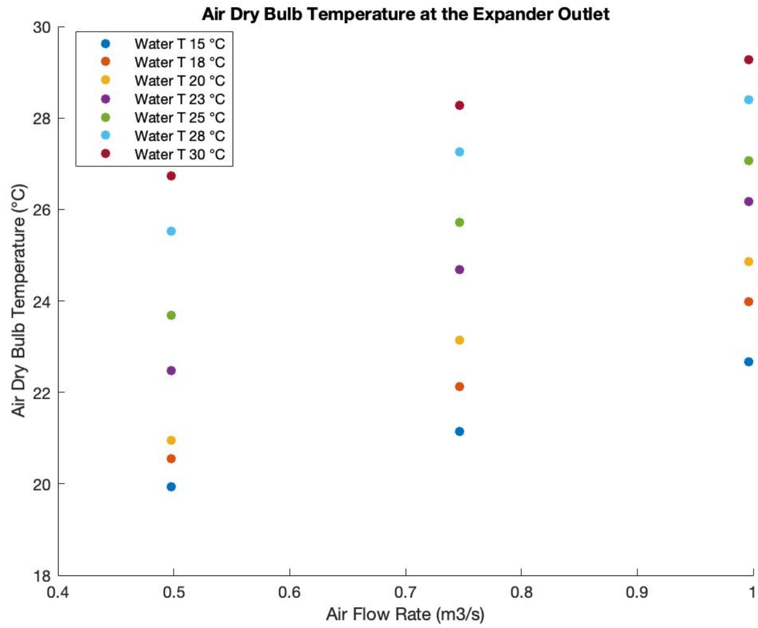

Fig. 8 *Evolution of the air dry bulb temperature at the outlet of the expander with the air flow rate at different water temperatures for the PDHS-OLW*

Table 3 Baseline PDHS - Open Loop Water (PDHS-OLW)

Inlet Condition	Without HX	With HX
Air Dry Bulb Temp., °C	35.00	35.00
Air Relative Humidity, %	47.00	47.00
Air Wet Bulb Temp., °C	25.50	25.50
Air Flow Rate, m^3/s	0.498	0.498
Air Pressure, kPa	101.3	101.3
Compressor Outlet Conditions		
Air Dry Bulb Temp., °C	44.87	44.87
Air Relative Humidity, %	30.64	30.64
Air Pressure, kPa	111.8	111.8
Pressure Ratio	1.100	111.8
Efficiency, %	90.00	90.00
Power, kW	5.781	5.781
HX Air Conditions		
Air In Dry Bulb Temp., °C	N.A.	44.08
Air Out Dry Bulb Temp., °C	N.A.	40.56
HX Cooling Load, kW	N.A.	1.999
HMX Air Conditions		
Air In Dry Bulb Temp., °C	44.08	40.56
Air Out Dry Bulb Temp., °C	25.74	24.44
Air Out Relative Humidity %	82.04	88.95
HMX Water Conditions		
Water Inlet Temp., °C	15.00	15.00
Water Outlet Temp., °C	25.19	23.95

Continued on next page

Table 3 – *Continued from previous page*

Water Mass Flow Rate, kg/s	0.24	0.24
Water Heat Flow, kW	10.119	8.895
Expander Outlet Conditions		
Air Dry Bulb Temp., °C	20.27	19.93
Air Relative Humidity, %	100	100
Air Pressure, kPa	101.3	101.3
Efficiency, %	90.00	90.00
Power, kW	4.109	4.090
Chiller Conditions		
Water Inlet Temp., °C	28.28	27.94
Water Outlet Temp., °C	33.08	32.75
Water Mass Flow Rate, kg/s	0.366	0.366
Load, kW	7.277	7.274
Load, Ton	2.067	2.066
Cooling Tower Conditions		
Air Inlet Temp., °C	20.27	19.93
Air Outlet Temp., °C	33.08	32.74
Water Inlet Temp., °C	33.09	32.75
Water Outlet Temp., °C	28.28	27.94
Water Mass Flow Rate, kg/s	0.366	0.366
Load, kW	7.277	7.274

5. CONCLUSIONS

This study demonstrates that the cooling system consisting of a chiller and a cooling tower which is supplemented by a precooling and dehumidifying system composed of a reversed Brayton cycle can be an effective strategy to reduce the steam condensation temperature in a power plant condenser and thus increase overall efficiency. The study also demonstrates that under certain scenarios, the PDHS alone can act as a cooling system to provide the cooling load to a building.

Providing water at constant temperature to the HMX (as from a building water supply) shows greater improvements to the cooling capacity of the system. While the system and the mass flow rates considered in this study were for the lower cooling load applications for the chiller (around 2 Tons), scaling up this system could potentially be a new eco-friendly solution for cooling larger buildings.

ACKNOWLEDGMENTS

The work was supported by the Gas Technology Institute (GTI Energy). This support is gratefully acknowledged.

NOMENCLATURE

A	area	(m^2)	Pr	Prandtl Number	(-)
c	specific heat capacity	($J.kg^{-1}.K^{-1}$)	$\dot{q}$	volume flow rate	($m^3.s^{-1}$)
COP	Coefficient Of Performance	(-)	$\dot{Q}$	Heat Load	(W)
CR	compression ration	(-)	Re	Reynolds Number	(-)
d	diameter	(m)	St	Stanton Number	(-)
e	specific enthalpy	($J.kg^{-1}$)	T	temperature	(C)
ER	expansion ration	(-)	w	absolute humidity	(-)
h	heat transfer coefficient	($W.K^{-1}.m^{-2}$)	$\dot{W}$	work	(W)
j	Colburn j-factor	(-)	α	thermal diffusivity	($m^2.s^{-1}$)
k	thermal conductivity	($W.K^{-1}.m^{-1}$)	ϕ	humidity ratio	($\%$)
$\dot{m}$	mass flow rate	($kg.s^{-1}$)	ρ	density	($kg.m^{-3}$)
NTU	Number of Transfer Unit	(-)	μ	dynamic viscosity	($Pa.s$)
P	pressure	(Pa)	ν	cinematic viscosity	($m^2.s^{-1}$)

SUBSCRIPTS

a	air	EXP	expander
AB	Air Blower	i	inlet
ai	inlet air	is	isentropic
ao	outlet air	o	outlet
C	compressor	s	surface
CT	Cooling Tower	w	water
db	dry bulb	wi	inlet water
dp	dew point	wo	outlet water

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