Water-Energy Nexus Modeling in Cooling Towers and Hot Springs

A Thesis

Presented in Partial Fulfillment of the Requirements for the

Degree of Master of Science

with a

Major in Mechanical Engineering

in the

College of Graduate Studies

University of Idaho

by

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December 2021

Abstract

Water and energy resources are essential to humanity’s existence which have shaped society's development. The limited water and energy resource forced humans to seek new ways to use them more efficiently. The recent increase in water and energy demand makes any water and energy conservation effort very valuable. In the absence of conservation efforts, providing sufficient supplies of these resources may not be applicable in the future. The present thesis found two applications for implementing a more efficient approach to their operation for water and energy conservation purposes.

Cooling towers are equipment for dissipating the excess heat by water evaporation. The study elaborates on the role of cooling tower modeling in implementing any water and energy consumption improvement plan by presenting a modeling approach, categorizing various methods of modifying water and energy consumptions through past studies. To map the future studies, summarize and organize the past efforts and find future research trends for upgrading water and energy usage in cooling towers have been done. The practical approaches to save water and energy such as design cooling tower based on the ambient air conditions, add dry section to existed wet cooling tower, and employ variable frequency drive fans in forced draft cooling tower have been proposed.

An improvement plan has been implemented on the existed mechanical draft wet cooling tower located at the main campus of University of Idaho by using Data Acquisition Device (DAQ) and ambient air condition real data. By adjusting outlet water temperature in the cooling tower developed model, the cooling towers load can be managed, which reduces its water and energy consumption. Consequently, the improvement plan reduced cooling tower fans energy consumption and water loss due to evaporation. For one month of operation of the improved plan, up to 28035.50 kWh energy saving, 179.49 m3 of water saving, and $1125.17 cost savings have been recorded.

The present thesis found low-temperature geothermal energy sources such as hot springs distributed in different areas as a potential for implementing small-scale energy plants for indirect energy usage. A techno-enviro-economical tool has been developed to approximate the technical, economic, and environmental aspects of hot spring power plant projects, including power generation capacity, initial investment, and possible income from the power plant. The developed technical model estimated the hot springs power generation plants from 9.3 kW to 303 kW depending on the temperature and water discharge mass flow rate. The developed economic model calculated the payback period of investing in a hot spring energy plant as low as six years, which is better cost-effective than other geothermal energy plants.

Acknowledgements

The author would like to acknowledge the Department of Mechanical Engineering at the University of Idaho and College of Engineering at the University of Idaho the John Bieber Scholarship and College of Engineering Development scholarship as well as the University of Idaho Facility for financial assistance. Additionally, the author deeply appreciates Dr. Behnaz Rezaie for her dedication, friendly supervision, and motivation that enabled this thesis to be possible. Finally, the author would like to acknowledge his family for their continuous and unparalleled love, help, and support.

Dedication

I dedicate my work to my family and friends. A special feeling of gratitude to my loving parents, whose words of encouragement and push for tenacity ring in my ears

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List of Nomenclature

A	Area (m^2)
a	Number of amortization years
a_{fi}	Surface area of the fill per unit volume of fill
C	Cost ($)
C_{pw}	Specific heat at constant pressure and at water temperature (J/kg K)
E	Energy (kJ)
f	Correction factor according to Berman
g	Acceleration of gravity (m/s^2)
h	Specific enthalpy (kJ/kg)
h_d	Mass transfer coefficient
i	The interest rate (%)
LEC	Levelized Energy Cost ($/kWh)
$\dot{m}$	Mass flow rate (kg/s)
Me_M	Transfer coefficient or Merkel number according to the Merkel model
Me_P	Merkel number according to the Poppe model
Me_e	Merkel number according to the effectiveness–(NTU) model
N	Lifetime (year)
P	The price ($)
$\dot{Q}$	Heat transfer crossing the component boundaries (kW)
q	Number of yearly payments
SIC	Specific Investment Cost ($/kW)
T	Temperature (°C)
t	Time (h)
U	Heat exchanger overall heat transfer coefficient ($kW/m^2\ {}^{o}C$)
V	Stream velocity (m/s)
$\dot{W}$	Work transfer crossing the component boundaries (kW)
Y	Yearly payment
z	Elevation (m)

Greek Symbols

ε	Effectiveness ratio
η	Efficiency (%)

Δ	Difference
ω	Humidity ratio (kg water vapor/kg dry air)
ρ	Density (kg/m^3)

Subscripts

a	Air
c	Cold
cond	Condenser
e	Electricity
evap	Evaporator
fi	Filler
fr	Front
h	Hot
in	Inlet
lm	LMTD
Loss	Loss water
m	Mechanical
max	Maximum
min	Minimum
net	Network
p	Pump
op	operation
out	Outlet
ref	Reference
s	Isentropic
ss	Supersaturated
v	Vapor
sys	System
T	Turbine
tot	Total
w	Water
wf	Working fluid
1, 2, ….	Cycle state

Abbreviations

APD	Advanced Pinch Design

CFC	Chlorofluorocarbons
CFM	Cubic Foot per Minute
CHP	Combined Heat and Power
COP	Coefficient of Performance
CRF	Capital Recovery Factor
CT	Carbon Tax
DAQ	Data Acquisition Device
EES	Engineering Equation Solver
Eq	Equation
ESCO	Energy Service Companies
FC	Fuel Cost
FCC	Fan Cycling Control
FMC	Frequency-Modulating Control
GWDS	Gravity Water Distribution System
HCCCT	Hybrid Closed Circuit Cooling Tower
HVAC	Heating Ventilation and Air Conditioning
KSD	Kim and Smith
IR	Inflation Rate
LMTD	Logarithmic Mean Temperature Difference
Mercaptobenzothiazole	MBT
MINLP	Mixed-Integer Nonlinear Programming
MSC	Multiple-Speed Control
MW	MegaWatt
MWNT	Multi-Walled Carbon Nanotubes
NE	Net Earnings
NREL	National Renewable Energy Laboratory
NTU	Number of Transfer Unit
OC	Optimum Control
ORC	Organic Rankine Cycle
I&M	Insurance and Maintenance
PI	Proportional Iterative
PP	Payback Period
PPWD	Parallel Path Wet-Dry
PWDS	Pressure Water Distribution System

RH	Relative humidity
RO	Reverse Osmosis
RTD	Resistive Temperature Detectors
SRC	Steam Rankine Cycle
TB	Tax Benefit
TRNSYS	Transient Systems Simulation
UI	University of Idaho
VFD	Variable Frequency Drive
WESCO	Water Efficiency Service Companies

Chapter 1: Introduction

1.1. Motivation

Water and energy resources are essential to humanity's existence which have shaped society development during history. As the world faces growing populations, humans will likely face an increase in energy and water demand. The recent increase in water and energy consumption and the limited available resources make any water and energy conservation effort very valuable. In the absence of conservation efforts, providing sufficient supplies of these resources may not be applicable in the future.

As far as I remember from my childhood in a small town, water was a critical element for life. Because the harvest of rice farmers in our town were highly depended on availability of freshwater. At some point, farmers expanded their rice fields which led to a great shortage of water resources. Our town faced a great water shortage, and farmers were banned from cultivating rice. That made some farmers to immigrate to other regions for new jobs. I lost some of my childhood friends in that immigration. That was my first observation and memory from water shortage in a community that made me think deeper about the importance of water in societies. Negotiating water issues without considering the energy-matter is not meaningful since water and energy are interdependent. In different industries such as power generation and HVAC, water and energy are critical for operating. For example, electric power generation was responsible for 41% of total water withdrawals in the United States in 2015 [1]. Therefore, investigating water and energy matter -termed as water-energy nexus in the current thesis- is a critical topic for the future of human life.

With current increase of water and energy demands, focusing on water consumption models and improving them is necessary for a sustainable future. Therefore, improving water and energy consumption and introducing a new energy resource from natural hot water became the main motivation in the present thesis. The study provides a roadmap towards more efficient water and energy usage by applying the fundamental concept of engineering coupled with computational package to reduce water and energy loss in cooling towers industry and to generate power from an untouched renewable energy source.

1.2. Objectives and Scope

The focus of the study is to investigate different approaches to use water and energy sources in a more efficient way. The present thesis shows different applications which consume water and energy in their operation system or have potential to harness its hydrothermal energy. The modeling studies are conducted using suitable modeling software in every project which is selected based on software's capability, input data, and goals of the project.

Since cooling towers silently evaporate tons of water and energy in each plant, there are great opportunities for reducing its water and energy. The present thesis shows the potential ways to reduce the cooling towers water and energy consumption applicable to energy plants and cooling plants. Also, the possibility of harnessing energy that stored in hot springs investigates in the present study. Hot springs are an important low-temperature geothermal energy source that are distributed on the earth's surface with a natural eruption mechanism. Indirect usage of hot springs has not been studied enough which is covered in the present thesis.

The various objectives of the thesis are summarized as follows:

1) To map the previous studies in modeling and improving performance of cooling towers by categorizing and explaining the objectives and related methods for reducing water and energy consumption.

2) To identify the environmental impacts of cooling towers for proposing methods to improve the environment sustainability.

3) To apply the cooling tower modeling approach on an actual cooling plant (district cooling system in the UI) with the goal of reducing it water and energy consumption.

4) To develop a technical model of power generation from hot springs as well as its economic model for a wide range of hot springs.

5) To investigate the impact of hot spring temperature and water discharge mass flow rate on the capacity of energy generation and economic model of the hot springs' energy plants.

1.3. Outline

The present thesis is comprised of four chapters that all focus on the water-energy nexus modeling of two applications.

Chapter 2 focuses on water conservation in cooling towers since cooling towers are very water-intensive applications in the industry. This chapter is a literature review of the previous studies in improving cooling towers' water and energy consumption by categorizing and explaining various objectives and methodologies. The different types and configurations of the cooling towers are elaborated, then the modeling history of cooling towers is presented, the environmental impacts of cooling towers are described. Potential areas for improving cooling towers' performance and their water and energy reduction are discussed in in that chapter as well.

In Chapter 3, practically a full-size cooling tower located at the University of Idaho energy plant in Moscow campus is modeled and simulated with intention of reducing water consumption. After validating the modeled cooling tower, by using the collected ambient air real data, the improvement plan for reducing cooling tower water and energy consumption is elaborated in this chapter. Reducing the energy, water, cost, and CO2 emission of cooling tower improved plan is also clarified.

Chapter 4 provides a guideline for harnessing energy from hot springs as natural energy storage in power generation plants. The capabilities of hot springs to generate electricity are assessed in the study by using the thermodynamic and financial approaches. A technical and economic model of the hot springs power generation plant in a wide range of temperatures and water discharge mass flow rates power is developed. The impact of hot spring temperature and water discharge mass flow rate on the power generation capacity developed economic model, and environmental aspect of hot spring energy plant are presented in Chapter 4.

The final chapter (Chapter 5) provides conclusions and a summary of attempts described in previous chapters. Furthermore, a sustainable roadmap for the future is outlined in regard to the water-energy nexus. Additionally, future research and discussion on possible complications that stand in the way of a sustainable future are presented.

Chapter 2: Advancing Water Conservation in Cooling Towers through Energy-Water Nexus

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2021 European Journal of Sustainable Development Research 5(3)

https://doi.org/10.21601/ejosdr/10952

2.1 Abstract

Life without water is not possible on the earth, while modern humans need water not only for drinking, sanitization, and agriculture but also for industrial activities including electricity and cooling generations. Hence, emphasis on water sustainability through different sectors including thermoelectric and cooling plants is an intelligent strategy while the tight connections of water and energy guide study towards energy-water nexus investigations. Cooling towers are equipment for dissipating the excess heat by water evaporation or they hidden gates for wasting water. The objective of the present study is to elaborate on the role of cooling towers in improving environment sustainability by presenting various methods of energy and water modeling, categorizing various methods for modifying water and energy consumptions through past studies and mapping future studies. regarding cooling towers. Presenting a history of energy-water modeling methods of cooling towers, the Markel, the Poppe, and the effectiveness– Number of Transfer Unit (NTU) models, has followed by assessing the environmental impact of cooling towers in the form of excess water consumption, plume, and energy usage. Summarizing and organizing the past efforts for upgrading water management in cooling towers have been in two directions either providing more water supply, or modifications of the cooling tower to use less water. Then the different methodologies for each direction are introduced for further elaborations. This study's practical outcome is proposing the methods of improving water sustainability for any cooling towers from past studies to assist engineers in the industry in modifying cooling towers water consumption. Showing the roadmap for the planning future investigations on the cooling towers based on the past efforts is another outcome of the present study to provide an insight for academia with research interest on cooling towers.

2.2 Introduction

The cooling tower is one of the key components in industries such as power generation including renewable (geothermal [2] and solar thermal [3]) and non-renewable [4] power plants, chemical and petrochemical plants [5], refrigeration and air-conditioning plants [6]. Thermoelectric generation and its required cooling are responsible for approx. 10% of the total water demand in the world [7]. The role of the cooling tower is dissipating heat from the hot stream of the process into the air [8] in power plants, district cooling plants, and cooling systems. To address the water scarcity for a sustainable future (in smart cities) cooling towers have to receive special attention. Replacing the evaporation of water for dissipating the heat with another method, or capturing the vapor, or both are the ideas to support water conservation. Considering several cooling towers that are installed already highlights the importance of capturing the vapor and/or reducing evaporation studies.

The design of cooling towers focuses on the water distribution system, fill, and drift elimination. The water distribution system introduces and spreads the process water as evenly over the fill through the use of water canals and nozzles. The fill is a system of packing that delays the fall of water and improves heat transfer, and drift eliminators at the air exit change the direction of airflow to reduce the volume of water transported out [9]. Within cooling towers, water is lost through three main modes. These modes are drift, blowdown, and evaporation [10]. Drift is the water losses associated with wind, evaporation loss occurs due to the heat transfer taking place, and blowdown is utilized to avoid the buildup of minerals and sediments within the cooling water that may damage other components within the system, and. blowdown is also a byproduct of the evaporation processes increasing the concentration of the minerals [10].

Oftentimes, cooling towers are oversized, and thus rarely operate at their design points [11]. One of the primary reasons for oversizing is to ensure proper cooling when ambient temperatures and humidity are high. At high ambient temperature and humidity, it is more difficult to reject heat from a cooling tower. The processing water temperature should be reduced to a specific temperature through the cooling towers, which is designed based on ambient conditions. Most of the time, however, ambient conditions are less extreme, and therefore the large size of the cooling tower is not required in these conditions[11].

National Renewable Energy Laboratory (NREL) has provided a protocol for cooling tower measurement and verification to ensure that water savings are properly recorded [12]. It has been intended for energy service companies (ESCOs) and water efficiency service companies (WESCOs) to determine water savings resulting from cooling tower efficiency projects. When measuring both baseline and post-upgrade operations of a cooling tower, it is suggested that multiple seasons of data be used to establish averages. However, a minimum of one season of cooling data is required for analysis. If a flowmeter is being used, the flow of water should be normalized by multiplying the use of water over the cooling season by the sum of the wet-bulb temperature ratios. The sum of wet bulb temperature ratios is found by dividing the historical monthly wet-bulb temperatures for a location by the recorded wet-bulb temperatures during that same month. These ratios can then be added together for all the cooling seasons. The result scales the water-usage during the testing period to historical averages.

The major goal of this study is shedding light on water conservation in cooling towers since water scarcity is a critical problem. The present study's objective is to map the previous studies in improving cooling towers water and energy consumption by categorizing various approaches and explaining their objectives and related attempts. As a foundation for presenting former studies, first different types and configurations of the cooling towers are elaborated then the modeling history of cooling towers are shown, the environmental impacts of cooling towers are described. The past studies are concluded in the form of practical methods for improving the cooling towers' energy and water conservations. Also, the direction for future studies is presented. The outcomes of this literature review study assist engineers with practical results and academia to show the future study roadmap, while supporting the environmental sustainability of cooling towers.

2.3 Cooling Tower Types

Cooling towers are classified based on different characteristics, although the following are the major ones:

- Airflow
  - Mechanical draft (Figure 2.1 (a) and (b))
  - Natural draft types (Figure 2.1 (c) and (d)) [13]
- Water consumption
  - Wet (Figure 2.1),
  - Dry (Figure 2.2),
  - Combination of wet-dry (hybrid) (Figure 2.3) [14].

Choosing the proper cooling tower for a plant is a balance of different parameters like plant efficiency, capital and operation cost, water consumption, water withdrawal, and environmental impact which are illustrated in Table 2.1 [15].

Table 2. 1: The summary of cooling towers tradeoffs

Cooling system	**Water withdrawal**	**Water consumption**	**Capital cost**	**Ecological impact**
Wet cooling tower	Moderate	Intense	Moderate	Moderate
Dry cooling tower	None	None	High	Low

2.3.1. Wet Cooling Towers

Wet cooling towers operation is changing water from liquid to vapor to release the excess heat in the cooling cycle [16]. The thermal performance and stability of the wet cooling tower are better than the dry cooling tower because thermodynamic parameters rely less on ambient temperature [17]. The advantages of wet cooling towers are:

- Saving energy and costs,
- Reducing Chlorofluorocarbons (CFC)s usage,
- Improving life cycle cost-effectiveness [18].

Wet cooling towers are also categorized by the movement of water and air inside of the cooling tower as follows:

- the crossflow towers: air flows horizontally across the falling water as it shows in Figure 2.1 (b) and (d),
- counterflow: the upward airflow that directly opposes the downward flow of the water providing which is shown in Figure 2.1 (a) and (c) [19].

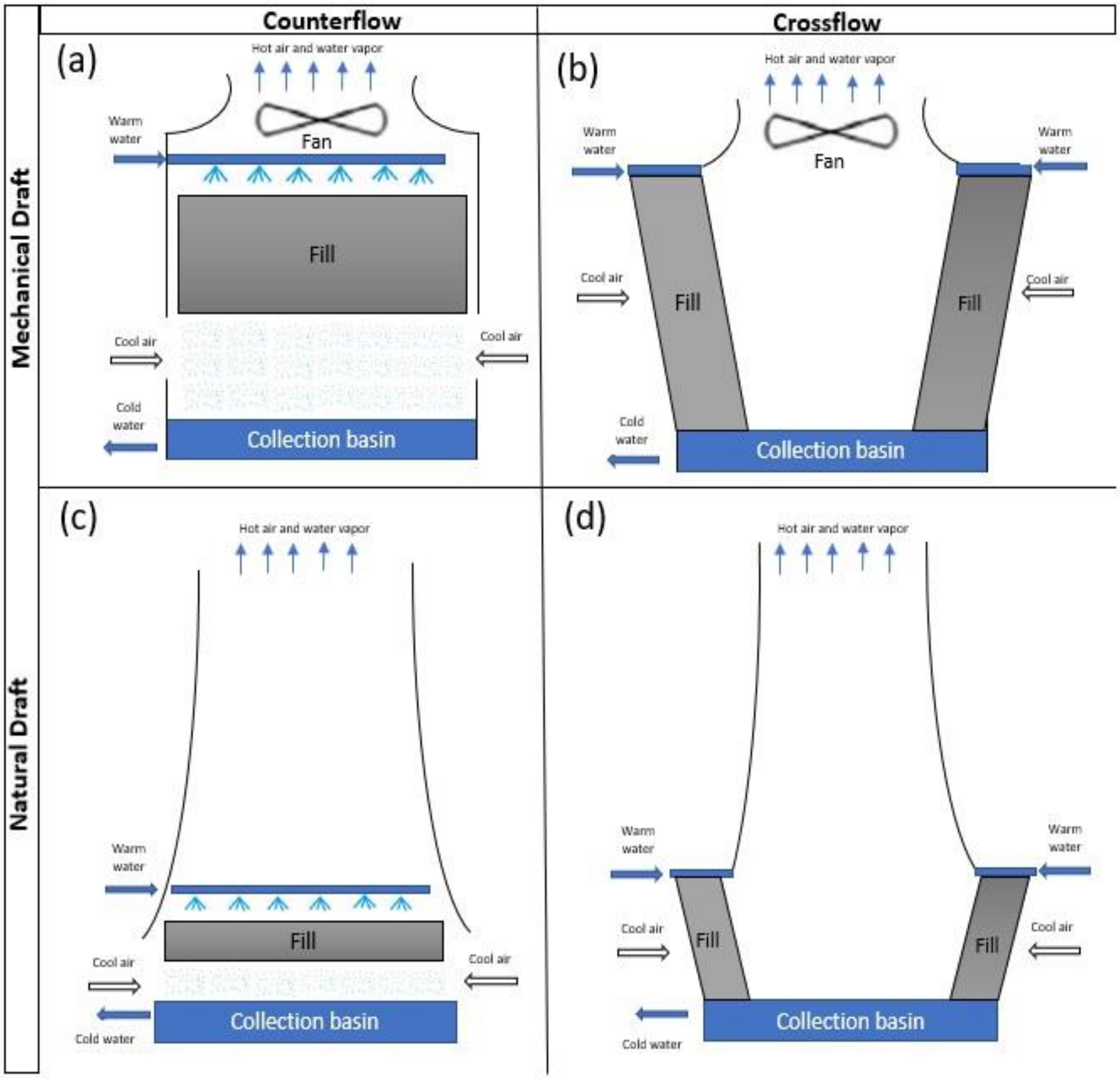

Figure 2. 1: The schematic of different types of wet cooling towers

Since counterflow cooling towers have been used more commonly, there are more studies on this type in comparison with crossflow [20]. The schematic of different types of wet cooling towers is shown in Figure 2.1.

In wet recirculating cooling tower systems, the warm water from a plant process condenser is pumped to a cooling tower where heat is dissipated to the ambient environment by evaporating water [21]. Surrounding ambient air can be forced into the cooling tower by one or more fans to accelerate heat dissipation which results in increasing the temperature of the incoming air [10]. Once the water has been cooled, it can then be returned to the plant process for reuse.

The closed recirculating cooling towers withdraw water, then circulate that instead of discharging [22]. Significant amount of water gets consume in the cooling towers, particularly freshwater in the power generation plants, depends on the type of fuel and power generation technology [23]. Coal has a high moisture content and contaminants that decreases its combustion properties. A coal-fired power plant requires more electricity for its pollution control operation because a coal-fired power plant releases more pollutants than a power plant that uses natural gas. Reducing the combustion properties and using more electricity in pollution control operation results in thermal efficiency reduction in coal-fired power plant. Also, the combined cycle thermoelectric technology uses waste heat to generate power, which enhances the power plant's thermal efficiency compared with steam turbine technology. Consequently, fuel types and power generation technologies influence the power plants thermal efficiency [23], [24]. Power plants with lower thermal efficiency convert less thermal energy to electric power, therefore they require more cool water for condensing [25]. Table 2.2 illustrates the water consumption (loss) of cooling towers in the different thermal cycles. It must be noted that wet cooling towers are not favorable in the region with water shortages due to the high water consumption of them [26]. Water availability is a key parameter in the decision-making of choosing the suitable cooling tower [27].

Table 2. 2: The average water consumption of cooling towers in different types of power plants

Fuel type	**Thermoelectric Technology**	**Cooling water consumption (m^3/MWh)**	**Reference**
Coal	Steam turbine	2.1/ 2.13	[28] [25]
Coal	Supercritical	1.5/1.3	[29]
Coal	Integrated gasification combined cycle	1.44/1.14	[30][31]
Natural gas	Steam turbine	2.6/2.5-4.4/3.12	[28] [32] [33]
Natural gas	Combined cycle	0.9/0.77	[28] [34]
Natural gas	Combined cycle with carbon capture and sequestration	1.48	[30]

2.3.2. Dry cooling towers

A dry cooling tower depends on convective heat transfer that is governed by the dry-bulb air temperature [35]. In a dry cooling tower, heat exchangers are placed in the core of the tower which uses air as the cooling medium, and hot water becomes cold in a closed circulating loop [36]. Dry cooling is applicable by either mechanical draft or natural draft [37]. Figure 2.2 shows dry cooling towers in the form of a mechanical draft and a natural draft. The dry cooling towers in comparison with wet cooling towers:

- Dissipate less heat in the cooling process,
- Consume less water, therefore,
- Need less maintenance [38]

Therefore, it can be concluded that dry cooling towers are better candidates for regions with water scarcity, while they are less effective.

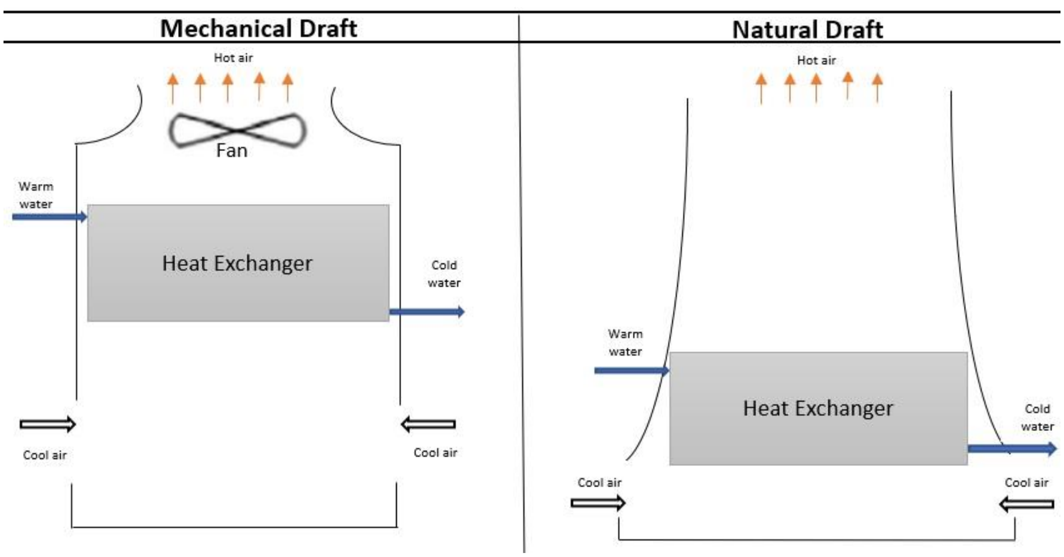

Figure 2. 2: The schematic of natural and mechanical draft dry cooling towers

2.3.3. Hybrid Cooling Towers

In the 1970s, cooling towers both wet and dry were combined as hybrid cooling towers [39]. Therefore, hybrid cooling towers' components have the advantages of both cooling towers [40]. They consume less water while still maintaining a similar heat load rejection capacity. Hybrid cooling towers can be used separately or simultaneously either for water conservation [41] or plume abatement purposes [42]–[44]. A hybrid cooling tower benefits depend on the heat load, the airflow rate, and the ambient air conditions [45]. The performance of hybrid cooling towers was investigated by numerical simulations and exprimnet on various operation conditions [45]. A computational procedure to predict hybrid cooling tower performance in a wide and variable range of working conditions was developed. The outcomes revealed the most important factor in designing the hybrid cooling tower was water to air ratio that was related to the operating condition of the cooling towers. Similarly, the water mass flow rate and air mass flow rate varied from 2.08 kg/s to 3.90 kg/s and 0.5 kg/s to 4 kg/s, respectively in tests. While implementing the dry cooling tower reduced water evaporation in compare with wet cooling tower but it increased the energy consumption. Since the pump power consumption in a wet cooling tower was less than fan in a dry cooling tower (similar performance), by increase of using dry cooling tower section, power consumption boosted from 0.14 kW to 35 kW [45]. The parallel airflow was flowing through both dry and wet sections while the cooling tower load water was running in a series path from the dry cooling tower to the wet cooling tower [46]. It was concluded that the performance of dry cooling and wet cooling towers was dependent on the adjustable airflow to each section. Consequently, when a hybrid setup met the cooling demand, the required power was significantly reduced by lowering the airflow to the dry section. Furthermore, the higher fractions of airflow going to the wet cooling tower was followed with a more favorable COP [45]. The schematic of different natural draft and mechanical draft hybrid cooling towers are shown in Figure 2.3.

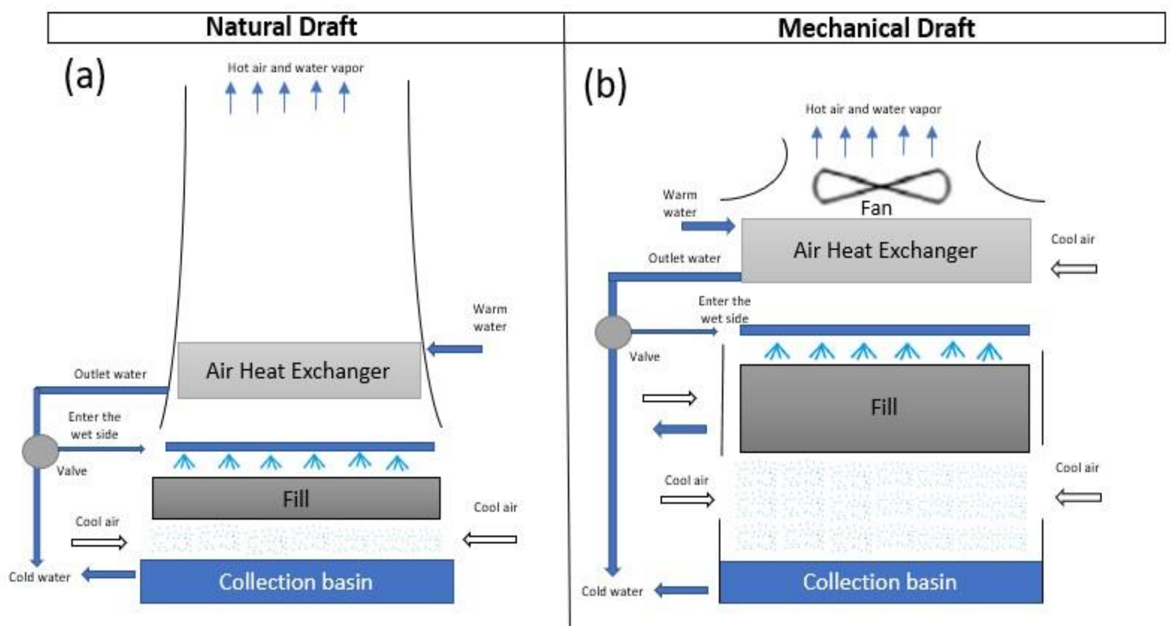

Figure 2. 3: The schematic of natural and mechanical draft hybrid cooling towers

In a study, the hybrid cooling tower was designed by adding several dry cooling towers units to the existing wet cooling tower at a power plant and a refinery station. The study's outcome led to reduce the water consumption and operational cost in the plant [7]. The results showed the total water consumption of the cooling tower at various hybrid ratios was reduced to 38%, leading to a cost savings as well [7]. In a similar study, by three types of cooling tower system dry, wet, and hybrid in a 660 MW power plant in China the thermodynamic models developed to study the effects of the dry bulb, wet bulb temperature, and humidity on the operation [17]. The results indicated that implementing hybrid cooling towers reduced 46% of water consumption compared with a wet cooling tower and reduced 45.84% fan energy usage compared with dry cooling [17]. The optimum hybrid cooling tower were designed with wet/dry cooling ration for a 12.5 MW steam power plant in Iran based on the ambient data of the prior five years [47]. The optimization was set to find theminimum investment and the minimum water consumption in a hybrid cooling tower. The temperature of the outlet water from cooling tower and the evaporation rate were determined by the ambient condition variations and also changing the fan speed in both dry and wet cooling towers. When the outlet water temperature from the dry heat exchanger was more than 23 °C, the water was conducted to wet cooling tower to provide required cold water (less than 23 °C). Otherwise, cooling process was performed without using wet cooling tower to reduce power plant water consumption. When wet cooling towers were not needed, the water losses were zero, therefore the annual decline of power plant water consumption was 1.21 million cubic meter. Besides, by increasing the share of dry side in a hybrid cooling tower, water consumption was significantly reduced. It was shown that usingan optimum designed hybrid cooling tower reduced 63% water consumption [47].

The flow resistance of the natural draft of a hybrid cooling tower with parallel airside cooling sections of wet and dry cooling was investigated to determine the annual thermal and economic performance in different ambient conditions [48]. The study was on wet, dry, and hybrid cooling towers coupled with a 660 MW steam power plant. Results indicated the operation costs of hybrid cooling tower was lower than dry and wet cooling towers while the net benefits of $200/h in 2010 and $100/h in 2018 were attained, respectively [48].

An experimental study was conducted by a finned tube instead of a bare tube for increasing the heat transfer surface area to release more excess heatin the dry section of a hybrid cooling tower. The impact of finned tubes in a Hybrid Closed Circuit Cooling Tower (HCCCT) was studied by switching from wet to dry modes depending upon the cooling load [49]. The study showedthe impacts of bare-type copper tubes and finned tubes on the cooling capacity in a hybrid closed circuit cooling tower was 22% and 26% increase in the cooling capacity of wet and dry modes, respectively. However, the operation cost was increased due to the higher pressure drop [49]. Additionally, the use of finned tubes required only 80% of the airflow that regular tubes required.

A method of retrofitting existing wet cooling towers to reduce water consumption was introduced by using air-cooled heat exchangers in an existing wet cooling tower [39]. The existing wet fill was lowered to make room for the dry section. An airflow control system was introduced for controlling the airflow to the minimum water loss. The system is known as a Parallel Path Wet-Dry (PPWD) system which is shown in Figure 2.3 (b). In PPWD system, water flews first into the dry section and then into the wet section. The air streams were in parallel and only mix in the plenum above the dry section. An annual water consumption reduction between 4.3% to 6.7% was estimated, depending upon the dry section heat exchanger design by the hybrid PPWD cooling tower in comparison to a wet type.

2.4. Modeling Studies on Cooling Towers

Modeling of cooling towers started with Lewis who analyzed the cooling tower thermodynamically in the 20th century [50]. Then, Robinson's equations for the cooling tower were developed based on Lewis' work [51]. Merkel proposed a model in which enthalpy was potential as the driving force for air-water exchange, heat, and mass convection transfer employing Lewis number [52]. Some underestimations of cooling towers sizing existed in Merkel's model [52]. Bourillot modified the Merkel model about insufficiency to define water consumption over excess simplification of the model [54]. Poppe and Roger added the heat and mass transfer coefficients for supersaturated and unsaturated air in their model to the Bourillot model and defined a new Merkel number formulation [55]. Braun et al. and Jaber et al. developed the effectiveness– Number of Transfer Unit (NTU) model and found a method for calculation of the cooling tower performance by improving the Merkel's model [56]–[58]. Both the Merkel and effectiveness-NTU models could accurately predict the outlet water temperature, both were inadequate in the evaluation of the evaporated water flow rate and properties of the outlet air while the Poppe model was able to predict the states of the outlet air accurately [59]. Finally, Klopper proposed a model based on the Poppe model to calculate the water evaporation rate based on the actual value of the Lewis factor [60].

Before further explanations, it must be noted that the wet cooling tower models are defined based on the ambient properties. The thermal performance of wet cooling towers strongly depends on the humidity and temperature of the ambient [61].

2.4.1. The Merkel Model

The air and water flow in the counterflow cooling tower are shown in Figures 2.4 and 2.5 where the air is in counterflow with a downwards flowing water stream. These figures show an idealized model of the interface between the water and the air for a counterflow cooling tower filler materials. Figure 2.4 shows a control volume in the filler. Figure 2.5 illustrates an airside control volume of the filler shown in Figure 2.4. Eq. (2.1) and (2.2) are obtained from mass and energy balances of the control volumes shown in those figures. The change in the enthalpy of the air-water vapor mixture and the change in water temperature as the air travel distance changes are shown in the Eq. (2.1), and (2.2), respectively. The main assumption of the Merkel method is neglecting the evaporation in the water mass flow rate ($dm_w = 0$). Also, the Merkel number can be obtained Eq. (2.3).

$$\frac{di_{ma}}{dz} = \frac{h_d a_{fi} A_{fr}}{m_a} \left(i_{masw} - i_{ma}\right) \tag{2.1}$$

$$\frac{dT_w}{dz} = \frac{m_a}{m_w} \frac{1}{c_{pw}} \frac{di_{ma}}{dz} \tag{2.2}$$

$$Me_M = \frac{h_d a_{fi} A_{fr} dz}{m_w} = \int_{T_{w,out}}^{T_{w,in}} \frac{c_{pw} dT_w}{\left(i_{masw} - i_{ma}\right)} \tag{2.3}$$

Here, i_{ma} and i_{masw} are the enthalpy of the air-vapor mixture per unit mass of dry air and the enthalpy of saturated air at water temperature in J/kg, respectively. C_{pw} is water specific heat at constant pressure and in J/kg K. Also, m_w and m_a are discharging the water mass and air mass from the cooling tower in kg, respectively. Also, Me_M and h_d denote transfer coefficient according to the Merkel model and the mass transfer coefficient, respectively. a_{fi}" is the surface area of the fill per unit volume of fill in m^2, and T_w is water temperature in °C.

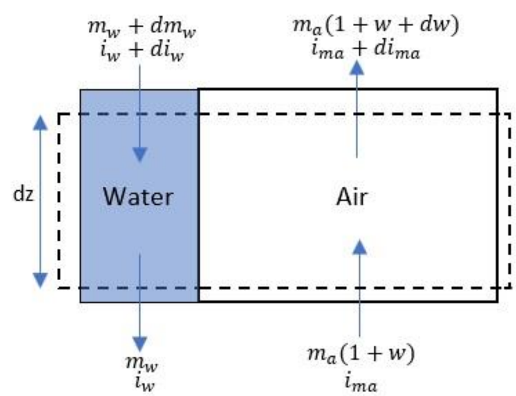

Figure 2. 4: Control volume related to the cross-section of the filler section

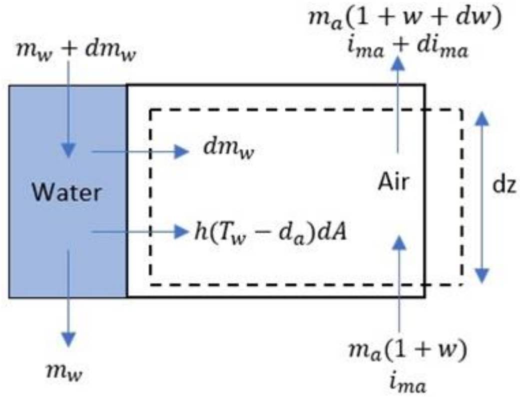

Figure 2. 5: Air volume control in the filling section

2.4.2. *The Poppe Model*

Merkel model has some simplifying assumptions such as:

- considering the Lewis factor relating to heat and mass transfer is equal to 1
- assuming that the air leaving the fill section is saturated with water vapor
- neglecting the reduction of water flow rate by evaporation in the energy balance [60]

The Poppe model is developed without the simplifying assumptions of the Merkel model. Thus, the control volumes in Figures 2.4 and 2.5 are still applicable to this model. Kloppers modified the Poppe model by derivation of the original equation to wet cooling towers as shown in Eq. (2.4) and (2.5) for unsaturated air conditions and Eq. (2.8) and (2.9) for supersaturated air conditions [60]. When the air becomes saturated before it leaves the fill, the potential for heat and mass transfer still exists because the water temperature is still higher than the temperature of the air. At this point, the excess vapor condenses as a mist and is suspended in the air which leads to having a supersaturated air [62]. Le_f is the coefficient of Lewis, according to Eq. (2.6) for unsaturated air conditions and Eq. (2.10) for supersaturated air conditions [63]. The Merkel number according to the Poppe approach is given by Eq. (2.7) for unsaturated air conditions and Eq. (2.11) for supersaturated air conditions.

$$\frac{dw}{dT_w} = \frac{c_{pw}\frac{m_w}{m_a}(w_{sw} - w)}{i_{masw} - i_{ma} + \left(Le_f - 1\right)[i_{masw} - i_{ma} - (w_{sw} - w)i_v] - (w_{sw} - w)c_{pw}T_w]} \tag{2.4}$$

$$\frac{di_{ma}}{dT_w} = \frac{m_w c_{pw}}{m_a}(1 + \frac{c_{pw}T_w(w_{sw}-w)}{i_{masw}-i_{ma}+\left(Le_f-1\right)[i_{masw}-i_{ma}-(w_{sw}-w)i_v]-(w_{sw}-w)c_{pw}T_w]}) \tag{2.5}$$

$$Le_f = 0.865^{0.667}\frac{(\frac{w_{sw} + 0.622}{w + 0.622} - 1)}{\ln(\frac{w_{sw} + 0.622}{w + 0.622})} \tag{2.6}$$

$$\frac{dMe_P}{dT_w} = \frac{c_{pw}}{i_{masw} - i_{ma} + \left(Le_f - 1\right)[i_{masw} - i_{ma} - (w_{sw} - w)i_v] - (w_{sw} - w)c_{pw}T_w]} \tag{2.7}$$

$$\frac{dw}{dT_w} = \frac{c_{pw}\frac{m_w}{m_a}(w_{sw} - w_{sa})}{i_{masw} - i_{ss} + \left(Le_f - 1\right)(i_{masw} - i_{ss} - (w_{sw} - w_{sa})i_v + (w - w_{sa})c_{pw}T_w) + (w - w_{sw})c_{pw}T_w} \tag{2.8}$$

$$\frac{di_{ma}}{dT_w} = c_{pw}\frac{m_w}{m_a}(1 + \frac{c_{pw}T_w(w_{sw} - w_{sa})}{i_{masw} - i_{ss} + \left(Le_f - 1\right)\left\{i_{masw} - i_{ss} - (w_{sw} - w_{sa})i_v + (w - w_{sa})c_{pw}T_w\right\} + (w - w_{sw})c_{pw}T_w}) \tag{2.9}$$

$$Le_f = 0.865^{0.667}\frac{(\frac{w_{sw} + 0.622}{w_{sa} + 0.622} - 1)}{\ln(\frac{w_{sw} + 0.622}{w_{sa} + 0.622})} \tag{2.10}$$

$$\frac{dMe_P}{dT_w} = \frac{c_{pw}}{i_{masw} - i_{ss} + \left(Le_f - 1\right)(i_{masw} - i_{ss} - (w_{sw} - w_{sa})i_v + (w - w_{sa})c_{pw}T_w) + (w - w_{sw})c_{pw}T_w} \tag{2.11}$$

Where w is the amount of humidity in the cooling tower in kg. i_{ss} and i_v are the enthalpy of supersaturated air per unit mass of dry air and the enthalpy of the water vapor in J/kg, respectively. Also, w_{sw} and w_{sa} denote the ratio of water saturation at the water temperature and the humidity ratio of saturated air at temperature T_a, respectively. Le_f is Lewis factor, and Me_P is Merkel number according to the Poppe model.

2.4.3. *The Effectiveness–(NTU) Model*

According to Jaber and Webb who design cooling towers by effectiveness–(NTU) method, Eq. (2.12)-(2.19) are developed [64]. Eq. (2.12), and (2.13) correspond to the differential equation of the heat exchanger. Based on Eq. (2.12), “m_a“ can be greater than “$m_w c_{pw} /(di_{masw} /dT_w)$“ or less; Thus, C_{max} is defined as the maximum amount of these two values, and C_{min} is defined as the minimum amount of these two values. Also, Eq. (2.18) calculates the Merkel number according to the effectiveness–(NTU) approach when the dry air mass flow rate (m_a) is greater than “$m_w c_{pw} /(di_{masw} /dT_w)$“, and Eq. (2.19) corresponds to this number when m_a is less than “$m_w c_{pw} /(di_{masw} /dT_w)$“ [64].

$$\frac{d(i_{masw} - i_{ma})}{(i_{masw} - i_{ma})} = h_d \left(\frac{\frac{di_{masw}}{dT_w}}{m_w c_{pw}} - \frac{1}{c_{pw}} \right) dA \tag{2.12}$$

$$\frac{d(T_h - T_c)}{(T_h - T_c)} = -U \left(\frac{1}{m_h c_{ph}} - \frac{1}{m_c c_{pc}} \right) dA \tag{2.13}$$

$$C = C_{min}/C_{max} \tag{2.14}$$

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{m_w c_{pw}(T_{w,in} - T_{w,out})}{C_{min}(i_{masw,in} - f - i_{ma,in})} \tag{2.15}$$

$$f = (i_{masw,out} + i_{masw,in} - 2i_{masw,m})/4 \tag{2.16}$$

$$\text{NTU} = NTU = \frac{1}{1-C} \ln \frac{1-\varepsilon C}{1-\varepsilon} \tag{2.17}$$

$$Me_e = \frac{c_{pw}}{di_{masw}/dT_w} NTU \tag{2.18}$$

$$Me_e = m_a NTU/m_w \tag{2.19}$$

Here, ε and f are the effectiveness ratio and a Berman’s correction factor, respectively. $i_{masw,m}$ is the enthalpy of saturated air at the mean water temperature in J/kg. $T_{w,in}$ and $T_{w,out}$ denote the inlet and outlet water temperature of cooling towers in ℃, respectively. U is overall heat transfer coefficient in $\frac{W}{m^2 °C}$. Q is the heat transfer rate in W. NTU shows the number of transfer units, and Me_e is Merkel number according to Effectiveness–(NTU) model. The Merkel, Poppe, and e-NTU models have been implemented in many cooling towers' studies with different objectives. Based on the modeling types and objectives, research papers have been summarized in Table 2.3 to map the applications of each cooling tower modeling methods since 2003.

Several studies were conducted to compare the Merkel, Poppe, and e-NTU models through experimental observation under test conditions. The summary of those studies can be as the following categories:

- **Air outlet temperature**: When the outlet air from the cooling tower was saturated, Merkel and e-NTU models were capable of predicting the temperature of that, while the Poppe model did not need any assumption for estimation of the outlet air temperature from the cooling tower [65].
- **Water outlet temperature:** When the draft in the cooling tower was the same, the outlet water temperature from the cooling tower in the three models were the same. A small difference in the outlet water temperature from the cooling tower was observed in Merkel and Poppe models over the outlet air from the cooling tower since outlet air assumed saturated in the Merkel model [66].
- **Heat rejected:** The loss of water due to evaporation in the cooling tower reduced the water outlet mass flow rate, which was ignored in the energy equation in the Merkel and e-NTU models while it was seen in the Poppe model. The estimated heat rejection rate by the cooling tower was larger by the Poppe model in comparison with Merkel and e-NTU models [66].
- **Evaporation rate:** The water evaporation rate in the cooling tower was underestimated by the Merkel model compared to the Poppe model [67]. Since evaporation was an important factor in designing the hybrid cooling towers, using the Poppe model was preferred [66], [68].
- **Lewis factor:** The Poppe model proposed a reliable equation to determine the Lewis factor value [65]. While in the Merkel model the Lewis factor was a constant number of 1. Most researchers believed that Merkel's assumption was not accurate since the Lewis factor was between 0.6 to 1.3 [69].

Table 2. 3: Modeling approaches and objectives of wet cooling towers studies

Study	**Objective**	**Model**
Irok et al., 2003 [70]	Performance analysis	Merkel
Khan et al., 2004 [71]	Performance analysis	Effectiveness–(NTU)
Söylemez, 2004 [72]	Performance optimization	Effectiveness–(NTU)
Papaefthimiou et al., 2006 [73]	Performance analysis	Merkel
Kranc, 2007 [74]	Performance analysis	Merkel
C. Ren 2006 [75]	Model improvement	Merkel
Jin et al. 2007 [76]	Model improvement	Merkel/Effectiveness–(NTU)
Qi and Liu 2008 [77]	Performance analysis	Poppe

Williamson et al., 2008 [78]	Performance analysis	Merkel
C. Q. Ren, 2008 [79]	Performance analysis	Effectiveness–(NTU)
Tyagi et al., 2008 [80]	Cost optimization	Effectiveness–(NTU)
Klimanek & Białecki, 2009 [81]	Model analysis comparison	Poppe
Marmouch et al., 2009 [82]	Performance analysis	Effectiveness–(NTU)
Lucas et al., 2009 [83]	Performance analysis	Merkel
Rubio-Castro et al., 2011 [84]	Cost optimization	Poppe /Merkel
T.-H. Pan et al., 2011 [85]	Performance optimization	Merkel
Picón-Núnez et al., 2011 [61]	Design optimal Cooling tower	Effectiveness–(NTU)
Smrekar et al., 2011 [86]	Performance analysis	Poppe
Rao & Patel, 2011 [87]	Performance optimization	Merkel
Gololo and Majozi 2012 [88]	Water consumption optimization	Poppe
Picardo & Variyar, 2012 [89]	Model improvement	Merkel
T. Pan et al., 2013 [90]	Performance optimization	Merkel
Grobbelaar et al., 2013 [91]	Performance analysis	Merkel
Khamis Mansour & Hassab, 2014 [92]	Performance analysis	Effectiveness–(NTU)
Hernández-Calderón et al., 2014 [93]	Model improvement	Poppe
Nasrabadi & Finn, 2014 [94]	Performance analysis	Effectiveness–(NTU)
Nasrabadi & Finn, 2014 [95]	Performance analysis	Merkel
Uzgoren & Timur, 2015 [96]	Performance optimization	Poppe
Y. Wang et al., 2015 [97]	Performance analysis	Poppe
Y.-J. Xu et al., 2015 [98]	Performance analysis	Merkel
Keshtkar & Mehdi Keshtkar, 2016 [99]	Performance optimization	Poppe
Singh & Das, 2016 [100]	Performance optimization	Merkel
Singla et al., 2016 [101]	Performance optimization	Merkel
Llano-Restrepo & Monsalve-Reyes, 2016 [102]	Model improvement	Merkel
Qi et al., 2016 [103]	Performance analysis	Poppe
Singh and Das 2017 [104]	Performance optimization	Merkel
Sharqawy et al. 2017 [105]	Performance analysis	Effectiveness–(NTU)
Gilani and Parpanji 2017 [106]	Performance analysis	Poppe
X. Huang et al. 2017 [62]	Performance analysis	Poppe
Y. Li et al. 2017 [107]	Performance optimization	Poppe
Zhou, Zhu, and Ding 2017 [108]	Performance analysis	Poppe
Ayoub et al., 2018 [66]	Model improvement	Merkel/ Poppe/ Effectiveness–(NTU)
Mishra et al., 2019 [109]	Performance analysis	Merkel
González Pedraza et al., 2018 [110]	Model improvement	Merkel
Liao et al., 2019 [111]	Performance optimization	Poppe
Jes´ et al., 2019 [112]	Model improvement	Poppe
Ke et al., 2019 [59]	Performance analysis	Merkel
Gilani et al., 2019 [113]	Performance optimization	Poppe

2.5. Environmental Impact

Using wet cooling towers has some negative impacts on the environment. However, using wet cooling towers reduces the emission of CFCs into the atmosphere, which are strongly harmful to the Ozon layer. The major categories that wet cooling towers cause negative impacts on the environments are with respect to water, plume, and energy.

2.5.1 Water

The wet cooling towers are widely used and considering the high water consumption in this type of cooling towers, and they are not desired in zones with water resources limitation [114]. Many ideas have been proposed and investigated as mitigation strategies for resolving water consumption issues. All approaches are in two categories:

- Improving the design parameters, operations, structure of the cooling towers to reduce water consumption. Many of these studies are explained under improving water conservation in the present study.
- Focusing on the circulating water in the cooling tower for lesser evaporation through different approached.

Some of these methods have been detailed in improving water conservation.

2.5.2 Plume

Cooling towers have environmental impacts by changing the ambient conditions, such as making visible plumes [115] and releasing hazardous materials [116]. When a wet cooling tower releases the moisture, it mixes with cooler atmospheric air while the vapor closely approaches the saturation point, the plume moisture condenses quickly and generates a visible plume [117]. Visible plume is not considered air pollutant but sometimes contains minerals and chemicals that can be hazardous [118]. For safety purposes, some countries have regulated the visible plume by cooling towers [119]. Using hybrid towers instead of evaporative towers reduces the plume by enhancing the performance of cooling towers [42].

The fog harvesting and Atmospheric Water Harvesting (AWH) for capturing the vapor to produce water were new approaches for plume abatement [120], [121]. Experimental data from outlet fog harvester systems demonstrate that the rate of water collection in cooling tower fog harvester was 3 to 5 times more than atmospheric air fog harvester due to a high relative humidity (RH) level in cooling towers. Moreover, the fog source in cooling towers was more permanent than atmospheric air in providing sufficient fog for harvester during the year [122]. Zapping the fog with the beam charged particles (ions) of electrically has improved capturing water droplets from fog harvester by directing droplets to the storage in a low-cost and low-energy method for collecting of 20% to 30% of wastewater in cooling towers [123].

2.5.3 Energy

Using a Variable Frequency Drive (VFD) fan saves energy in cooling towers which have been studied as a Proportional Iterative (PI) feedback controller with a temperature zone setting to manage the water outlet temperature [124]. The goal was to reduce energy consumption and reduce or eliminate frequent on/off fan switch that was typical in cooling towers. The control strategy reduced energy consumption by 38% within the simulations and result was validated by using the experimental data. The study of seasonal climate change impact on the cooling tower performance for reducing its power consumption by using VFD reduced up to 60% of annual power consumption in different weather conditions [125]. A novel counterflow cooling tower that utilizes a VFD fan, a by-pass loop equipped with microfilters and UV lights was proposed in a study [126]. UV lights kept the water at cleaner levels and reduced required blowdown water in the cooling tower. The results of the experiment indicated that at 50% capacity, the upgraded cooling tower had an energy savings of 76% with an overall water savings of 23% in comparison with the original design, a single-stage fan and no water filtration system[126].

2.6. Water

To advancing the technology of the cooling towers for reducing water consumption studies with different approaches have been completed. Some of the studies were focusing on advancing the structure of cooling tower, some aiming at operation parameters, and some finding more water. The methodologies for the investigations were in a wide range including experiment, modeling, simulation, and optimization, and modifications.

In the following paragraphs, the major studies have been classified and elaborated. Finally, the methods of water consumption reduction in wet cooling towers have been summarized.

2.6.1 Water Focus

The circulating water is the medium that carries the cooling and exchanges it for the heat in the buildings in a cooling cycle. The cooling load and the volume of circulating water needs to be adjusted regularly to maintain the cooling performance of the cooling cycle. The lost water due to evaporation in blowdown, and piping leakages must be compensated via make-up water in cooling towers [127]. Filtration of water is a widespread practice in the industry to soften the water to reduce the make-up water and protect equipment and pipes from fouling and corrosion. The excess water consumption and maintenance of equipment are costly for any energy/cooling plant. Also, governmental incentives are always an effective means for enticing customers to adopt water-saving methods. Generally, various options available for reducing make-up water within wet cooling towers are as follow [128]:

- *Soft water make-up:* removing calcium and magnesium, which are two main formers of scale. This reduces blowdown requirements while increases the pipe corrosion and the cost of producing make-up water,
- *Acid Feed:* increasing solubility of calcium and magnesium for allowing more cycles of concentration. Although the acid may present a safety concern or corrosion,
- *Reverse Osmosis (RO) Concentrate:* blending with potable water for make-up,
- *Rainwater:* collecting and using rainwater as make-up water, some anti-bacterial process may be required,
- *Air Handling Condensate:* replacing other equipment in large quantities of condensate,
- *Fixing Leaks:* repairing leaks always reduce water usage. They are wasteful and oftentimes can be overlooked if small, but their waste adds up over time.

A relatively new type of cooling tower known as a water-jet cooling tower was used for the experiment and numerical simulations study [129]. It was designed with a significant advantage over traditional wet cooling towers for contaminated water such as seawater or oil-water mixtures. In this study, the packing material was substituted with fine water droplets, made from special high-pressure nozzles [129]. The thermodynamical study results on heat rejection load, wet-bulb temperature, water to air ratio, tower spray zone height, droplet diameter, discharge droplet velocity, and air velocity in water-jet cooling tower showed that the droplet was diameter increased while exergy efficiency was decreased. By the increase of water to air ratio, the exergy efficiency for a smaller droplet size increased while the heat transfer decreased. The feasibility of using seawater instead of freshwater in circulating wet cooling tower in an experimental work showed the impact of saltwater concentrations on the cooling tower performance [103]. The study presented a valuable theoretical basis for developing seawater cycling instead of freshwater usage which causes to decrease 9.86% of the cooling tower performance [103]. In another study, the effect of thermophysical properties of seawater on the thermal performance of the cooling tower caused efficiency reduction of the cooling tower for 5-20% [130].

The study of water harvesting from cooling towers fog was conducted in an experiment by mounting woven metal meshes at the outlet plane of the cooling tower cell. The fog droplet capture efficiency was analyzed by using different mesh configurations. Varying the geometry of the net frame and changing the spacing between adjacent wires in the woven mesh led to the total water reduction by

40% in a 500 MW power plant in the optimal case [131].

The influence of the wettability of a mildly hydrophilic metal mesh for fog harvesting purposes in cooling towers was experimentally investigated on a small scale. The outcome indicated that from 8% to 23% of water content in the fog flow stream was collected [132].

Multi-walled carbon nanotubes (MWNTs) and nanoporous graphene nanoparticles within a cooling tower were investigated experimentally to understand the effects on energy and water conservation performance of a mechanical wet cooling tower [133]. The cooling tower thermal efficiency increased by dispersing nanoparticles in water which enhanced its water thermal conductivity. Therefore, the heat transfer increased through the cooling tower. For inlet water temperatures of 40 ˚C, the use of MWNTs and nano-porous graphene nanoparticles with a 0.1% mass concentration caused water reduction by 10% to 19%. The water saving was due to the increase of sensible heat effect and reduction in the latent heat effect. Reduction in the latent heat effect was due to the surface tension change of water as nanoparticles were created resistance against evaporation [133]. Another experimental study on the performance of cooling towers in six different beds by using nanofluid showed the metal reticular bed as the most suitable bed when using nanofluids because using ZnO/water nanofluid instead of pure water in the cooling tower improved the thermal efficiency of the tower up to 9.45% [134]. However, the economic analysis of this idea, particularly for largescale cooling towers, hazardous impacts of using nanofluids in the water, and the feasibility of performing in the real plant were neglected by the authors.

The effects of increasing the cycles of concentration (cycles of concentration refers to the concentration of dissolved solids in the cooling tower water) within a power plant cooling system for reducing water consumption were investigated by increasing the cycles of concentration in blowdown which led to overall water usage reduction [135]. The used water was flew through pretreatment, pH adjustments, chlorination control, and adding inhibitors for improving the quality of water for reusing. It was concluded that despite the treatments, phosphate deposits were observed at higher cycles of concentration. Corrosion and scale control using MBT and $ZnSO_4$ effectively were reduced corrosion in the carbon steel and brass pipes of the heat exchangers. Increasing the cycles of concentration from 6.5 to 9 resulted a water savings of 1.1×10^6 m^3 of water per year while cooling requirements were satisfied.

A lab-scale experimental study was conducted to recover evaporated water in the cooling tower by implementing an independent cold-water circulating loop through an array of copper tubes on both sides of the tower, led to the capture of 11% of vapor. Also, a combination of copper tubes assisted with metal foams improved the result [136].

2.6.2 Cooling Towers Modifications

To capture the vapor out of the cooling tower, a design was utilized for cooling sprays of water drawn by pumps from the bottom of the cooling tower to condense water vapor that flew under vacuum conditions through the vertical channel. The study was on a wet cooling tower which was equipped with an air to air heat exchanger to cool the hot and humid exiting air [20]. When the warm-wet air out of the tower was cooled through heat exchanger by an auxiliary fan more condensed water was collected through the cooling tower. 35.4% of evaporation was saved when the temperature difference between the warm humid air and the cooler ambient air was 15˚C. At a temperature difference of 3˚C, water savings was 15.1%.

TRNSYS (Transient Systems Simulation) was used for modeling, simulation, and evaluation of a cooling tower to study its performance and water loss in four control strategies and six different drift eliminators with two different water distribution systems. The control strategies were Fan Cycling Control (FCC), Multiple-speed fan motor control (MSC), Frequency-modulating control (FMC), and Optimum Control (OC) [9]. The two different distribution systems were the Pressure Water Distribution System (PWDS) and Gravity Water Distribution System (GWDS). FCC was a capacity control method on cooling towers that kept the system working until the thermostat sent a signal once the hotel's set-point temperature was reached. MSC strategy controlled outlet-water temperature- by adjusting the suitable fan rotational speed in three stages of velocity to meet the set-point value. FMC used VFD coupled with a standard fixed-pitch fan to control outlet-water-temperature by modifying the fan rotational speed in the various stages (more than three stages compared to MSC). This work also presented the new strategy called OC to find an optimum operating point for a coupled chiller and cooling tower. Three of the drift eliminators were composed of a zig-zag structure and fiberglass plates separated at distances of 55, 37, and 30 mm, respectively. The remaining drift eliminators utilized a plastic honeycomb structure, a 45˚ tilted rhomboid mesh, a 45˚ tilted lower half, and a 135˚ tilted upper half. The results concluded that the best FCC control strategy operation for reducing water loss was FMC, OC, and MSC following close behind. There were no substantial benefits between the controls due to most of the water loss occurring in blowdown (70%) while only 0.3% accounted for drifting. Also, the annual energy cost savings in the selected hotel in the southeast region of Spain was up to 3240 € by using the FMC control method. [9].

The heat transfer characteristics and pressure drop of a cooling tower were studied theoretically and experimentally [137]. The experiments were testing the changes in the inlet/outlet air flow rate and inlet temperature and flow rate of water. It was found that increasing the airflow rate into the cooling tower resulted in decreased exiting temperature and decreased outlet water temperature . However, the outlet air temperature approached a minimum amount as the airflow was increased, and the pressure drop across the cooling tower rapidly was increased as the airflow was increased in the tower [137].

A study of hybrid cooling towers concerning water conservation was conducted by numerical simulation and experiment on parallel and series configurations of a hybrid system in Tabriz Refinery (Tabriz, Iran) [41]. Based on the heat exchanger area of the refinery, the water consumption reduction was between 37% to 23% during the summer months for the series and parallel configuration, respectively. Additionally, the payback period for recovering the costs of the upgrades was around seven years for series configuration, while the parallel configuration did not return the initial investment. The impact of heat-exchangers arrangement and hybrid ratio on water loss in different seasons at Tabriz refinery plant is shown in Figure 2.6 [41]. The method of retrofitting an existing wet cooling tower into a hybrid cooling tower with PPWD configuration to reduce make-up water was developed in unit five of the Isfahan (Isfahan, Iran) thermal power plant [39]. The goal was to modify the system with minimal effort from the plant personnel labor and cost. The first hybrid used the existing wet towers packing in the same place, while the second one lowered the packing by 1.4 m to increase the dry section cooling area. Under identical scenarios, the first hybrid setup lowered water consumption by 7.0 % while the second setup, with the larger fraction of total cooling, lowered water consumption by 9.4% [39].

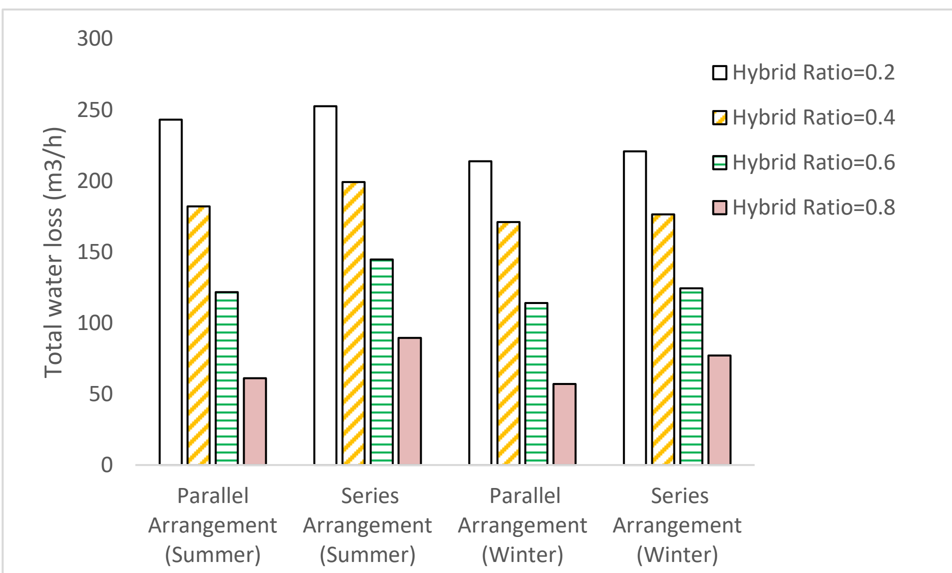

Figure 2. 6: The impact of heat-exchangers arrangement and hybrid ratio on water loss in different seasons at Tabriz refinery plant

2.6.3 *Optimization Approach*

Optimization as a tool for finding the optimum parameters for the ideal performance has been applied in some cooling tower studies. It should be noted that the optimization of cooling towers and related systems by using air volumetric flow and cooling water flowrate as parameters have three main shortcomings:

- Optimization models often are assumed that process parameters can be modified at any time. Adjustments in process parameters of existing cooling towers require input from operators and equipment manufacturers.
- Fluctuations in environmental conditions are often disregarded.
- There is a lack of systematic identification and assessment of efficiency measures [10].

A dynamic model was developed to determine the impact of air temperature and humidity ratio in a power plant's cooling tower performance. The 24-hours information of the ambient weather condition, cooling tower water temperature, and its water mass flow rate was collected to develop a dynamic model. The model was developed based on Poppe model for unsaturated and supersaturated air conditions [138]. Three objective functions were defined for minimizing operating cost, minimizing the accumulation of water in the tank, and minimizing makeup water flowrate. The results indicated that the makeup water flow rate could be reduced up to 57% while the costs and energy usage were at minimum possible value [138]. Optimization of a cooling tower in 90 kW pilot power plant for minimal operating cost by considering fan's speed, make-up water mass flow rate, and valve positions were conducted [11]. It was observed that the outlet temperature of the cooling tower had to keep as high as the heat load to minimize the cost of system. The most economically minimizing action was to increase the cooling tower water mass flow rate when the load demand was increased and then increase the fan speed by an increase of cooling demand. The reduction in water and energy consumption in the cooling tower was estimated to save $0.05 per hour in an optimized pilot plan. The worst scenario was using additional makeup water in the cooling tower which led to a $0.68 per hour increase in operating cost compared with the non-optimal scenario [11].

The mathematical model for designing a sustainable natural draft wet cooling tower was developed to estimate water consumption as a function of the power plant location in Spain [139]. The objective functions were defined to find the optimum water consumption, tower size, and cost as a function of humidity, temperature, and atmospheric pressure for different power plant sizes ranged from 40 to 450 MW. The study has compared the effect of different weather conditions in designing natural draft of wet cooling tower to reduce water consumption in power plants. The outcomes showed a reasonable agreement between the calculated results and the practical ones, plus the higher level of temperature and humidity led to higher water consumption in wet cooling towers. A power plant in the southeast region of Spain with higher temperature and humidity range consumed around 2 L/kWh water, while a similar plant in the northwest of Spain with lower temperature and humidity used about 1.5 L/kWh water at the same time [139].

The optimization of a cooling water system using mixed-integer nonlinear programming (MINLP) model was performed to reduce water consumption, improve the system operation, and reduce the capital cost of a cooling network using series, parallel, as well as combined series and parallel configurations [139]. When their MINLP optimization algorithm was employed, the combined series and parallel configuration had minimized the cost [139] .

A cooling tower using Advanced Pinch Design (APD) to allow for maximum amounts of heat transfer to occur within a system optimized. The APD algorithm let interaction between the cooling tower performance and heat-exchanger network configuration to be considered simultaneously. The objective function was to minimize the cooling tower annual cost which was depending on water consumption, pump power consumption, cooling tower approach temperature, and the ambient wet bulb temperature. In the traditional dry cooling system, parallel configuration was used in heat-exchangers network. However, APD suggested heat exchangers network for the minimum energy and water consumption by maximizing water re-use in the cooling tower. Figure 2.7 shows the parallel, KSD method, and APD method configuration of the heat exchangers network. APD arrangement was determined as an optimal heat exchanger network in this study because APD provided the minimum annual cost for the cooling tower compared with parallel arrangement an KSD method arrangement. Optimal heat-exchanger arrangement was achieved through an advanced synthesis algorithm using pinch point temperature in heat-exchangers network with four units. The synthesis algorithm adjusts water inlet temperature, water outlet temperature, and pinch point temperature to minimize the cooling tower total annual cost as well as its water consumption. In comparing APD to another optimization algorithm known as the Kim and Smith (KSD) method costs were decreased from $72,000 per year and $59,000 per year for the conventional and KSD systems respectively, to $52,000 for the APD system. The APD algorithm was then modified to optimize for minimal make-up water, which resulted in costs of $39,000 per year while 46% of make-up water was saved [140].

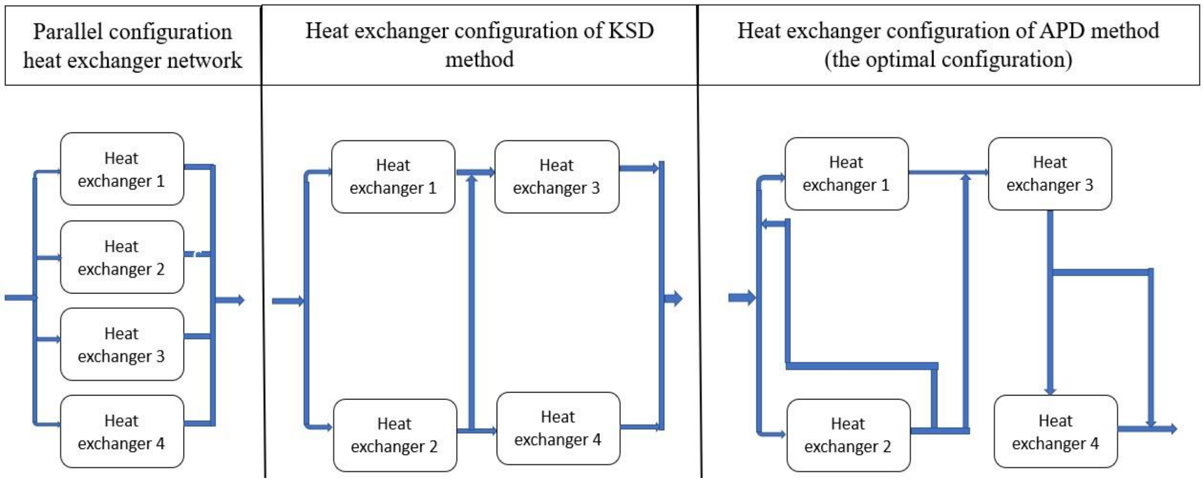

Figure 2. 7: Different heat exchangers network configurations

The optimizing heat transfer in a cooling tower by studying the water distribution system was conducted by measuring the water inlet velocity and temperature fields by using a mobile robot equipped with high-quality sensors [142]. The ambient air velocity as well as the air temperature and density in the vicinity of the cooling tower were measured. It was concluded that the optimal water/air mass flow ratio should be small and uniform across the entire area of the cooling tower. By keeping the water/air mass flow ratio constant, entropy generation, and thus exergy destruction, was minimized which resulted in lower outlet water temperature and greater efficiency of the cooling tower [142]. Multi-objective optimization and experimental evaluation of a forced draft cooling tower using different fill options have been conducted [100]. The fills were wooden splash, wire mesh, and honeycomb. Objective functions for tower range (temperature difference between inlet water and outlet water), Merkel number, effectiveness, and evaporation rate were formulated using experimental data and then simultaneously optimized using genetic algorithm and different water and air flow rates. The most optimal combination of mass flow rates of water and air with wire mesh packing type provided a 5.8% increase in the effectiveness of the cooling tower as well as a 18.4% reduction in water consumption [100].

2.7. Research Trend

Going through a comprehensive literature reviewing on cooling towers with the focus on water sustainability in the present study suggests the opportunity for mapping the completed studies. The authors present the conclusion of their study in the form of the research trends to facilitate the future study plans for advancing water conservations in cooling towers. Figure 2.8 illustrates the map of the research trend in advancing sustainability in wet cooling towers.

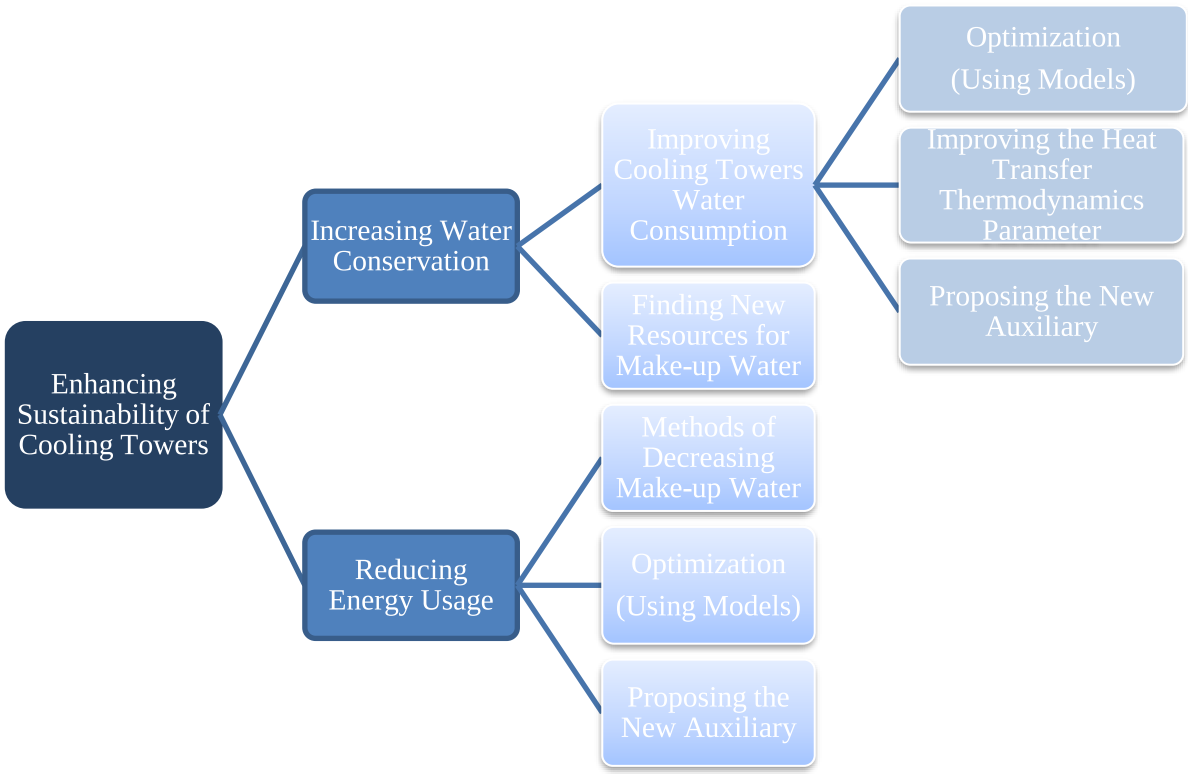

Figure 2. 8: Research map for enhancing sustainability of cooling towers

Finally, the proven effort for decreasing water usage in cooling towers from the past studies can be concluded as:

- Deploying cooling loops by reuse or recycle water [22],
- Changing wet cooling tower to hybrid or dry cooling tower [21],
- Adding the dry cooling section to current wet cooling towers [22],
- Using air to air heat exchanger on the cooling tower [21],
- Applying the airflow control method by taking advantage of fan VFD to reduce the water and energy consumption of cooling towers [22],
- Installing a RO water filtration system or other water filtering systems to reduce required blowdown [10],
- Reduction in blowdown with increasing of concentration cycles [21],
- Decreasing pump power demand by installing a new pump motor for improved efficiency [10],
- Reduction of the number of backwashing cycles [10],

- Changing the fan control mode to variable speed with parallel fan combination [10].

2.8. Conclusion

Considering the dependency of life to water and its scarcity reveals the urgency of studying on reducing water consumption in the cooling towers since cooling towers are the gate for the invisible dissipation of water. The present study elaborated on the roles of cooling towers in cooling cycles while narrated the history of energy-water modeling methods of cooling towers through the Markel, the Poppe, and the Effectiveness–(NTU) Models since water and energy are deeply connected in cooling towers. The major classifications and configurations of cooling towers showed the wet cooling towers consume the highest water while dry cooling towers use the minimum water in a trade-off with performance. The hybrid cooling towers have the advantages of both while they are more costly. The past challenges for boosting water conservation in cooling towers were allocated in focusing on water supply sources or modifications of the cooling tower. There are some practical approaches to reduce the environmental impact of cooling towers such as using VFD, adding a dry section, and employing water filtration to reduce its water and energy consumption. Finally, the trend of future studies presents the potential investigations for water and energy reduction opportunities in the light of water-energy nexus. Proposing the new auxiliary equipment, finding methods of decreasing make-up water, and optimization of the energy models are the major path for future investigation to reduce energy consumption in cooling towers. Similarly, optimizing cooling towers water consumption, improving the heat transfer parameters in cooling towers, proposing the new auxiliary equipment, and findings new make-up water resources are future paths for boosting water conservations in cooling towers.

Credit authorship contribution statement:

Saeed Ghoddousi: Conceptualization, Methodology, Investigation, Formal Analysis, Resources, Data Curation, Writing, Visualization. Austin Anderson: Resources, Investigation, Writing. Behnaz Rezaie: Conceptualization, Methodology, Validation, Formal Analysis, Writing, Review & Editing, Supervision, Project Administration.

Chapter 3: Sustainable Modeling for Improving Water-Energy Conservation in Cooling Towers (Energy-Water Nexus)

Saeed Ghoddousi and Behnaz Rezaie

Forthcoming in Entropy

3.1 Abstract

The role of cooling towers in dissipating heat from the hot stream in the cooling process. In wet cooling towers water gets evaporated for releasing heat from the hot stream, thus water consumption is high and so is energy consumption during the operation. The purpose of the present study is finding methods to reduce the water consumption and energy is the wet cooling towers. Since a wet cooling tower uses the ambient air to cool down the hot water, its performance strongly depends on the ambient air conditions like temperature and humidity. The impact of the ambient air conditions has not been studied seriously. In the present study a model for wet cooling towers is developed by considering the properties of the ambient air. Then the model is simulated and analyzed for various operation conditions. The outcomes of the simulation are being compared with the actual data for validation. Also, an improvement option is suggested for reducing the cooling tower water and energy consumption. By adjusting the operation water outlet temperature in the cooling tower, fans' power consumption was reduced as well as water loss due to evaporation. As results, the monthly electricity and water consumption of the cooling tower were reduced 40% and 17.5%, respectively. In addition, an improved operation of the cooling tower eliminated 18377.02 kg CO2 emissions in electricity generation and provides an annual cost saving of $4,500.

3.2 Introduction

The wet cooling tower is the equipment which removes heat from the hot stream of the cooling process by evaporating water [143]. The evaporation in the thermal power plants cooling towers in the United States of America (U.S.A.) alone led to consuming millions of fresh water daily in 2003 [144]. Thermoelectric generation and its required cooling are responsible for approximately 10% of the total water demand in the world [145]. Also, wet cooling towers are widely used in Heating Ventilation and Air Conditioning (HVAC) systems to reduce the temperature of circulated condenser water [146].

The thermal performance of wet cooling towers strongly depends on the humidity and temperature of the ambient air [61]. The study of the impact of weather conditions on the performance of the wet cooling tower has shown that even a slight increase in temperature relative to the design temperature of the cooling tower drastically reduces the cooling tower performance [66]. A developed mathematical model of a cooling tower to assess the tower performance for different temperate has illustrated that decreasing the tower water flowrate was a more effective approach than increasing the tower air flowrate to enhance the cooling tower performance [95].

A recent literature study presented various methods of enhancing the cooling towers performance with respect to water and energy conservation [143]. That study showed the practical (and instant) approached of operation improvements for the industry, as well as more sophisticated theoretical methods like various modeling, modifications, and optimizations of the design parameters. Advancing the performance of cooling towers has been investigated in several studies to find the optimum parameters using optimization approaches. A 24-hours dynamic model was developed based on Poope model for performance investigation of a power plant's cooling tower in different ambient air conditions. The dynamic correlations for air temperature and humidity ratio were developed by using actual data to simulate the dynamic behavior of the cooling tower. By using the developed model 57% of make-up water was reduced in cooling tower [138]. The real-time optimization in a forced draft cooling tower by employing an artificial neural network in controlling fan speed has showed a 6.7% annual energy saving by the optimized cooling tower [147]. The multi-cell induced-draft cooling tower's fans speed were optimized based on outlet water temperature. In the optimized cooling tower, the performance of the cooling tower's fans was improved by changing their rotation speed while achieving the same cooled water temperature (real data).

The cooling tower operating conditions was defined by the Braun model as objective function for finding the optimal operation criteria of the cooling tower [111]. Cooling tower fan speed modeling has been done in the selected case study resulted in 15.6% (17879 kW) savings in the actual power consumption for the 68 sampled days. A forced draft cooling tower was modeled and optimized with different types of heat exchangers (evaporators) [148]. The water evaporation rate was optimized by maximizing the cooling tower temperature range and effectiveness. Results led to an 18.4% enhancement in cooling tower water consumption.

Replacing the Variable Frequency Drive (VFD) fans instead of two-speed fans (low and high speed) was investigated to improve energy consumption in a cooling tower [124] Selected cooling tower had six fans that provided 4000 m^3/h cold water for a tandem cold mill and continuous annealing line. Four of these fans were replaced with VFD fans to control the speed for reducing water and energy consumption. The cooling tower outlet water temperature was controlled to maintain the set point at the highest allowable temperature based on cooling needs. The results showed that the cooling tower energy consumption was reduced by 38% (about 506.4 kWh of daily energy usage was reduced) at the highest allowable setpoint temperature (32.5 °C) [124].

An improvement in the cooling tower performance was investigated by reducing fans' operational costs in four major Brazilian cities with different average daily wet and dry bulb temperatures [125]. The cooling tower model was developed using Merkel model. The outlet water temperature was modeled as a function of inlet air mass flow rate and ambient air condition. Adjusting cooling tower fans speed using VFD resulted in up to 35.95% reduction in the cooling tower fans operational cost. Also, it was shown that using more VFD fans led to achieving more energy and cost savings in different weather conditions. Also results reveal that the cooling towers which were located in cities with the higher yearly ambient air temperature variations have a great potential to save energy by using VFD and reducing the fan speed in colder days [125]. Finding an optimal outlet water temperature set point in different ambient air condition in the cooling tower was studied to reduce the its energy consumption [149]. The developed framework used cooling load and wet bulb temperature as input to find optimal cooling tower outlet water temperature in different air conditions. The optimal cooling tower performance was found based on one-year historical data from both cooling system and ambient air condition. Results revealed that by finding an optimal water outlet temperature setpoint based on cooling needs and weather conditions, around 9.67% annual energy saving would be achieved.

Modeling of a wet cooling tower by considering the ambient air properties and using the real-time data for purpose of reducing water and energy consumption is the novelty of this study. The modeling and improvement options in the presented study provides guidance for researchers and engineers to enhance cooling towers performance toward advancing the water and energy conservation, cost savings over the reduction of utility consumption, and environmental protection against pollution for electricity generations.

3.3 Methodology

3.3.1. Cooling Tower Modeling

There are several modeling approaches for cooling tower such as Merkel and Poppe’s models. Because the Merkel model neglects the water loss due to evaporation through the cooling towers, and the present study focuses on cooling towers water consumption, the Merkel model cannot be employed in the present work. Although the Poppe model is accurate in predicting outlet air condition in a wet cooling tower [66], [68] this model needs several measured data from a cooling tower and its ambient condition, which are not available in the selected cooling tower. Also, this model is more complex and has computationally laborious tasks requiring help from supercomputers [150]. Consequently, Poppe’s model is not also preferable model in the present study. Braun's model predicts exiting air conditions and estimates the water loss due to evaporation in the cooling tower [111]. Braun model neglects heat transfer from the walls, assumes steady-state energy balance condition, and considers constant airflow rate. Therefore, the Braun model has been selected for modeling the cooling tower performance. Braun's model was developed based on a steady-state situation and transient calculations were not being used in this model. Braun's model depends on Number of Transfer Unit (NTU) value in cooling towers. NTU value in the cooling tower could be varied based on the performance characteristic of the cooling tower such as filler heat transfer capability, ambient air conditions, and water/air inlet and outlet conditions [58]. The NTU is not given in tower specifications and is normally unknown for existing cooling towers [111]. The following equations show the main calculation steps to determine NTU for the Braun model:

C_s denotes saturation specific heat as shown in Eq. (3.1). Based on C_s definition, the enthalpy and temperature differences have linear relation. The water effectiveness is derived in terms of C_s using an average slope between inlet and outlet water conditions.

$$C_s = \left(\frac{dh_s}{dT}\right)_{T=T_w} = \frac{h_{s,w,i} - h_{s,w,o}}{T_{w,i} - T_{w,o}} \tag{3.1}$$

Here, C_s is saturation specific heat (J/kg. ℃). $h_{s,w,i}$ and $h_{s,w,o}$ are water saturation enthalpy at the inlet and outlet of the cooling tower, respectively. $T_{w,i}$ and $T_{w,o}$ are water inlet and outlet temperature in ℃ , respectively.

The ratio of the actual heat transfer to the maximum possible air-side heat transfer if the exiting air stream were saturated at the incoming water temperature is defined by ε_a which is calculated using Eq. (3.2) and (3.3).

$$N = \frac{\dot{m}_a C_s}{\dot{m}_{w,i} C_{pw}} \tag{3.2}$$

$$\varepsilon_a = \frac{1 - \mathrm{e}^{(-NTU(1-N))}}{1 - N\mathrm{e}^{(-NTU(1-N))}} \tag{3.3}$$

C_{pw} is water specific heat at constant pressure and in J/kg K. Also, $\dot{m}_{w,i}$and $\dot{m}_a$ are inlet water mass flowrate and air mass flowrate in kg/s, respectively. NTU is number of transfer units. The exiting air enthalpy is determined from overall energy balances on the flow streams.

$$h_{a,o} = h_{a,i} + \varepsilon_a\left(h_{s,w,i} - h_{a,i}\right) \tag{3.4}$$

Where, $h_{a,o}$ and $h_{a,i}$ are air outlet air enthalpy and inlet air enthalpy in J/kg, respectively.

An effective saturation enthalpy value is calculated by integrating Eq. (3.5) for constant $h_{s,w,e}$ as follows:

$$h_{s,w,e} = h_{a,i} + \frac{\left(h_{a,o} - h_{a,i}\right)}{1 - \mathrm{e}^{(-NTU)}} \tag{3.5}$$

The exiting humidity ratio is calculated by integrating over the tower volume as shown in Eq. (3.6). Also, Eq. (3.7) calculates the outlet water mass flowrate which is derived from the overall mass balance.

$$\omega_{a,o} = \omega_{s,w,e} + (\omega_{a,i} - \omega_{s,w,e})e^{-NTU} \tag{3.6}$$

$$\dot{m}_{w,o} = \dot{m}_{w,i} - \dot{m}_a(\omega_{a,o} - \omega_{a,i}) \tag{3.7}$$

Here, $\omega_{s,w,e}$, $\omega_{a,o}$, is effective saturation humidity ratio value and air humidity ratio ($\frac{kg\ water\ vapor}{kg\ dry\ air}$), respectively.

The water outlet temperature is determined from overall energy balances on the flow streams as shown in Eq. (3.8).

$$T_{w,o} = \frac{\dot{m}_{w,i}\left(T_{w,i} - T_{ref}\right)C_{pw} - \dot{m}_a(h_{a,o} - h_{a,i})}{\dot{m}_{w,o}C_{pw}} \tag{3.8}$$

the T_{ref} the reference temperature for zero enthalpy of liquid water (℃)

The loss water due to evaporation which is equal to the make-up water is calculated with Eq. (3.9) as follows:

$$\dot{m}_{Loss} = \dot{m}_{w,i} - \dot{m}_{w,o} = \dot{m}_a(\omega_{a,o} - \omega_{a,i}) \tag{3.9}$$

Water savings can be calculated based on improved model water loss and reference model water loss as shown in Eq. (3.10) [39], [151].

$$Water\ saving = \sum(\dot{m}_{Loss,ref} - \dot{m}_{Loss}) \times (operation\ time) \tag{3.10}$$

Here, $\dot{m}_{Loss,ref}$ is cooling tower’s water loss in the 100% energy load (kg/s). Water saving is amount of water that has been saved by reducing air inlet volumetric flowrate (kg). Operation time is every time interval of performing improvement approach(s).

For removing heat from the circulating water in cooling towers there is an electricity demand for running the inside fan(s) to provide the cool air flow. The saturated outlet air leaves the cooling tower with evaporated water. The evaporated water is replaced by the make-up water in the cooling tower outlet water. Consequently, cooling towers' operation greatly demand electricity (energy) and water. The forced inlet air volumetric flowrate to the cooling tower determines the cooling tower fan energy consumption which is shown in Eq. (3.11) and Eq. (3.12) [39], [151]. Also, the energy saving that is captured via adjusting the air inlet volumetric flow rate is determined using Eq. (3.13) [39], [151].

$$\dot{V}_a = \frac{\dot{m}_a}{\rho_a} \tag{3.11}$$

$$P_{fan} = P_{fan,ref}(\frac{\dot{V}_a}{\dot{V}_{a,ref}})^3 \tag{3.12}$$

$$Energy\ saving = \sum(P_{fan,ref} - P_{fan}) \times (operation\ time) \tag{3.13}$$

Here, ρ_a is air density at the ambient air temperature ($\frac{kg}{m^3}$). $\dot{V}_a$ is inlet air volumetric flow rate ($\frac{m^3}{s}$), and $\dot{V}_{a,ref}$ is inlet air volumetric flow rate on the 100% energy load ($\frac{m^3}{s}$). Also, P_{fan} is cooling tower's fan power demand (kW), and $P_{fan,ref}$ is cooling tower's fan power demand in the 100% energy load (kW). $Energy\ saving$ is the energy that has been saved by reducing air inlet volumetric flowrate (kWh).

Also, cooling tower range is defined as the difference in inlet water temperature and outlet water temperature and can be written as:

$$R = T_{w,i} - T_{w,o} \tag{3.14}$$

3.3.2. Modeling software

Engineering Equation Solver (EES) is a software for thermodynamic modeling that has thermodynamic properties for water and air as well as psychometric chart data. Thus, it has been used for modeling and simulation in this study.

3.4 Case Study: University of Idaho Energy Plant

The cooling district energy system in the University of Idaho (UI) main campus in Moscow, Idaho, U.S.A. (46.7288° N, 117.0126° W) which supplies cooling to a total of 61 buildings on campus [152].. The absorption chiller utilizes steam derived from burning wood chips to produce chilled water for cooling purposes in hot months [153]. Six mechanical draft counter flow cooling towers are used to remove heat from the absorption chiller. One of these cooling towers with 680 tons of capacity (1 ton represents the capability to cool 0.68 m^3/h of water from a 35 °C entering water temperature to a 30 °C leaving water temperature at a 25.5 °C entering wet-bulb temperature) is selected as a case study in the present work, as shown in Figure 3.1. The cooling tower has two VFD fans with power of 29.82 kW for each to provide a 78 m^3/s airflow rate at 100% load.

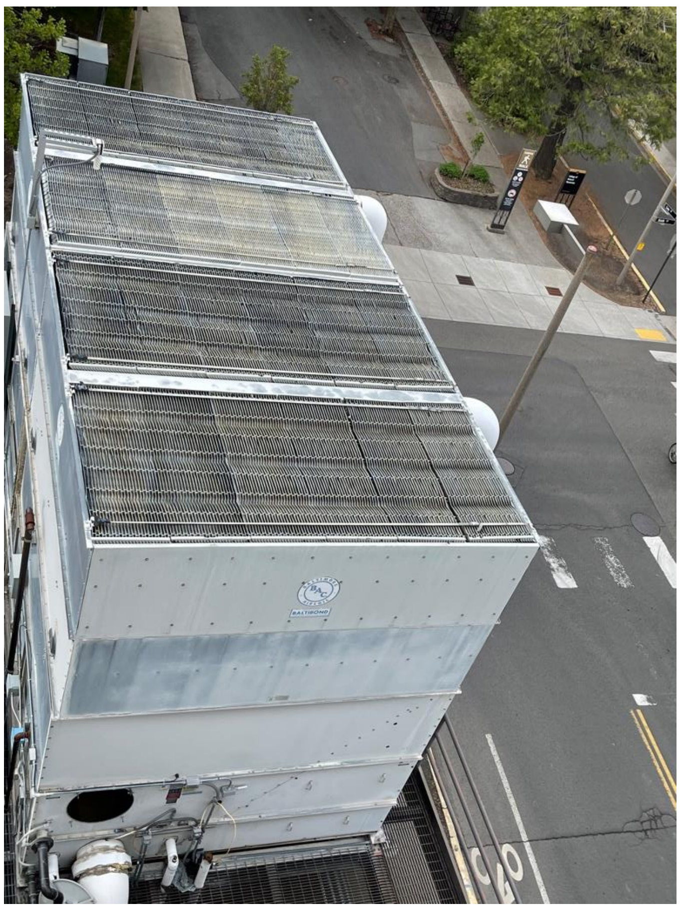

Figure 3. 1: Selected cooling tower located at UI campus at Moscow, ID, USA

Seven sensors including flow rate, humidity, and temperature sensors have been already installed on the selected cooling tower to provide the real-time data for the present study. The air sensors were measuring both temperature and relative humidity with accuracies of +/- 0.2 ˚C and +/- 2%, respectively. The water temperature sensors were measured the inlet and outlet were Resistive Temperature Detectors (RTD) with an accuracy of +/ 0.26 ˚C. The Data Acquisition Device (DAQ) and sensors were designed and installed in the earlier phase of the cooling tower project as real-time data acquisition.

Since NTU value cannot be obtained from cooling tower specification and it depends on ambient air condition, different ambient air temperature needs to be considered. 53 different ambient air temperatures ranged from 4.5 °C to 33.5 °C and their related data ($T_{w,i}$, $T_{w,o}$, $RH_{a,i}$, $\dot{m}_a$, $\dot{m}_{w,i}$, P) are selected regardless of the time that they collected as shows in Table 3.1.

Table 3. 1: Input data to find average cooling tower NTU value

	$T_{a,i}$ °C	$RH_{a,i}$ %	$\dot{m}_a$ kg/s	$\dot{m}_{w,i}$ kg/s	$T_{w,i}$ °C	$T_{w,o}$ °C	P atm
1	4.4	79	98.9	65.8	37.8	31.4	1
2	5.0	73	98.7	65.8	39.9	32.7	1
3	5.6	73	98.5	65.8	39.9	32.7	1
4	6.1	74	98.3	65.8	40.3	33.2	1
5	6.7	74	98.1	65.8	40.9	33.7	1
6	7.2	68	98.0	65.8	39.5	32.3	1
7	7.8	73	97.7	65.8	37.8	31.4	1
8	8.3	77	97.5	65.8	37.3	31.0	1
9	8.9	66	97.1	65.8	36.9	31.0	1
10	9.4	64	97.2	65.8	39.5	32.7	1
11	10.0	57	97.0	65.8	40.8	33.6	1
12	10.6	80	96.6	65.8	41.6	34.8	1
13	11.1	61	96.6	65.8	41.6	35.3	1
14	11.7	74	96.3	65.8	37.8	31.4	1
15	12.2	86	96.0	65.8	42.1	34.9	1
16	12.8	72	95.9	65.8	36.5	31.8	1
17	13.3	55	95.8	65.8	42.5	35.7	1
18	13.9	67	95.5	65.8	38.2	32.3	1
19	14.4	70	95.3	65.8	41.6	34.9	1
20	15.0	51	95.2	65.8	41.6	34.4	1
21	15.6	47	95.1	65.8	38.6	32.3	1
22	16.1	46	94.9	65.8	42.9	36.2	1
23	16.7	48	94.7	65.8	41.9	34.0	1
24	17.2	46	94.5	65.8	42.1	35.7	1
25	17.8	50	94.2	65.8	39.5	33.6	1
26	18.3	47	94.2	65.8	41.6	34.4	1
27	18.9	45	93.9	65.8	41.6	34.4	1
28	19.4	51	93.7	65.8	41.2	34.9	1
29	20.0	40	93.5	65.8	42.5	35.7	1

30	20.6	44	93.3	65.8	42.1	35.7	1
31	21.1	38	93.2	65.8	42.1	35.7	1
32	21.7	38	92.8	65.8	42.1	35.7	1
33	22.2	34	92.9	65.8	42.5	35.7	1
34	22.8	34	92.6	65.8	41.6	35.3	1
35	23.3	28	92.6	65.8	42.5	35.7	1
36	23.9	33	92.4	65.8	42.5	35.7	1
37	24.4	14	92.4	65.8	42.5	35.7	1
38	25.0	25	92.0	65.8	40.8	34.4	1
39	25.6	23	91.9	65.8	42.1	35.3	1
40	26.1	26	91.8	65.8	42.5	35.7	1
41	26.7	21	91.6	65.8	42.5	35.7	1
42	27.2	21	93.0	65.8	42.5	35.7	1
43	27.8	22	91.2	65.8	42.9	36.6	1
44	28.3	22	91.0	65.8	42.9	36.2	1
45	28.9	29	90.7	65.8	42.1	36.6	1
46	29.4	25	90.6	65.8	41.6	35.7	1
47	30.0	19	90.5	65.8	43.4	36.6	1
48	30.6	26	90.2	65.8	44.3	37.9	1
49	31.1	22	90.2	65.8	43.4	37.5	1
50	31.7	22	89.9	65.8	44.7	37.5	1
51	32.2	22	89.8	65.8	43.8	37.5	1
52	32.8	15	89.7	65.8	42.9	36.2	1
53	33.3	25	89.7	65.8	45.1	38.3	1

Table 3.2 includes the inlet and outlet water mass flow rate, inlet water temperature, inlet air temperature, inlet air Relative Humidity (RH), ambient air pressure, and inlet air volumetric flowrate data were collected for one month period from 1 am 14 September 2020 to 12 pm 13 October 2020 (720 h). Other technical and economic input such as number of the cooling tower's fan, fan's power, electricity cost, and water cost data are summarized in Table 3.2 with their references. Table 3.2 data are used for actual cooling tower model, improvement study to modify outlet water temperature, and validation.

Table 3. 2: The technical and economic data of the modeled cooling tower

Inlet and outlet water mass flow rate	65.8 kg/s
Inlet water temperature	Varies based on cooling load 28.9-43.9 ˚C
Inlet air temperature	The ambient air temperature at each time interval 4.5-32.7 ˚C
Inlet air Relative Humidity (RH)	The ambient air RH at each time interval 15-86%
Ambient air pressure	101.3 kPa
Inlet air volumetric flowrate (100% load)	78 m^3/s
Number of the cooling tower's fan	2
Fan's power	29.82 kW(each)
Electricity cost	0.09 $/kWh [154]
Water cost	0.65 $\$/m^3$[11]

3.5 Modeling the Cooling Tower

Since the present study's focus on water and energy consumption, the model which can predict the cooling tower water and energy loss is Braun's model. The performance parameters and cooling tower outlet air/water conditions are calculated in the developed model. First, NTU value which is critical in developing Braun's model must be calculated. Following approach are done to calculate the NTU value.

3.5.1. Effectiveness-NTU analysis

The NTU needs to be determined to model a cooling tower and to predict outlet water/air condition. The NTU calculation has been done based on its water outlet temperature by employing Eq. (3.1) - (3.8) through the shown sequential in Figure 3.2. The water inlet and outlet temperatures, air inlet temperature, and relative humidity, and mass flow rate of the inlet air and water are used as input of modeling as shown in Figure 3.2. The outcome of the developed model is the water outlet temperature based on the assumed NTU value. If the difference between simulated water outlet temperature and actual water outlet temperature is less than 0.01 °C, the assumed NTU value is correct, and the developed cooling tower model is reliable.

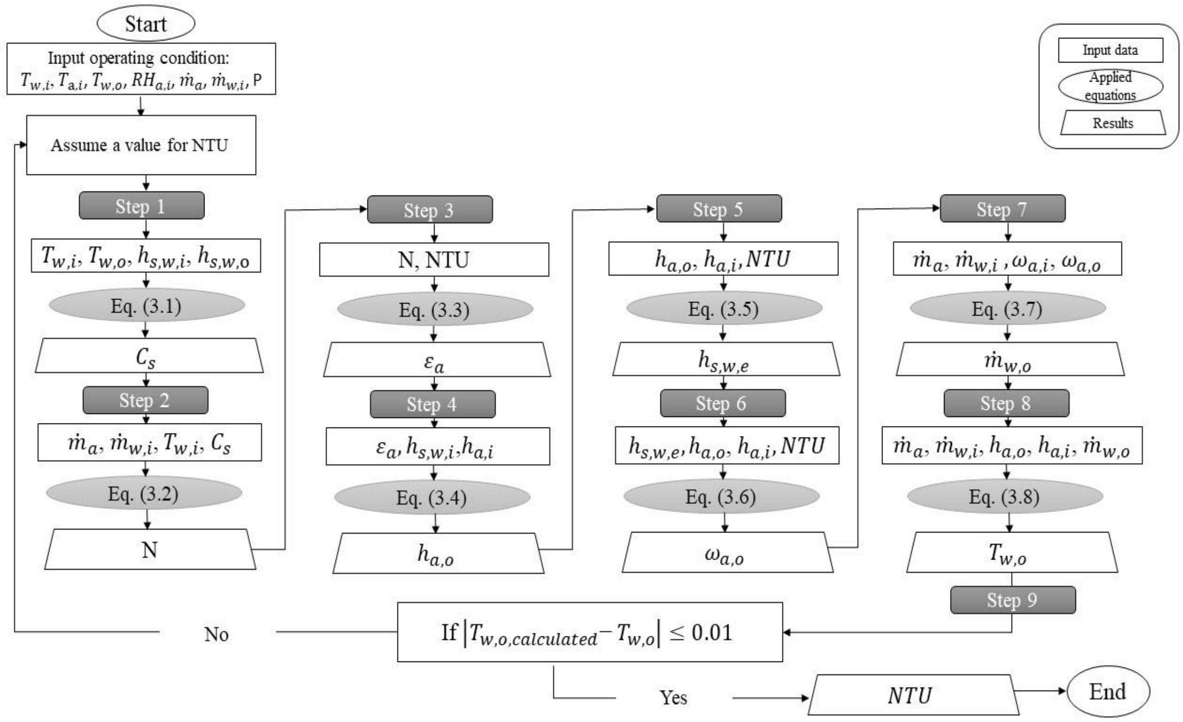

Figure 3. 2: Flowchart of using the developed model for finding NTU value

3.6 Analysis

3.6.1. Effectiveness-NTU analysis

To accurately model the cooling tower based on developed model, it is necessary to find the cooling tower NTU value. Then, further modification (such as modifying outlet water temperature and inlet mass flow rate) are applicable based on calculated NTU value. Therefore, the analysis in the current study has three parts. In the first part, study calculates NTU value based on input data ($T_{w,i}$, $T_{a,i}$, $T_{w,o}$, $RH_{a,i}$, $\dot{m}_a$, $\dot{m}_{w,i}$, P). Collected input data from Table 3.1 are applied to Eq. (3.1)- (3.8) as shows in Figure 3.2. 53 NTU values are calculated in every inlet ambient air temperature. The average of 53 calculated NTU values is assumed as the cooling tower NTU value in the present modeling study.

3.6.2. Actual cooling tower model analysis

In the second part, the study applies calculated average NTU value and other input data ($T_{w,i}$, $T_{a,i}$, $RH_{a,i}$, $\dot{m}_{w,i}$, P) from Table 3.2 (one week data from 1 am 14 September 2020 to 12 pm 20 September 2020) to Eq. (3.1)-(3.8) based on first 9 steps described in Figure 3.3 to find outlet water temperature as well as other cooling tower performance parameters.

3.6.3. Cooling tower improvement analysis

A model is developing based on the ambient air conditions and average calculated NTU value to reduce energy and water consumptions in the cooling tower. The cooling tower of the case study is in operation during the hot season (June to mid-October) at 100% load to meet the cooling demand. The average outlet cold water temperature is about 35 °C, covering the cooling demand. The outlet water temperature varies based on inlet water temperature and ambient air conditions. Since the fan worked in 100% load without considering ambient air condition and inlet water temperature, the outlet water temperature could be less than 35 °C which was not required. By addressing the issue of going lower than 35 °C, the excess energy consumption for unnecessary cooling would be eliminated from the cooling tower operation. In the third part of analysis, a thermodynamic model sensitive to the water outlet temperature is needed Eq. (3.1) -(3.12) have used in the illustrated sequences in Figure 3.3 for modeling of the cooling tower without using the excess water and energy for operation. All data from Table 3.2 (one month period from 1 am 14 September 2020 to 12 pm 13 October 2020) have been applied to the related equations based on Figure 3.3 to find energy and water savings. The model outputs are the possible water and energy saving by the cooling tower. The developed model is using for adjusting the inlet air volumetric flow rate to keep the outlet water temperature at 35°C.

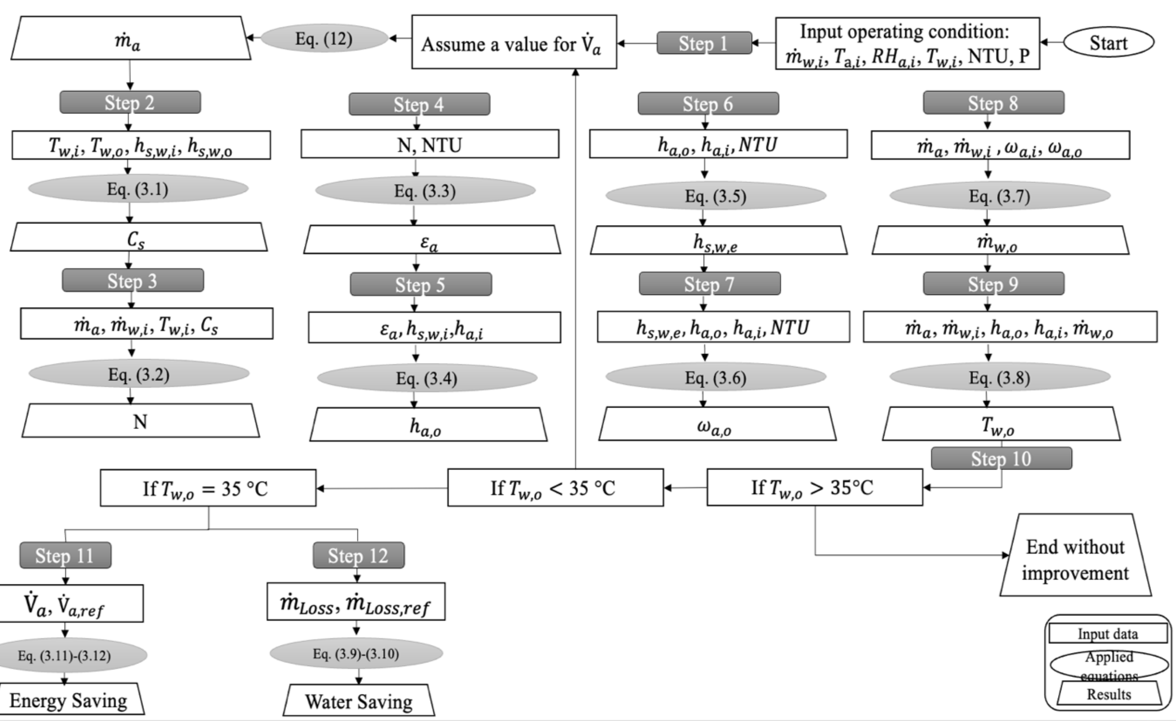

Figure 3. 3: Flowchart of using the developed model for improving the cooling tower water and energy consumption

3.7 Results and Discussion

The purpose of the study was improving the cooling tower performance of UI energy plant by reducing its water and energy consumption. In the first step, the cooling tower modeling was performed to determine its outlet water temperature based on ambient air and water inlet conditions using EES. The NTU value which is key parameter in modeling cooling tower was calculated. Then, NTU value and other input data from one-month cooling tower operation were used to modify outlet water temperature and to improve performance of the selected cooling tower. The results of all analysis are presented in the following subsections.

3.7.1. Effectiveness-NTU results

The outcome of Eq. (3.1)- (3.8) based on Figure 3.2 and Table 3.1 data have provided different NTU values in various ambient air temperatures which is shown in Figure 3.4. The average of calculated NTU values in different ambient temperatures is 0.188 which is considered as the cooling tower NTU in this study.

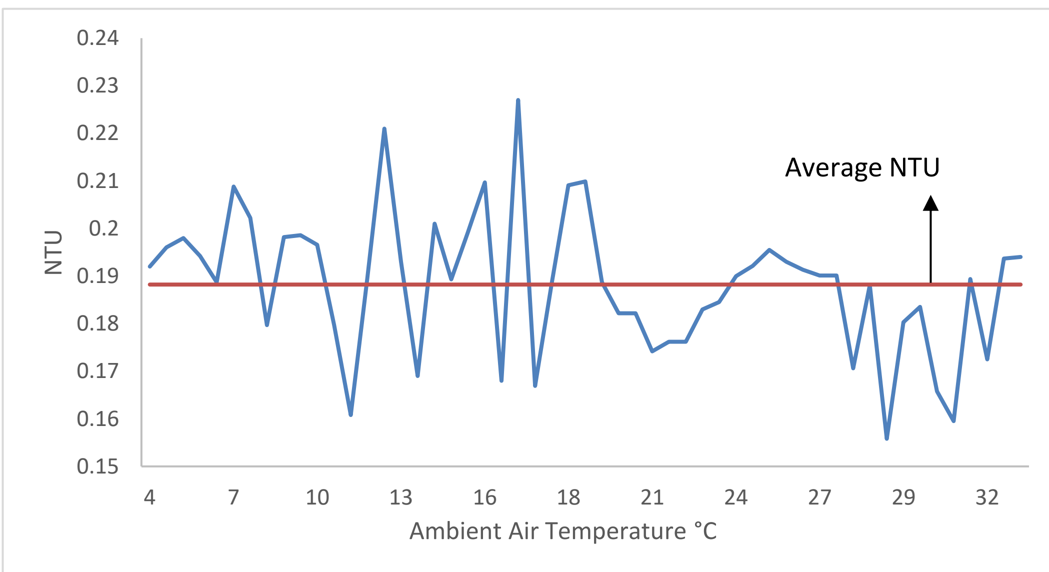

Figure 3. 4: NTU at the different ambient air temperature

3.7.2. Actual cooling tower model results

The water and air data on September 15, 2020, between 1 am to 2 am from Table 3.2 is implemented in the developed model and Eq. (3.1) - (3.8) based on the steps which are shown in Figure 3.3 for predicting the outlet air and water condition of the cooling tower. The outlet water temperature based on the simulated results has less than 0.1% error which shows a reasonable agreement with the actual data.

Table 3. 3: The outcome of developed model for one hour operation

Outlet water mass flow rate	65.2 kg/s
Actual outlet water temperature	34 °C
Simulated outlet water temperature	34.1 °C
Simulated outlet air temperature	13.7 °C
Simulated outlet air RH	100%
Outlet air volumetric flow rate	78 m^3/s
Cooling tower range	7.1 °C
ε_a	0.14
Water loss	0.6 kg/s

To analyze the cooling tower performance in different ambient air conditions, Table 3.2 data ($T_{w,i}$, $T_{\mathrm{a},i}$, $RH_{a,i}$, $\dot{m}_a$, $\dot{m}_{w,i}$, P) and calculated average NTU value of the cooling tower from 1 am 14 September 2020 to 12 pm 20 September 2020 (168 h) have been implemented into Eq. (3.1)-(3.8). By applying the first 9 steps of Figure 3.3, the model can predict outlet water temperature as well as air-side effectiveness and cooling tower range in selected 168h. These results showed that the average air-side effectiveness value in different ambient air conditions was 0.148. However, this value was lower than past studies ranged which were between 0.3 to 0.6 [71]. Filler is a medium used inside the cooling towers to increase the heat exchanging surface area between air and water [143]. The low airside effectiveness of the cooling tower cam be a result of non-efficient filler in the cooling tower which needed to be replace with a new one. The average cooling tower range in the same period was calculated using Eq. (3.14) at 6.8 °C. The cooling tower range value could be varied between 0.6 to 7.8 °C or more based on the cooling demand and performance of the cooling tower [111]. The selected cooling tower showed a high R value because of working on 100% load in the selected time to meet the cooling demand.

3.7.3. Improved cooling tower results

Cooling tower performance improvement were performed by applying selected data from Table 3.2 (total 168h from 1 am 14 September 2020 to 12 pm 20 September 2020) on Eq. (3.1) to (3.13) based on 12 steps described in Figure 3.3. The results reveal the improved outlet water temperature, inlet air volumetric flow rate, fans' power consumption, water loss mass flow rate, air-side effectiveness, and cooling tower range in selected 168h. The outcomes of the improved model versus actual model for evaluating the performance and predicting energy and water savings of the selected cooling tower for one set of data which was worked on September 15, 2020, between 1 am to 2 am, are presented in Table 3.3.

Table 3. 4: The outcome of actual and improved cooling tower modeling in one hour operation

	Actual model	Improved model
Water loss mass flow rate	0.63 kg/s	0.54 kg/s
Outlet water temperature	34 °C	35 °C
Inlet air volumetric flow rate	78 m^3/s	63.7 m^3/s
Cooling tower range	7.1 °C	6.2 °C
ε_a	0.14	0.145
Fans' power consumption	59.64 kWh	33.83 kWh

Changing the outlet water temperature was achieved by adjusting the inlet air volumetric flow rate. The results of applying data from Table 3.2 was selected (total 168 h from 1 am 14 September 2020 to 12 pm 20 September 2020) to Figure 3.3 equations are shown as in Figure 3.5. The inlet air volumetric flow rate was adjusted to have constant outlet water temperature at 35 °C, even the cooling tower was working at 100% load. Consequently, the cooling tower needs to work on 100% load at those time intervals, as shown in Figure 3.5.

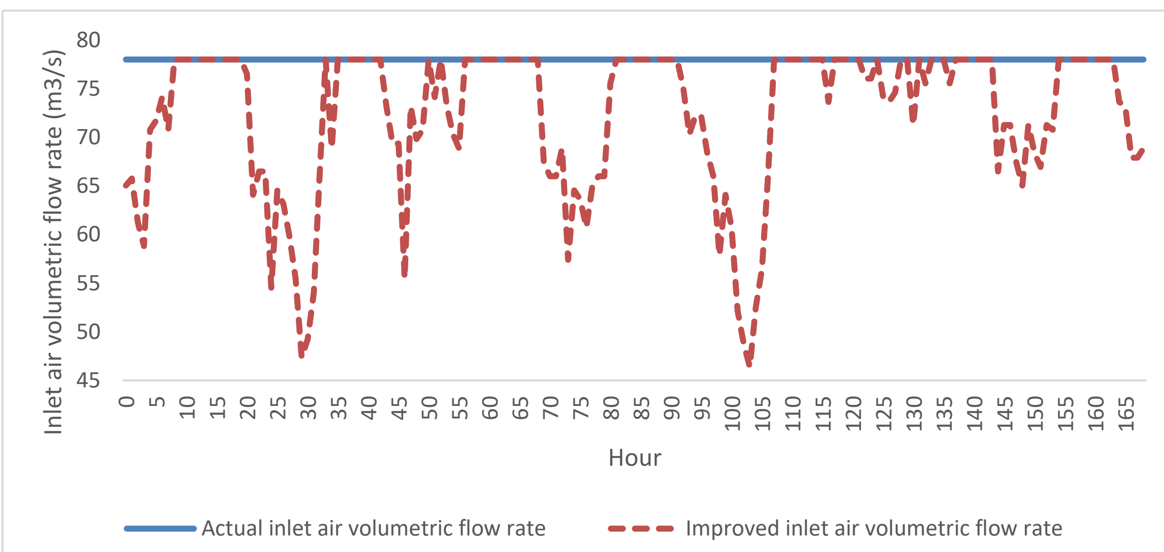

Figure 3. 5: Actual cooling tower air inlet volumetric flowrate versus the improved output

The improvement plan has been performed for 1 week of operating the cooling tower (total 168h from 1 am 14 September 2020 to 12 pm 20 September 2020). The improvement has been achieved by applying data from Table 2 to the Eq. (3.1) to (3.13) using 12 steps of Figure 3.3. To better clarify the cooling tower water and energy savings in one-day operation time, the Figures 3.6 and 3.7 are provided. The water and energy savings of the improvement plan for 15 September 2020 as a sample day is shown in Figures 3.6 and 3.7. Figure 3.6 demonstrates how much energy could be saved in every hour of a sample day if the improvement plan was performed. It can be seen in Figure 6 from 11am to 6pm of the selected day, the cooling tower could not meet the 35 °C as the improved outlet water temperature setpoint, and the fans must work in 100% load. Consequently, there was no potential for saving between 11am to 6pm, and the actual and improved values for energy consumption were equal. Figure 3.6 illuminates that in the higher ambient air temperature the improvement approach could not be performed. While in colder time of a day the improvement approach led to save fans' energy consumption. 409.05 kWh energy saving could be captured by reducing fan power consumption on the selected day. Also, 5.23 m^3 water could be saved on the selected day with an improvement approach which shows in Figure 3.7. Based on Figure 3.7, by performing the improvement approach, water loss through the cooling tower has been reduced as well as the fan's energy consumption.

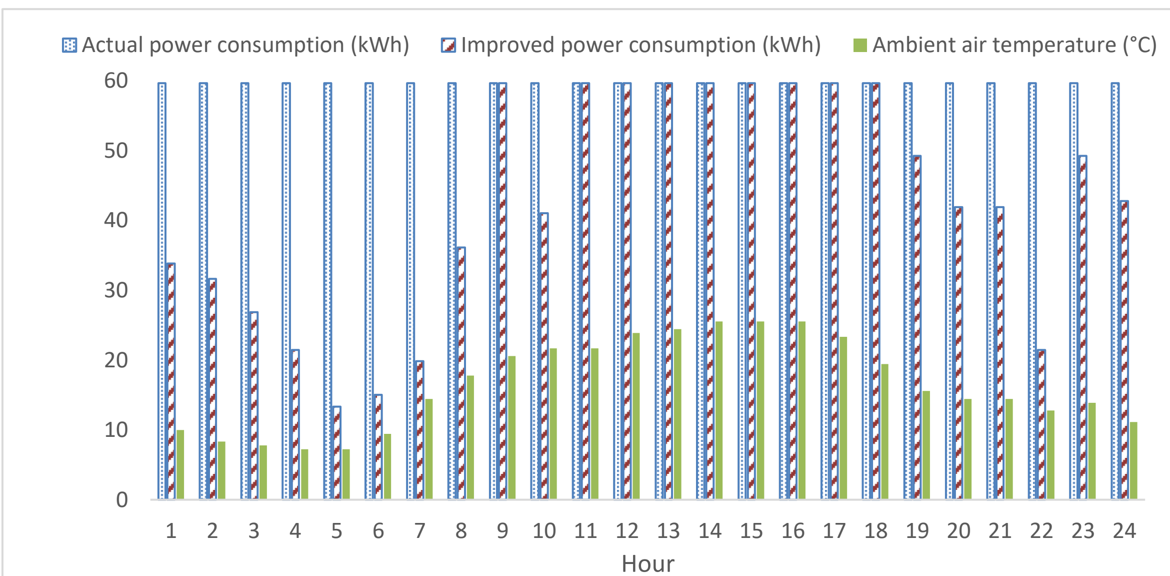

Figure 3. 6: Actual cooling tower fan power consumption versus improved output

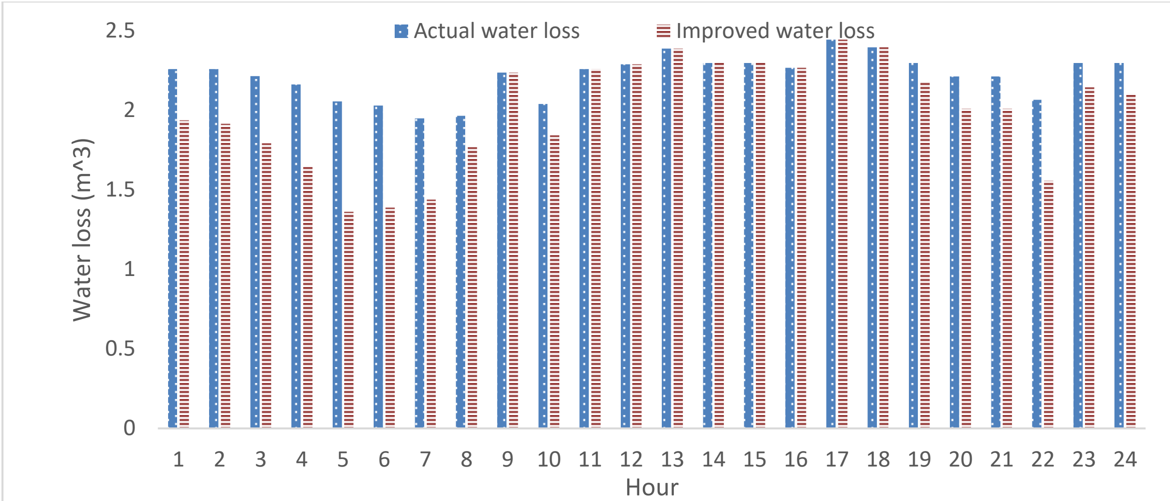

Figure 3. 7: Actual cooling tower water loss versus improved output

To show the improvement approach results in longer operation time, one-month data has been taken form Table 3.2. The improvement study has been done for one month working of the selected cooling tower from 1 am 14 September 2020 to 12 pm 13 October 2020 (720 h) by applying data from Table 3.2 to the Eq. (1) to (13) using 12 steps of Figure 3.3 to investigate water and energy savings in the cooling tower. During selected time, the cooling tower worked at different performance conditions such as water inlet temperature and different ambient air conditions. Based on the developed model during the selected time, the cooling tower fan consumed 28035.50 kWh energy on its operation time. While, the improved model results showed 16830 kWh energy consumption at the same period. As a result, applying the cooling tower improvement plan could save up to 40% of the energy usage of the selected cooling tower in the selected month. Using electricity cost in the cooling tower location from Table 3.2 reveals that up to $1008.50 could be saved in the selected month. Figure 3.8 shows the cooling tower's fans energy consumption in every hour for both actual and improved model during selected month operation.

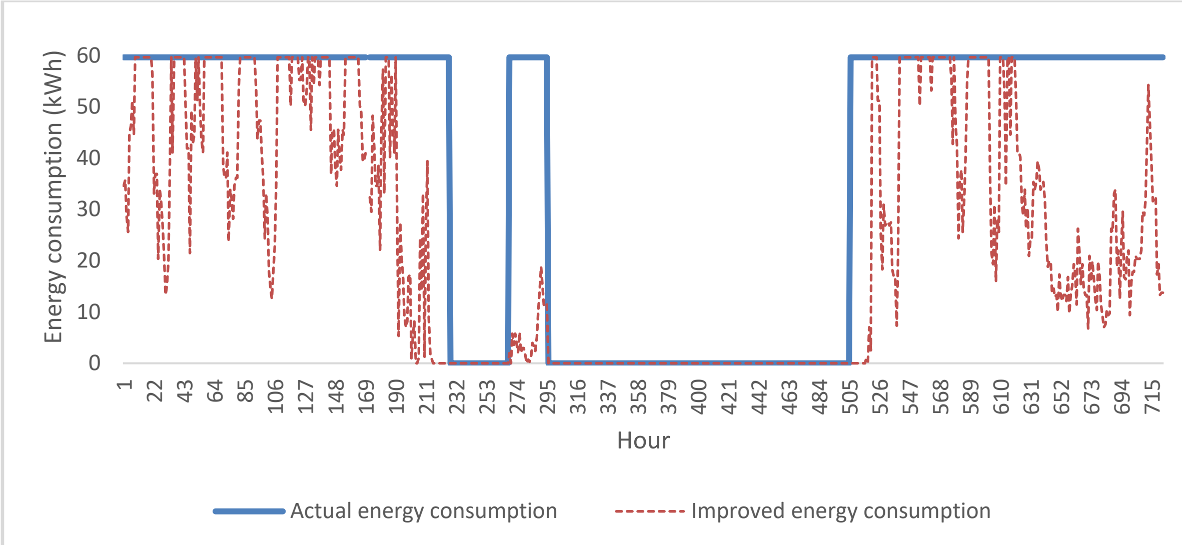

Figure 3. 8: Cooling towers fans energy consumption in actual and improved model in one month operation

One-month loss water savings due to evaporation from improved approach in the same time period as Figure 3.8, is shown in Figure 3.9 which illuminates that the water loss due to evaporation through the cooling tower depends on the cooling tower operation condition and its ambient air conditions. Actual and improved models' results reveal that the selected cooling tower can reduce up to 17.5% water consumption which is 179.49 m3. Applying the water cost from Table 3.2 shows that $116.67 could be saved in the improved operation of the cooling tower in the selected period.

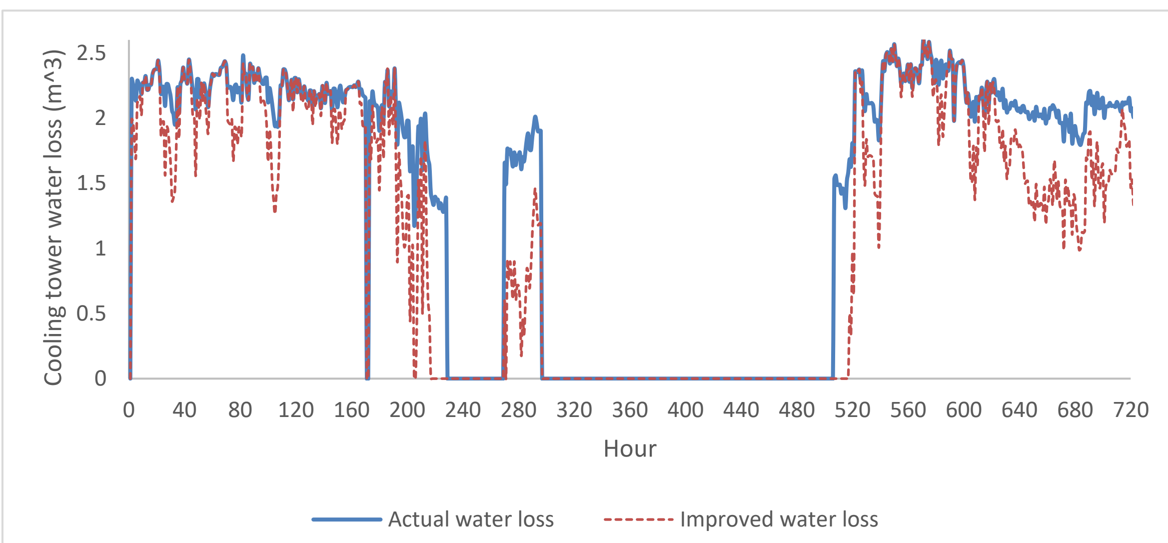

Figure 3. 9: Cooling tower water loss in actual and improved model in one month

During one month of cooling tower operation, water and energy improvement option analysis shows that up to \$1125.17 of operation cost could be reduced. Most of the saving is provided by reducing fans' energy consumption. Since the selected cooling tower usually works four months a year, the annual savings could be estimated up to \$4500. In addition to the energy, water, and money savings derived from the improved plan, the improvement approach provides some environmental benefits. Since the average CO2 emission in the US power generation sector was 0.41 kg/kWh [155], the improvement option can reduce up to 18,377.02 kg of CO2 emissions in the US power generation sector. By considering that selected cooling tower operates in four months in a year, annual water saving using the improved model is estimated to be 689.96 m^3 which is a significant saving due to the future water shortage challenge.

3.8 Validation

In order to validate the cooling tower model, a comparison between actual data and cooling tower's model was made based on provided data, as shown in Figure 3.10. The cooling tower outlet water temperature is predicted using Braun's model by applying one week data from Table 3.2 (total 168h from 1 am 14 September 2020 to 12 pm 20 September 2020) to the Eq. (3.1) -(3.8) with first 9 steps described in Figure 3.3 for every hour. The NTU value was defined as 0.188. Also, the actual outlet water temperature has been recorded using the installed water temperature sensors in the same period. A suitable agreement (less than 1% error) between predicted and actual water outlet temperature has been observed as shown in Figure 3.10. This fact shows that the model can predict the cooling tower performance parameter well.

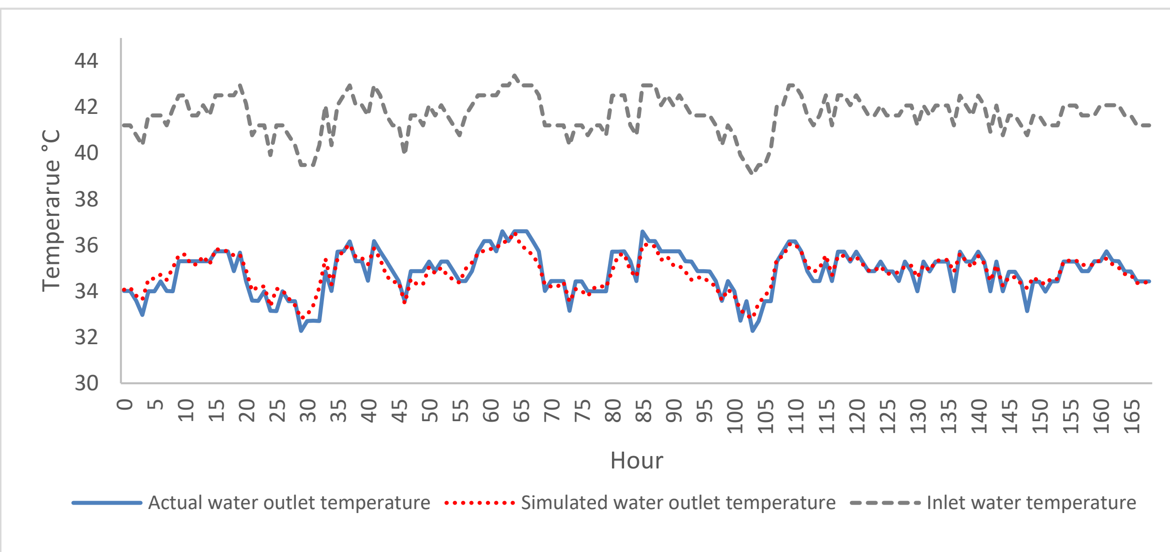

Figure 3. 10: Validation of the modeling results by using actual outlet water temperature

3.9 Conclusion

Improving the sustainability of the wet cooling towers can be achieved by reducing their energy and water consumption as well as their environmental impacts. The purpose of this study was to present a modeling approach in the wet cooling tower and evaluate its performance and other critical parameters by considering its ambient air condition. Also, this study presented an improvement approach to reduce water and energy consumption in the mechanical draft wet cooling tower.

A Braun's model is determined to model an existed mechanical draft wet cooling tower at the UI, Moscow campus. DAQ has been employed for recording the data from cooling tower and ambient air condition. The results of modeling are verified by actual data from DAQ. Based on the modeling outcomes, the NTU value for the modeled cooling tower is equal to 0.188. Also, the average air-side effectiveness (ε_a) is calculated 0.148. The low NTU and ε_a shows that the selected tower did not work efficiently. To improve the performance of the tower, replacing the fouled old filler with the new efficient filler in the tower is suggested which enhances heat exchanging capability of the cooling tower.

The selected wet cooling tower works in its 100% load without considering ambient air and operation conditions of the cooling tower. This operation plan led to waste a considerable amount of water and energy while there was a possibility of reducing cooling tower load in the colder ambient air condition or lower cooling demand. The cooling tower outlet water temperature is adjusted to a value that can also meet the cooling demand of the facility and reduce the cooling tower's load. Consequently, the water outlet temperature is set to 35 °C which was the average outlet water temperature from the cooling tower. This option leads to reducing cooling tower fans' energy consumption and its water loss due to evaporation. For one month operation of the improved plan, up to 28035.50 kWh energy and 179.49 m^3 of consumed water could be captured, and about $1125.17 of the operation cost could be reduced.

The cost, energy, and water savings in the proposed improved model makes a cooling tower more sustainable with lower environmental impact. The proposed adjustment does not need any initial investment or installing any new equipment. This practical approach makes money and improves sustainability from nothing that will attract engineers, operators, and researchers in the future. Future research could focus on savings through the pumps for delivering water from the tower to the facility and its related system improvement.

The developed model can improve the water and energy consumption of every cooling tower considering its ambient air conditions. Applying real time data from different cooling towers in the different climate conditions on the developed model could be a great area for future studies. Since most of the cooling towers have constant outlet water temperature set point without considering its ambient air condition, applying proposed model to define new set point which depends on ambient air condition can provide significant energy and water savings in the future studies. Also, cooling tower pumps energy saving from reducing water loss can be investigated in the future studies to illuminate indirect energy savings from the developed model.

Chapter 4: Guideline for Electricity Generation from Hot Springs (Natural Energy Storage Systems): A Techno-Enviro-Economic Assessment

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Sustainable Energy Technologies and Assessments

https://doi.org/10.1016/j.seta.2021.101407

4.1. Abstract

The scattered hot springs on the globe are natural thermal energy storages that are available for industrial and recreational advantages. A hot spring is a hydrothermal system that can be used for power generation for a future with sustainable power supply. Therefore, there is a demand for a comprehensive study for the estimation of power capacity of hot springs based on their mass-flowrate and temperature. A techno-enviro-economic study is conducted to estimate the power generation capacity of hot springs. The organic Rankine cycle energy plant is modeled and simulated for various temperatures and mass-flowrates of hot springs in the present study. For hot springs with temperature and mass flowrate from 60 to 90 ^{o}C and 5 to 50 kg/s, respectively, power generation capacity alters from 9.3 kW to 303 kW and the levelized energy cost is from 0.02 $/kWh to 0.11 $/kWh. The results indicate that hot springs with a higher temperature and mass-flowrate have a higher thermal efficiency and power generation capacity for the organic Rankine cycle plants while payback period, levelized energy cost, and specific investment cost shrink. The hot springs with higher temperatures and/or larger mass-flowrates are financially ideal candidates for initiating the power generation plants.

4.2. Introduction

The accessibility of energy resources is an essential concern for each community. While near 80% of primary energy comes from fossil fuels [156], projects are in progress to achieve half of the electricity generation from renewable sources by 2030 [157]. Renewable energy technologies are cleaner sources of energy while they are independent of fossil fuel shortage [158], particularly when several countries such as Indonesia are estimated to deplete their fossil fuel reservoirs in five years [159], [160]. Geothermal energy is known as a renewable energy source with great potential which is already underutilized around the world [161]. Geothermal energy as natural energy storage is an enormous source of energy. Recent estimation shows that around 43×10^{15} GJ energy is stored at a depth of 3 km from the Earth's surface [162]. The commercial application of geothermal energy resources was recorded more than 100 years ago [163], while it has been forecasted that more than 8.3% of the world's power generation would be harnessed by geothermal energy in 2050 [164]. Total geothermal power generation was reached 116,000 GWh in 2018 and can be extended to 282,000 GWh by 2030 [165]. The geothermal energy media are dry steam, liquid water, and a mixture of steam and water in various temperatures, that go through extraction technologies [166].

Hot springs, mud pools, and geysers are also functional for hydrothermal systems in geothermal fields [167]. These media are commonly used in the direct utilization of geothermal energy for agricultural, swimming pool, and greenhouse purposes [168]. However, the indirect utilization of geothermal energy is implemented for electricity power production [169], like Chena hot springs commercial power plant that generates 210 kW of electrical power with 8.2% thermal efficiency [170]. Thermal springs with temperatures between 21 to 149°C exist in more than 80 countries [171]. Springs with a temperature more than the humans' body are known as hot springs [172]. In the United States of America (USA), 1072 thermal springs have been identified in 23 States [173]. Also, 3398 hot springs have been recorded in China [174]. These numbers present the promising capabilities of hot springs for implementing direct or indirect power generation.

The high initial investment is one of the major barriers for geothermal energy plants [175], [176]. While the cost of drilling can increase to 50% of the total cost of a typical geothermal power plant project [177] due to the hardness of the rock drilling, harnessing power from hot springs as a heat source will significantly reduce the total cost of a geothermal project [166]. Natural eruption in a hot spring makes it natural energy storage with more accessibility in comparison with the deep-well geothermal source. Using deep wells for serving the geothermal energy plant may increase the risk of geological changes but utilizing hot springs with a natural eruption system as a heat source decreases the environmental impact in geothermal power plants [178]. Also, developing new improved systems to utilize accessible geothermal energy especially shallow geothermal energy sources is a unique opportunity for the future of renewable energy systems [175], [179].

The objective of this study is to provide a guideline for harnessing energy from hot springs as natural energy storage in power generation plants. Although using low-temperature energy sources has received more attention in recent years, there is not a comprehensive investigation of power generation by hot springs through ORC. The capabilities of hot springs to generate electricity are assessed in the study by using the thermodynamic and financial approaches. A technical model of the hot springs for power plants is developed as well as an economic and environmental model for the hot springs with a wide range of temperatures and water discharge mass flow rates. The study is aimed to find the impact of hot spring temperature and water discharge mass flow rate on the energy generation capacity of a potential power plant. The distinctiveness of this study is in the estimation of the power generation from a hot spring by knowing the mass flow rate and the temperature of that. Besides the power generation estimation, the technical, economical, and environmental aspects of the energy plant are approximated as well. Having these data is vital for a sustainable future for encouraging investors to support the energy sector.

4.3. Background

The natural energy storage (geothermal) resources are divided into three main categories based on their temperatures as follows [180]:

- High-temperature reservoirs (i.e., $T > 220°C$) are the most suitable ones for commercial electricity production.
- Medium-temperature reservoirs (i.e., $100 < T < 220°C$) are the most available resources,
- Low-temperature geothermal resources (i.e., $T < 100°C$) have the lowest power generation among geothermal resources.

The organic Rankine cycle (ORC) is an environmentally friendly approach for generating power proposed to utilize low-grade heat sources [181], [182]. ORC has attracted much attention because of its outstanding performance in power generation from low-temperature heat sources in recent years [183].

The ORC can specifically address two main concerns of current energy systems:

1- Decreasing the energy intensity of buildings and industry by recovering waste heat [184] through the use of Combined Heat and Power (CHP) systems.
2- Developing a way to convert renewable energy sources (solar [185], geothermal [186], and biomass [187]) to electricity more efficiently [188].

The ORC is conceptually similar to a Steam Rankine Cycle (SRC), but the working fluid is an organic fluid instead of water [189]. The fundamental parameter in utilizing ORC refers to the organic working fluid that should be safe, environmentally friendly, and economical [190]. When water implemented as the working fluid, the following conditions must be considered [191]:

- additional pressure in the evaporator,
- excess complexity and cost of turbines,
- superheating to prevent condensing during expansion.

While employing organic fluid instead of water has other benefits including less heat for the evaporation process, the smaller temperature difference between the evaporation and condensation processes, and no superheating required within the cycle [192]. Performing a techno-enviro-economic analysis in selecting suitable working fluid is important in the ORC applications since it has a great role in the cost-effectiveness of energy plants. Most ORC low-temperature power generation applications have employed R245fa and R134a as working fluid. However, the working fluids such as R227ea with less boiling point temperature have a higher capability with low-temperature sources. Also, a techno-enviro-economic analysis using R227ea in a low-temperature geothermal application has not been done before [193].

Consequently, the application of geothermal energy sources using ORC systems, especially for low-temperature heat sources, has been increased recently [194], [195]. Figure 4.1 shows the important historical progress in the application of ORC in geothermal energy technologies [196], [197].

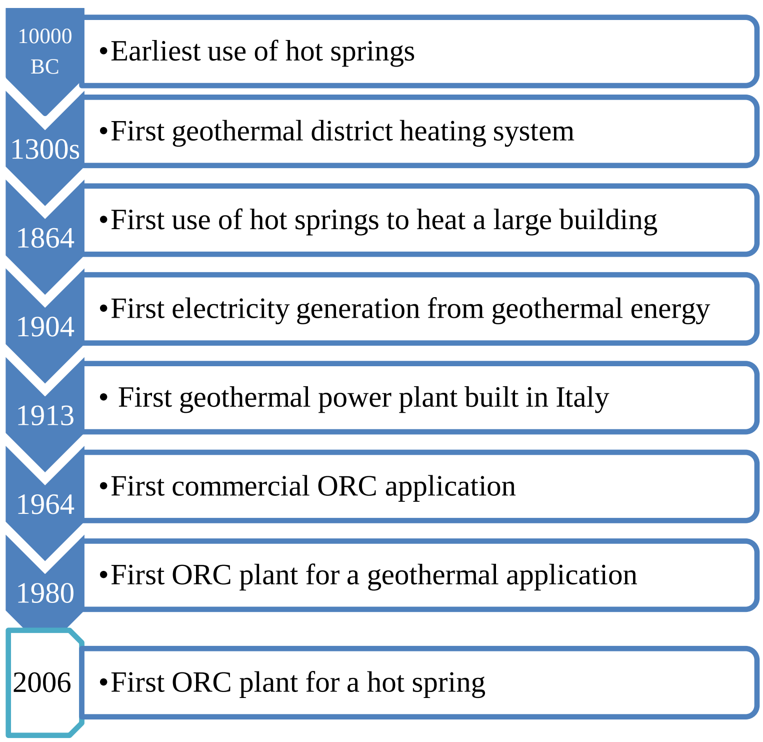

Figure 4. 1: Historical progress records in exploiting from ORC and geothermal energy

4.4. Methodology and Modeling

4.4.1. Thermodynamic Modeling

The single-stage ORC consists of the pump, generator, evaporator, condenser, and expander, as illustrated in Figure 4.2. A closed-loop ORC is designed for the circulation of working fluid in the cycle, separating it from the hot source (hot spring) and cold source (cold water or air). The operation of the cycle contains four steps [198], [199]:

1- The high-pressure liquid turns to vapor working fluid in the evaporator by receiving heat from a hot source and exits the evaporator.
2- The high-pressure vapor expands via the expander joint with the electrical power generator, and therefore, generates electricity.
3- The low-pressure vapor working fluid exits from the expander enters the condenser to get cooled down. The low-pressure vapor turns to saturated liquid by rejecting heat to the ambient cold air in an air-medium heat exchanger.
4- The low-pressure saturated working fluid enters the pump to increase its pressure. The pump makes the working liquid pressure equals the evaporator pressure level. Then, the high-pressure working fluid is ready to enter the evaporator to repeat the cycle.

Applying the mass continuity equation (Eq. (4.1)) to the energy plant is needed to find the mass flow rates of inlet and outlet streams for each component [200], [201].

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = \frac{dm_{sys}}{dt} \tag{4.1}$$

Where $\dot{m}_{\text{in}}$ and $\dot{m}_{\text{out}}$ are mass flow rates (kg/s) at the inlet and outlet of a system, respectively, and $\frac{dm_{sys}}{dt}$ is mass storage rate within a system

Applying energy rate balance to the energy plant allows a deeper understanding of the energy level of the ORC components in the form of work and heat. The energy balance based on the first law of thermodynamic can be written as [201]:

$$[\sum (\dot{Q} - \dot{W})_{in} - \sum (\dot{Q} - \dot{W})_{out}] + \sum \dot{m}_{in}(h + \frac{V^2}{2} + gz)_{in} - \sum \dot{m}_{out}(h + \frac{V^2}{2} + gz)_{out} = \frac{dE_{sys}}{dt} \tag{4.2}$$

Here, h is the specific enthalpy in kJ/kg, $\dot{Q}$ and $\dot{W}$ are heat and work transfer crossing the component boundaries in kJ/s, respectively. Also, V is stream velocity in m/s, and z is stream elevation in m. $\frac{dE_{sys}}{dt}$ is the energy storage rate within a system in kJ/s.

To find the heat transfer area of the evaporator and condenser, the Logarithmic Mean Temperature Difference (LMTD) is used [202]. The LMTD is a well-known method to evaluate the performance of the heat exchangers and is suitable for the evaporation and condensation process in the heat exchangers [203]. The LMTD values associated with the evaporator and condenser can be written as follows [202].

$$\Delta T_{lm} = ((T_{h,out} - T_{c,in}) - (T_{h,in} - T_{c,out}))/\ln\left(\frac{(T_{h,out} - T_{c,in})}{(T_{h,in} - T_{c,out})}\right) \tag{4.3}$$

where ΔT_{lm} is the logarithmic mean temperature difference in °C. $T_{h,out}$ and $T_{h,in}$ are hot stream temperature in the outlet and inlet of the heat exchanger (i.e., evaporator or condenser) in °C, respectively. Also, $T_{c,out}$ and $T_{c,in}$ are cold stream temperature in the outlet and inlet of the heat exchanger (i.e., evaporator or condenser) in °C, respectively.

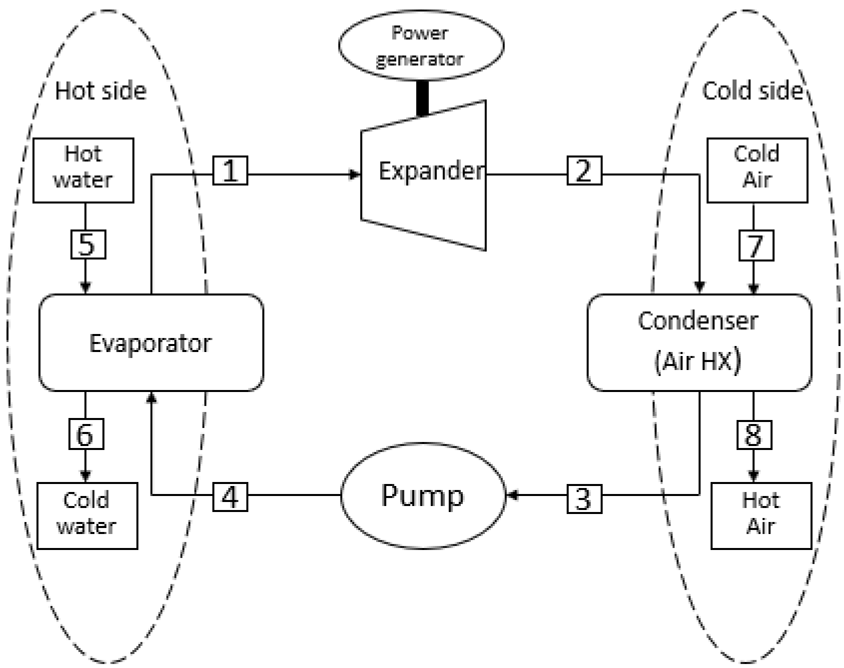

Figure 4. 2: Schematic of ORC power generation by using hot springs

Eq. (4.1) and (4.2) are applied to each component of the ORC cycle presented in Figure 4.2. The following assumptions are used to develop the thermodynamic model of the system Eq. (4.4) to (4.7).

- ORC system is at steady-state condition,
- A constant amount of working pressure in condenser and evaporator,
- Inlet and outlet sections of each component are at thermodynamic equilibrium,
- The pump and the expander (i.e., turbine) are adiabatic,
- The changes in potential energy and kinetic energy are negligible.

By applying Eq. (4.2) on the ORC components, Eq. (4.4), (4.5), (4.6), and (4.7) are developed in which they are used for energy rate estimation for different components within the hot spring power plant. Regarding the heat exchangers, it should be noted that the working fluid enters the evaporator in saturated liquid temperature (point 4) and leaves the evaporator in saturated vapor temperature (point 1). Also, working fluid enters the condenser in saturated vapor (point 2) and leaves the condenser in saturated liquid condition (point 3).

Evaporator: $\dot{Q}_{in} = \dot{m}_w \times (h_5 - h_6) = \dot{m}_{wf} \times (h_1 - h_4) = U_{evap} \times A_{evap} \times \Delta T_{lm,evap}$ (4.4)

Expander: $\dot{W}_T = \eta_m \times \dot{m}_{wf}\left(h_1 - h_{2,s}\right) = \eta_m \times \eta_p \times (h_1 - h_2)$ (4.5)

Condenser: $\dot{Q}_{out} = \dot{m}_a \times (h_8 - h_7) = \dot{m}_{wf} \times (h_2 - h_3) = U_{cond} \times A_{cond} \times \Delta T_{lm,cond}$ (4.6)

Pump: $\dot{W}_p = \dot{m}_{wf}(h_{4,s} - h_3)$ (4.7)

Where the U_{evap} and U_{cond} are the overall heat transfer coefficient of the evaporator and condenser in kW/m^2 °C, respectively. Also, A_{evap} and A_{cond} are the heat transfer areas of the evaporator and condenser in m^2, respectively. η_m , η_T, and η_p represent the generator efficiency, isentropic efficiency of the expander, and isentropic efficiency of the pump, respectively. In addition, $\dot{m}_{wf}$, $\dot{m}_w$, and $\dot{m}_a$ denote the mass flow rate of the working fluid, water, and air in kg/s, respectively. $\dot{W}_T$ and $\dot{W}_p$, show produced work in the expander and required work in the pump in kW, respectively.

Applying Eq. (4.2) on the ORC cycle leads to Eq. (4.8). The work output and thermal efficiency of the ORC are determined by Eq. (4.8) and (4.9) as follows.

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_p \tag{4.8}$$

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{4.9}$$

Here, $\dot{W}_{net}$ is the power output of the cycle in kW, and η is the thermal efficiency of the cycle. Also, $\dot{Q}_{in}$ represents the heat transfer rate to organic fluid in the evaporator in kW.

4.4.2. Enviro-Economic Modeling

Cost estimation is an essential part of a feasibility study. The economic model addresses both economics of the energy plant as well as its environmental impact. The overall cost of the plant is categorized into three major costs: initial investment, operation cost, and environmental business cost [204].

4.4.2.1. Initial investment

The initial investment estimation in the ORC system was estimated by finding the cost of each component. Two heat exchangers in the roles of the evaporator and condenser were implemented in the design. Therefore, two heat exchangers, a pump, an expander, and an electricity generator were the main component of the ORC system. The cost of labor, working fluid, and pipes were not considered since they vary from one project to another. Ultimately, the total hardware cost for the major hardware is estimated by Eq. (4.10) [204].

$$C_{tot} = C_{evaporator} + C_{condenser} + C_{turbine} + C_{pump} + C_{generator} \tag{4.10}$$

The Specific Investment Cost (SIC) is the ratio of the total hardware cost to generated power as shown in the below equation [205], [206]:

$$SIC = \frac{C_{tot}}{\dot{W}_{net}} \tag{4.11}$$

The cost of a loan on the energy plant is calculated through the Eq. (4.12), which results in a yearly stream of payments for the loan [202], [207]:

$$Y = C_{tot} \times \left(\frac{i(1+i)^q}{(1+i)^q - 1}\right) \tag{4.12}$$

Here, i represents the annual interest rate for the loan, and q is the number of yearly payments. Also, Y denotes yearly payment.

Considering the variation of the value of money over time, the initial investment in the present time is written as [204]:

$$\text{Initial investment} = \sum_{a=1}^{q} Y(1+IR)^{-a} \tag{4.13}$$

where *a* represents the number of amortization years ranging from 1 to q, and IR is the inflation rate.

4.4.2.2. Operation cost

The present value of the operation cost accounts for fuel, insurance, and maintenance costs which vary during the life of the energy plant because of the changing value of the money, are presented in Eq. (4.14) [202], [204].

$$\text{Operation cost} = \sum_{g=1}^{n} (FC + I\&M)(1+IR)^{-g} \tag{4.14}$$

Here, FC is the annual fuel cost which equals 0 for a renewable energy power generation system. Also, I&M is the Insurance and Maintenance of annual costs, and g is the performance years of the energy plant ranging from 1 to n.

4.4.2.3. Environment Business Cost

The environmental effects of using the energy plant can be quantified in the forms of carbon taxes and tax benefits as presented in the below equation [202], [204]:

$$\text{Environment business cost} = \sum_{g=1}^{n} CT(1+IR)^{-g} - \sum_{g=1}^{n} TB(1+IR)^{-g} \tag{4.15}$$

By combining the cost of the initial investment (Eq. (4.13)), operation cost (Eq. (4.14)), and environmental business cost (Eq. (4.15)), the overall cost of the energy plant can be determined by Eq. (4.16) [204]:

Overall cost= Initial investment + Operation cost+ Environmental business cost (4.16)

The Levelized Energy Cost (LEC) is the ratio of present value of the overall cost to total net power output by ignoring the field capital cost and field cost. LEC is a useful tool for comparing different energy generation systems that are calculated using Eq. (4.17) [208]:

$$LEC = \frac{Overall\ cost}{\dot{W}_{net} \times t_{op} \times n} \tag{4.17}$$

where t_{op} is the operation hours in a year, and n is the energy plant lifetime in a year.

Based on the present value of the initial investment and the annual net earnings (NE), the static Payback Period (PP) of the energy plant can be estimated by the following equation [209]:

$$PP = \frac{Initial\ Invetment}{P_e \times \dot{W}_{net} \times t_{op}} \tag{4.18}$$

Here P_e represents the price of electricity in $/kWh.

4.4.3. Energy Plant Simulation Software

Aspen Plus (V9) is the leading chemical engineering process simulator in the market that allows the user to build the process model and simulate using complex calculations. The software provides capability of various thermodynamic modeling and simulation suitable for this study by having the ability of modeling components in an energy plant and a library of thermodynamic properties for various organic working fluid. Aspen Plus is also capable of modeling the heat exchangers with the phase-change process in the evaporator and condenser in the ORC plant.

4.5. Sizing the ORC System

The schematic model of a single-stage ORC plant for converting the energy of the hot springs to electricity is demonstrated in Figure 4.3. To estimate the capability of a power generation with an ORC system from a hot spring the components need to be sized.

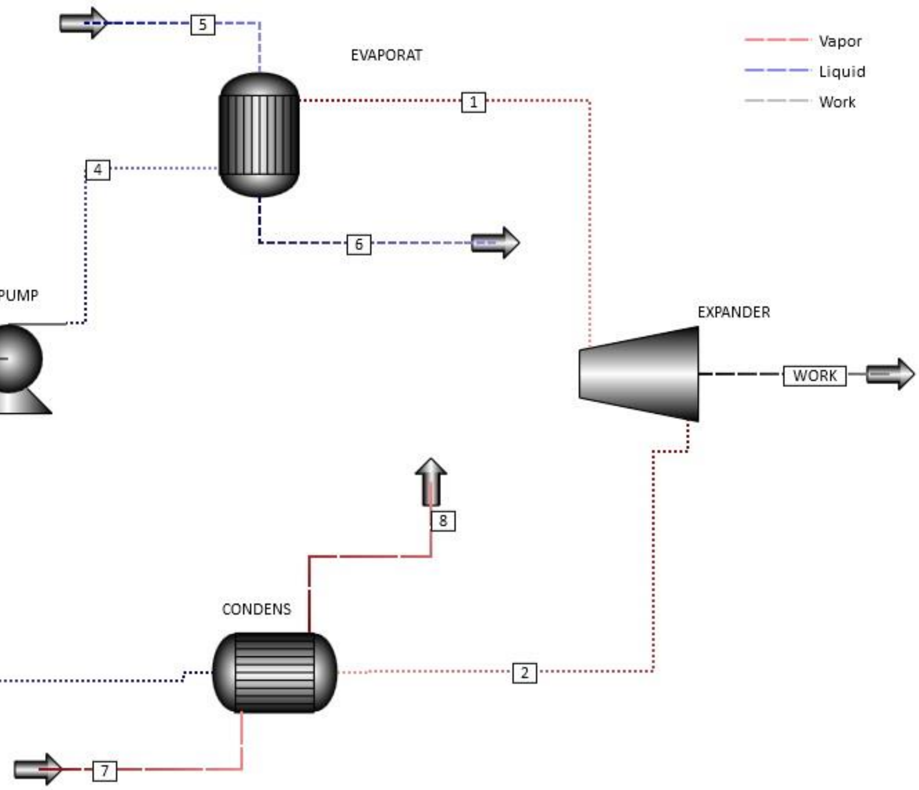

Figure 4. 3: Simplified model for ORC power generation

4.5.1. Organic working fluid selection

R227ea as a non-flammable and non-toxic organic fluid refrigerant is suitable for low pressure, low-temperature applications with high efficiency [210]–[214], especially when the hot source temperature is lower than 92 °C [215], [216]. Since changes of the entropy with the temperature in R227ea is positive, it is known as dry organic fluid; hence, this organic fluid does not need the superheat before its expansion process in ORC [213] while R227ea is selected for the low-temperature ORC system. Table 4.1 exhibits the properties of R227ea [214].

Table 4. 1: Physical properties of R227ea

Properties	Data
Chemical name	1, 1, 1, 2, 3, 3, 3-Heptafluoropropane
Chemical formula	CF3CHFCF3
Molecular weight	170.03 g/mol
Boiling temperature	-15.6 °C
Critical temperature	102.8 °C
Critical pressure	29.8 bar

4.5.2. Organic working fluid pressure

There are high and low pressures in an ORC system that are determined by the organic fluid pressure in the evaporator and condenser. Since the condenser is an air-cooled heat exchanger, the low pressure of the cycle cannot be lower than 5 bar due to ambient temperature which is 18 ^{o}C. Therefore, the condenser pressure is considered as 5 bar. The evaporation pressure categories are shown in Table 4.2 based on the hot springs temperature range [212], [213], [217]. The pressure values have been assumed in the present study based on temperature range to capture the most energy from the hot springs for power generation which is shown in Table 4.2.

Table 4. 2: Evaporator pressure based on hot spring temperature

Evaporator pressure (bar)	Hot source temperature (^{o}C)
9	60- 70
12	71-80
15	81-90

4.5.3. Component specification

The pump needs the energy to maintain the required pressure of working fluid through the cycle. The expander provides mechanical energy for the power generator by expanding the working fluid. The isentropic efficiency of the pump and expander are assumed to be 75%. Power generator which is joint with the expander to convert mechanical energy to electrical energy has an efficiency of 95%. Also, overall heat transfer in the evaporator and condenser is assumed for 0.85 kW/ m^{2} ^{o}C.

4.5.4. Cold source temperature

Although water is a better medium for the condenser in geo-technology, over the possibility of water shortage in the region, the air has been considered as a circulating medium for the condenser. Using the air-medium heat exchanger as a condenser in the ORC plant not only reduces the water consumption of the plant by using the cool air on the higher elevations of hot springs, but it leads to financial advantages over the initial investment and operation cost. It is reported that large-scale geothermal power plants consume 20 L/MWh water on average [218], while the air-cooling condenser ORC plant is free of freshwater intake. Since hot springs are located in mountainous areas with colder temperatures (where the volcanic activity of the earth is probable), using the air as a medium in the condenser is practical. The cold source temperature is the ambient temperature of 18 ^{o}C, then.

4.5.5. Hot source temperature

The evaporators evaporate the organic fluid when the hot source temperature is more than 59 °C. Usually, the hot springs with a temperature between 60-90 °C are widely spread and the current study covers this range. There are few springs with a temperature of more than 90 °C throughout the world.

4.5.6. Hot spring's mass flow rate

The water discharge mass flow rate from the hot springs has been assumed as the mass flow rate of the hot fluid. The mass flow rate range of hot springs has considered between 5 kg/s to 50 kg/s to cover the most common hot springs.

4.5.7. Economic assumption

The life span of the ORC system has assumed 20 years [219], and the annual I&M cost has assumed 3% of the principal cost. The amortization of 10 years with an interest rate of 8%, and an inflation rate of 2% [204]. The production hours in a year for the ORC have considered 8000 hours per year [220]. The USA average electricity cost in 2020 [221] of 0.10 $/kWh has been defined as electricity cost in this study.

All the technical and economic input data and assumptions in sizing the ORC plant and operating points parameters are summarized in Table 4.3. Information for each point is assumed based on the values which are the norm in the power generation industry in Table 4.3. Points in that table have been reflected in Figure 4.3.

Table 4. 3: Assumptions in sizing the energy plant

Technical study assumptions	Working fluid	R227ea
	Ambient air pressure	1 bar
	Working fluid pressure in the condenser	5 bar (Points 2 and 3)
	Working fluid pressure in the evaporator	(Points 1 and 4) 15 bar (hot source temperature ≥ 81 °C) 12 bar (71≤hot source temperature <81 °C) 9 bar (60 ≤ hot source temperature <71 °C)
	Pump isentropic efficiency	75%
	Expander isentropic efficiency	75%
	Electricity generator efficiency	95%
	Heat exchanger overall heat transfer coefficient	0.85 kW/m² °C

	Air temperature (cooling source)	18 °C (Point 7)
	Hot springs temperature	60-90 °C (Point 5)
	Hot springs discharge mass flowrate	5-50 kg/s (Points 5 and 6)
Economical study assumptions	A lifetime of energy plant	20 years
	Amortization period	10 years
	I&M cost	3% of the total hardware cost
	Interest rate	8%
	Inflation rate	2%
	Working hours per year	8000 hours
	Electricity cost	0.10 $/kWh

4.6. Analysis

4.6.1. Technical analysis

By thermodynamic modeling and simulation of the ORC plant, the electricity generation capacity, thermal efficiency, required pump power, and total heat transfer area in evaporator and condenser were determined for a wide range of hot springs. Table 4.3 presents the data used for the analysis. Figure 4.4 shows the flow chart of the technical analysis based on the available data and developed equations and the outcomes in different steps of the analysis.

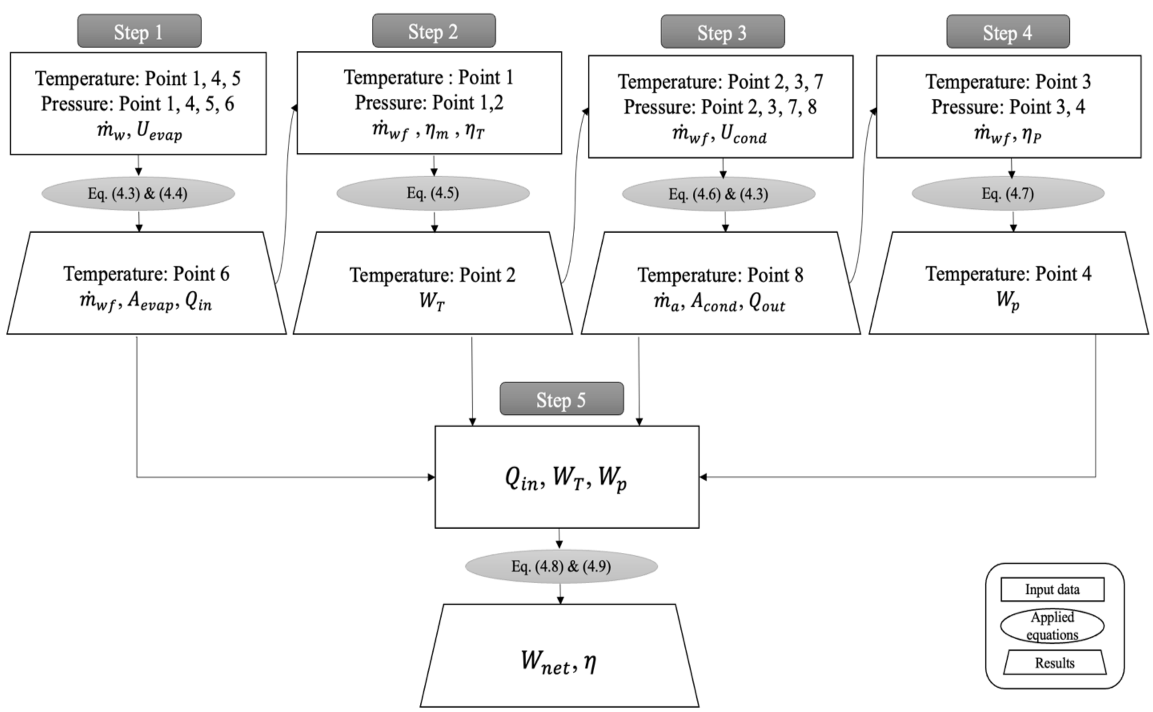

Figure 4. 4: Flowchart of the Technical Analysis of ORC energy plant

4.6.2. Economic analysis

The economic analysis of the energy plant is determined by considering major costs and revenues during its lifetime. Based on the results of the technical analysis of the energy plant, the specification of every component in the designed energy plant is provided. The cost of every component is estimated by using the average of three actual quotes. The heat transfer area, fluids mass flow rates, and total transferred heat in the heat exchangers are provided as required data to design a practical heat exchanger with a proportionate cost. Expander and electric generator prices depend on the estimated power generation in technical analysis. Heat exchangers, expanders, and pumps price quotes were estimated according to the technical results. The total hardware cost was determined using the suppliers' quote by Eq. (4.10). By implementing the value of the power generation and total hardware cost in Eq. (4.11), the SIC value has been determined. Implementing interest rate and amortization information from Table 4.3 to Eq. (4.12) leads to finding annual payment of the loan. The influences of the inflation rate on the initial investment were illustrated using Eq. (4.13) and the inflation rate value in Table 4.3.

4.6.2.1. Initial investment

The proposed economic tool helps for an estimation of the financial aspects of the energy plant projects for hot springs in different locations. To have some references for estimation of the initial investment for the ORC energy plant the actual quotes were collected from the American suppliers on major components. Quotes were obtained for three different sizes of each component and, also three quotes for each size were obtained. The average of three quotes for every size of different components in the ORC plant is presented in Table 4.4.

Table 4. 4: Collected price for ORC components

Component	Unit		Cost ($)
ORC heat exchangers (Shell and tube for evaporator Plate and frame for condenser)	Total heat transfer area	A= 35 m^2	37,000
		A=100 m^2	68,000
		A=188 m^2	101,000
ORC expander and electric generator	Generated power	10 kW	25,000
		75 kW	150,000
		300 kW	300,000
ORC pump	Required Power	5 kW	1,000
		10 kW	11,000
		15 kW	22,000

Through curve fitting for three sizes of equipment equations for finding the cost of each component are developed as shown in Figure 4.5. Initial investment equations for ORC heat exchanger based on the required area per m^2, for the expander and electric generator based on generated power per (kW), and pump based on required power per kW. Summation of the major hardware cost for the ORC heat exchanger and ORC expander and electric generator and, ORC pump results in estimation for the total hardware cost of the energy plant without considering the installation, pipes, and labor costs.

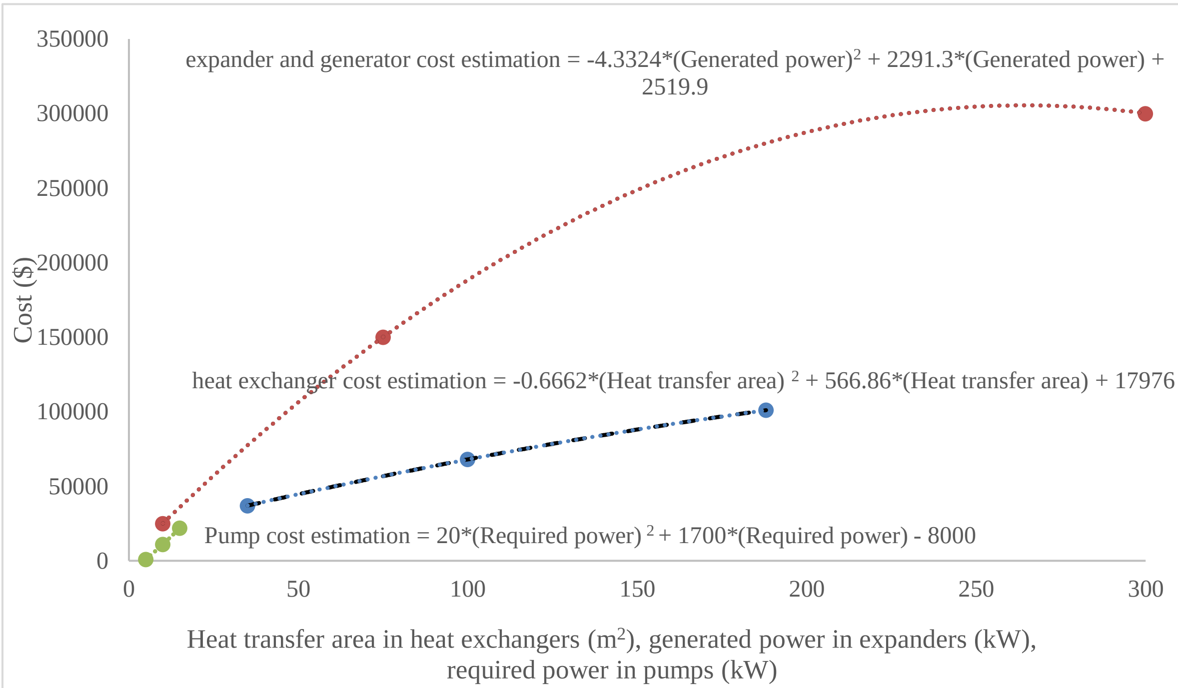

Figure 4. 5: Cost estimation curve for ORC components

4.6.2.2. Operation cost

Operation cost (I&M) was assumed for 3% of the total hardware cost in Eq. (4.14). The influence of the inflation rate on the annual I&M cost was determined by applying the inflation rate from Table 4.3 on Eq. (4.14).

4.6.2.3. Environment business cost

Although CO_2 emission (as an indicator of the environmental impact) is a vital term in analyzing the environmental business cost in energy systems [176], carbon tax and tax benefit affect ORC geothermal energy plants' overall cost. The federal, provincial, and local environmental policies and regulations dictate the carbon tax and tax benefit for different geothermal energy plant projects.

By plugging the initial investment, operation cost, and environmental business cost, in Eq. (4.16), the overall cost of the energy plant can be estimated. Implementing yearly working hours and lifespan of the energy plant from Table 4.3 into Eq. (4.17) results in the energy plant levelized energy cost. Finally, to find the PP of the hot spring energy plant, Eq. (4.18) is used by considering the electricity price from Table 4.3. PP depends on the initial investment, power generation, working hours of the hot spring energy plant in a year, and the electricity price in the consuming location.

Figure 4.6 presents the economic analysis of the ORC energy plant in sequence in a flowchart format.

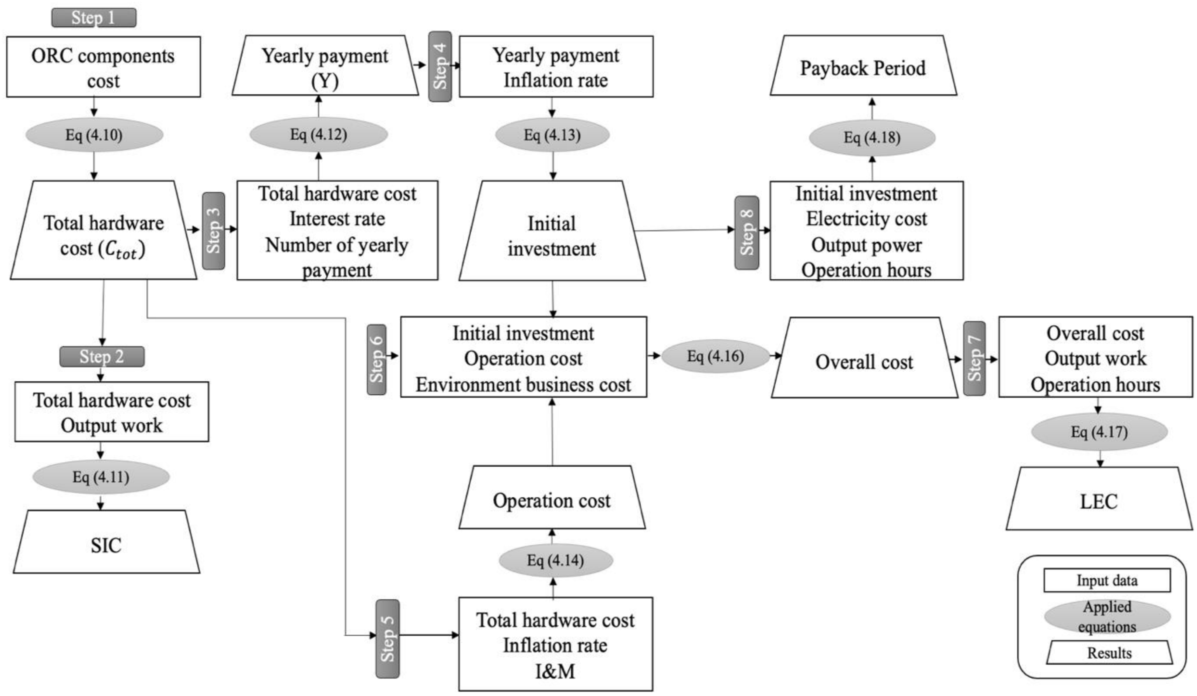

Figure 4. 6: Flowchart of the Economic Analysis of ORC energy plant

4.7. Results and Discussions

The study was designed to provide a guideline for estimation of power generation capacity for hot springs besides the economic aspects of the power generation plants through a dedicated techno-enviro-economic analysis (tool). The outcomes of the developed tool for a hot spring with 76 ^{o}C temperature and 25 kg/s discharge mass flow rate located in the southeast of the Washington state in the USA with 20 years lifespan is presented in Table 5 as an example. The electricity cost is 0.08 \$/kWh in the selected location.

To clarify the impact of interest rate and using loan on the financial characteristics of the hot spring energy plant, two options are investigated for the selected hot spring in Table 4.5. Option 1 assumed that all the costs were covered by an investor without getting loan, while Option 2 determined financial characteristics based on using loan and employed the related assumptions in the Table 4.3.

To investigate the environmental business cost of the hot spring energy plant, it should be noted that there are no carbon tax and tax benefit regulations in Washington state, USA. Therefore, there is not environmental business cost. As Table 4.5 illustrates, using the loan and paying for the interest makes a hot spring energy plant project less economically desirable.

Table 4. 5:Component's specifications in the designed ORC plant for a selected hot spring

Maximum evaporated working fluid mass flow rate (kg/s)	13.4	Total generated work in ORC (kW)	96.8
Total heat transferred in the evaporator (kW)	1620.0	Total heat transfer area in heat exchangers (m^2)	112.2
Generated power in expander (kW)	105.9	The thermal efficiency of the cycle (%)	5.9
Required work in the pump (kW)	9.1		
Option 1 (without loan interest)			
SIC ($/kW)	3696	Initial investment ($)	357,840
Annual Operation cost ($)	10,735	Annual environmental business cost ($)	0
Overall Cost ($) (for 20 years)	533,376	Annual NE ($)	80,937
LEC ($/kWh)	0.034	PP (years)	5.76
Option 2 (with loan interest)			
SIC ($/kW)	3696	Initial investment ($)	479,029
Annual Operation cost ($)	10,735	Annual environmental business cost ($)	0
Overall Cost ($) (for 20 years)	654,565	Annual NE ($)	80,937
LEC ($/kWh)	0.042	PP (years)	7.73

4.7.1. The Power Generation Capacity

The temperature and mass flow rate of the hot spring has a direct positive impact on the power generation capacity of the hot spring. To show the influence of the temperature and mass flow rate on the output power, the temperature was changed from 60 ^{o}C to 90 ^{o}C while the mass flow rate was altered from 5 kg/s to 50 kg/s. The approximate power generation through the energy plant of a single-stage ORC with the mentioned temperature and mass flow rate ranges would be 10 kW to 303 kW as illustrated in Figure 4.7. A hot spring with a water discharge mass flow rate of 50 kg/s a 90 ^{o}C has an approximate net power output of 303 kW, while a hot spring with a water discharge mass flow rate of 5 kg/s at 90 ^{o}C is capable of power generation of about 30 kW. If the temperature of the latter hot spring drops to 60 ^{o}C, the power generation falls lower than 10 kW. It can be concluded that by an increase of the temperature and discharge mass flow rate of hot springs the generated power intensely expands.

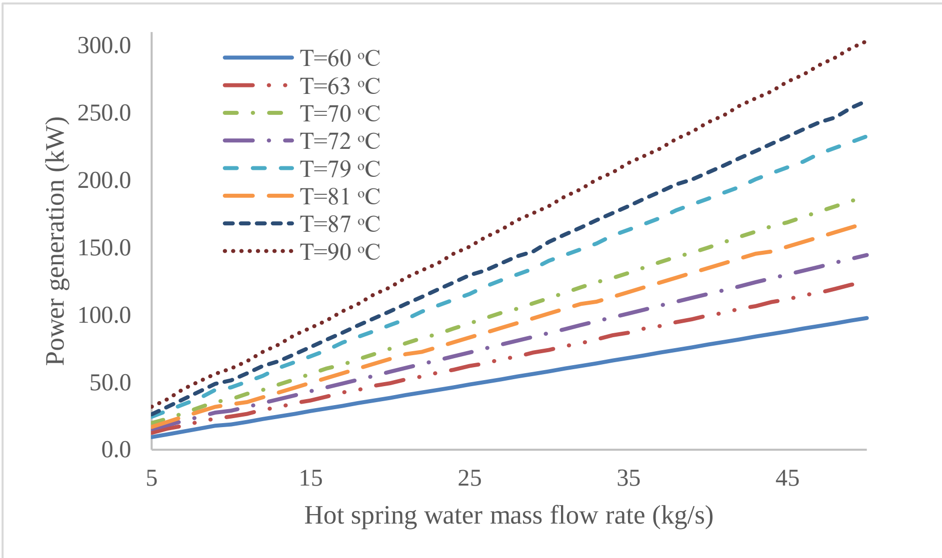

Figure 4. 7: The total working output of ORC using hot spring

4.7.2. Evaporator's role

The major cycle for the energy plant was considered ORC since it was the ideal thermal cycle to work with low-temperature energy resources. To find the thermal efficiency of the energy plants, three different evaporator pressures of 15 bar, 12 bar, and 9 bar were assumed for working fluid (R227ea). The ORC's thermal efficiency for three different pressure levels has resulted in 7.2%, 5.9%, and 4.2%, respectively. Figure 4.8 demonstrates the thermal efficiency of the energy plant on a T-S diagram for the working fluid (R227ea) in three different evaporator pressure range. Since the evaporator and condenser pressure are determined by hot spring temperature and working fluid thermophysical properties, it can be concluded that the energy plant thermal efficiency is influenced by the temperature of hot spring and working fluid selection.

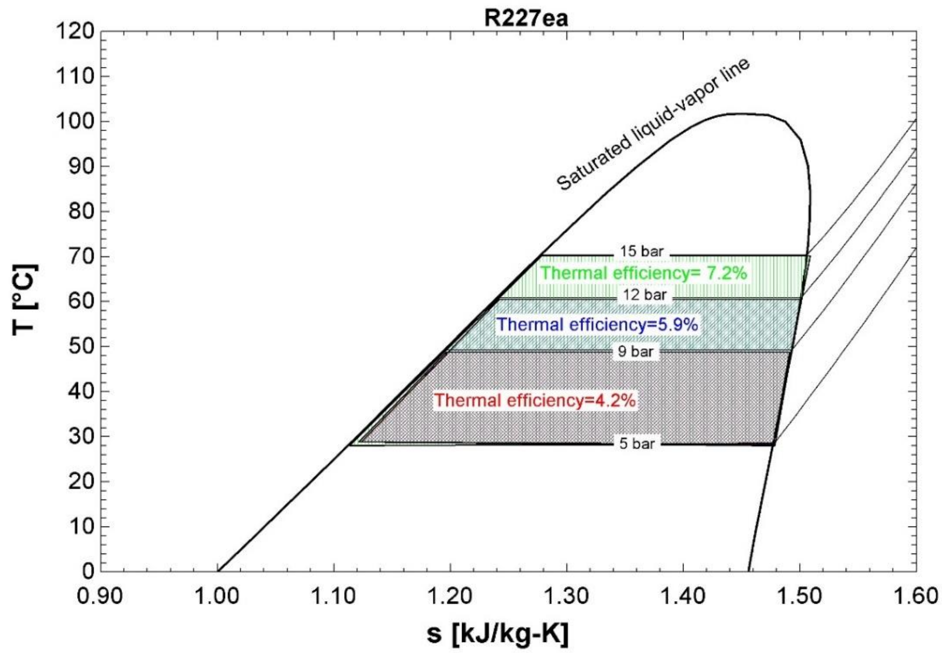

Figure 4. 8: Thermal efficiency of the ORC in the T-S diagram for three working pressures in the evaporator

In the ORC plant, two heat exchangers have been used as evaporator and condenser, refer to Figures 4.2 and 4.3. Since heat-exchanger cost accounts for up to 50% of the total ORC project costs [222], hot springs with higher discharge water mass flow rate and higher temperature are more costly. The heat transfer area is a crucial parameter in the performance of heat exchangers that impacts the size of the heat exchangers. The size and type of the heat exchangers are the major parameters for determining their cost. The heat transfer area in the heat exchangers depends on the heat rate exchange between the hot springs (temperature and mass flow rate) and the working fluid (heat capacity) in the ORC energy plants. For example, a hot spring with a water discharge mass flow rate of 5 kg/s at 60 ^{o}C needs a shell and tube heat exchanger as an evaporator with a 20 m^2 heat transfer area to exchange heat with the working fluid (R227ea) at 1.8 kg/s of mass flow. Similarly, a hot spring with a 50 kg/s water discharge mass flow rate at 90 ^{o}C requires a shell and tube heat exchanger as an evaporator with a 230 m^2 area for exchanging the energy with the working fluid (R227ea) at 34.2 kg/s of mass flow. It must be noted that the heat exchange area has been calculated for the clean condition of heat exchangers while for more accurate results the fouling factors of the hot springs and working fluids must be considered. The results in Figure 4.9 illustrate that how the total heat transfer area in heat exchangers depends on the hot spring discharge mass flow rate and its temperature. It can be seen that the evaporator's total heat transfer area also depends on the working fluid pressure level.

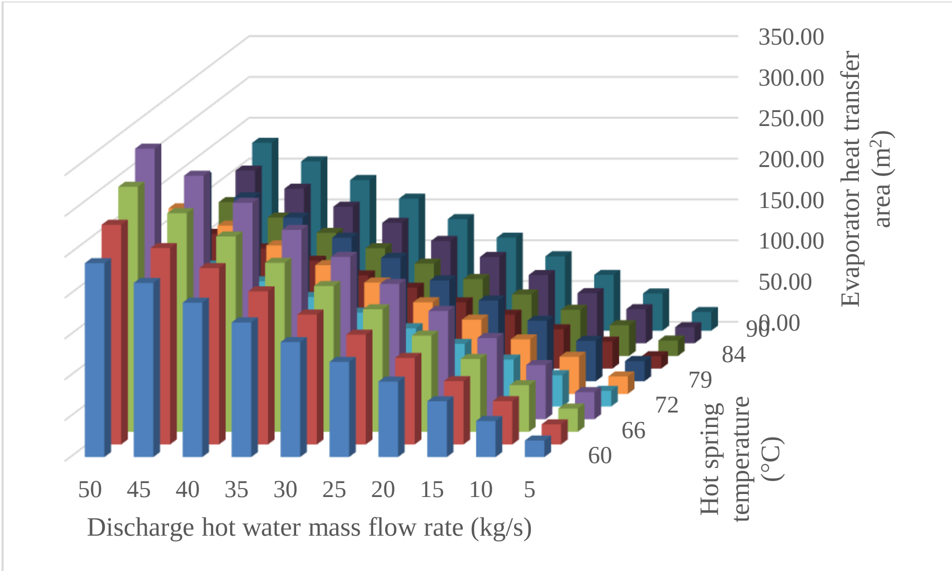

Figure 4. 9: The heat exchanger total heat transfer area for different hot springs

4.7.3. Economic Character

The SIC has a substantial role in the expansion of the geothermal energy plant. SIC value for the existing geothermal ORC plants is between 1300 \$/kW for a 1 MW plant to 5200 \$/kW for a 4 MW power generation [222]. The SIC for the ORC plant with 60 ^{o}C hot water temperature and 5 kg/s hot water mass flow rate is estimated at 9488 \$/kW, while 90 ^{o}C hot springs with 50 kg/s mass flow rate are approximated for1869 \$/kW as illustrated in Figure 4.10. Hot springs with higher temperatures and larger water discharge mass flow rates are financially more viable because of the lower cost of the energy plant per amount of generated power. It can be concluded that hot springs with a higher temperature and larger water mass flow rate are more efficient economically.

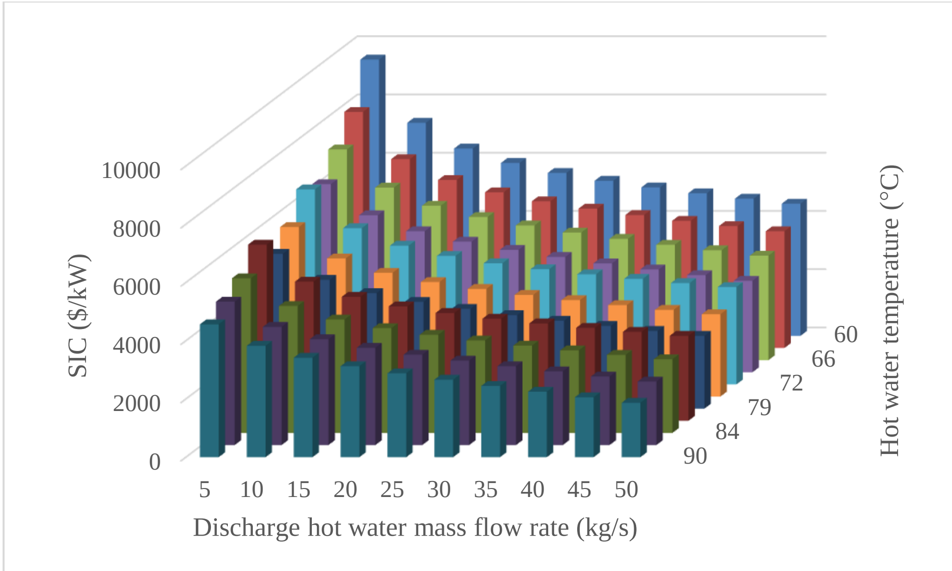

Figure 4. 10:the SIC value for different hot springs

The temperature and mass flow rate of the hot springs affects the LEC of the energy plant. The average LEC for a geothermal power plant is about 0.080 $/kWh [223]. LEC shrinks by increasing the hot spring discharge water mass flow as shown in Figure 4.11. The LEC value also reduces by increasing the temperature of hot spring water as illustrates in Figure 4.11. For example, in a hot spring with 90 °C temperature, the LEC decreases from 0.052 $/kWh to 0.021 $/kWh when its water mass flow rate increases from 5 kg/s to 50 kg/s. Therefore, harnessing the energy of hot springs in an energy plant with a higher water mass flow rate and temperature leads to a smaller LEC value. The deep-well geothermal power plant has a lower LEC since some costs such as drilling cost in hot spring energy plant are eliminated.

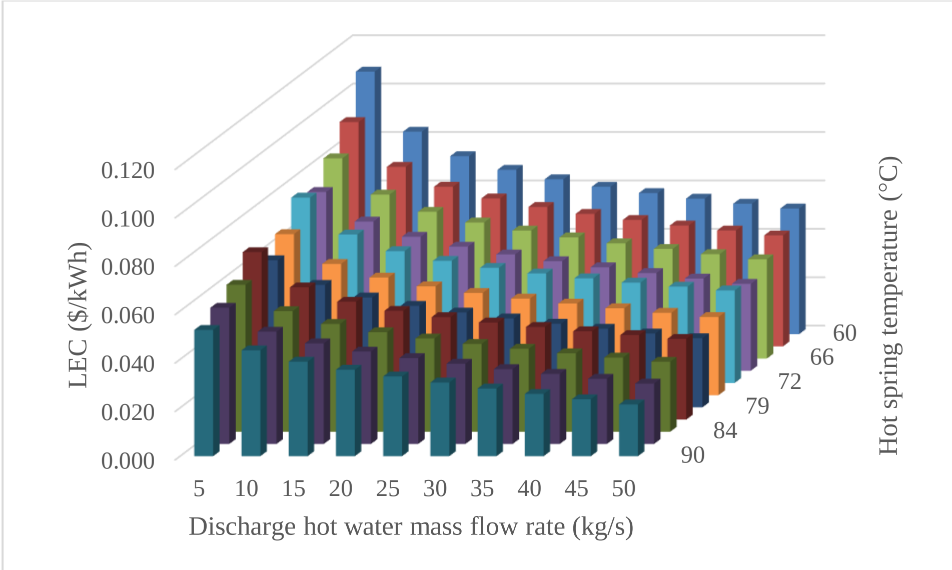

Figure 4. 11:LEC value for hot springs with different mass flow rates and temperatures

The average PP for geothermal plants is 6 years [223], although the temperature and discharge mass flow rate of the hot springs greatly impacts this norm. The impacts of the temperature and discharge mass flow rate of hot springs with an electricity cost of 0.10 $/kWh are from 2.99 to 15.19 years for PP as presented in Figure 4.12. The longest estimated PP is 15.19 years for a hot spring with 60 °C temperature and 5 kg/s discharge mass flowrate. The shortest PP is 2.99 years for a hot spring with 90 °C temperature and 50 kg/s discharge mass flow rate.

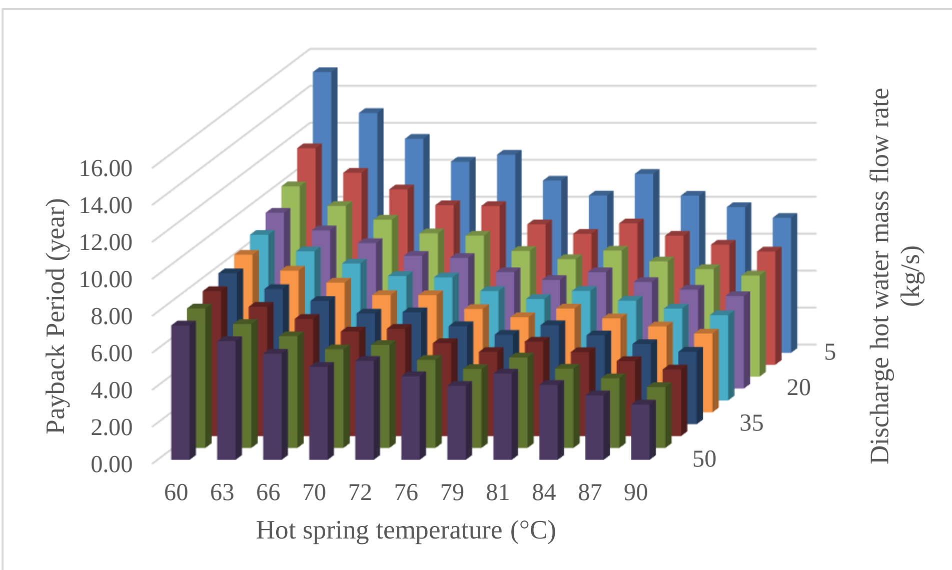

Figure 4. 12:The impact of hot water temperature and mass flow rate on the Payback Period

Figure 4.13 clarifies the range of temperature and discharge mass flow rate in which the PP of the energy plant is more than 6 years. It can be observed that for the lower discharge mass flow rate and lower temperature hot springs, the PP is higher than 6 years and for hot springs with higher temperature and higher mass flow rate, PP is less than 6 years. It makes the hot springs with higher temperatures and/or higher discharge mass flow rate ideal candidates for establishing the energy plants.

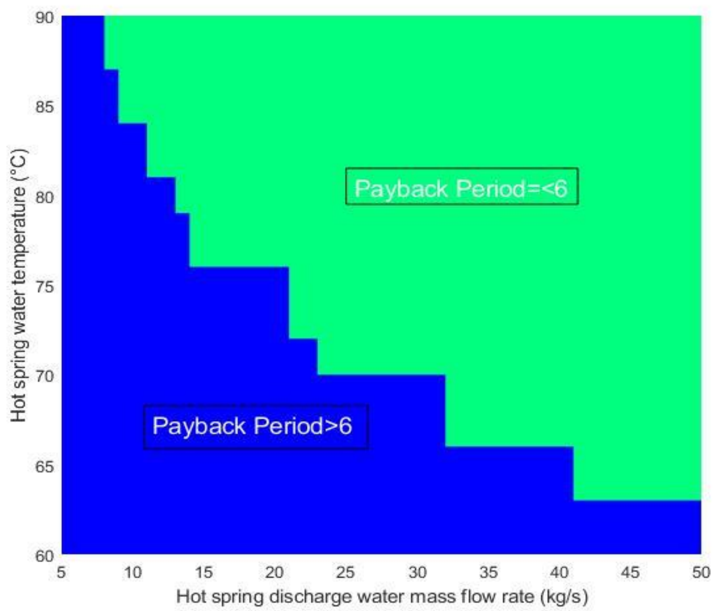

Figure 4. 13: The energy plant PP in different hot spring temperature and water discharge mass flow rate

The cost of the electricity has an impact on the PP for hot springs energy plants. The electricity cost varies based on location, for example in the US, it changes between 0.04 to 0.24 $/kWh. To show the impact of the electricity cost Figure 4.14 is prepared for a hot spring with a discharge mass flow rate of 30 kg/s with various temperatures of 60, 76, and 90 °C. It can be observed that by increasing the price of electricity from 0.06 to 0.16 $/kWh, the PP shrinks to 5.57, 3.65, and 2.78 years respectively. The high electricity price and high temperature of hot springs support the lower PP for the hot spring energy plants.

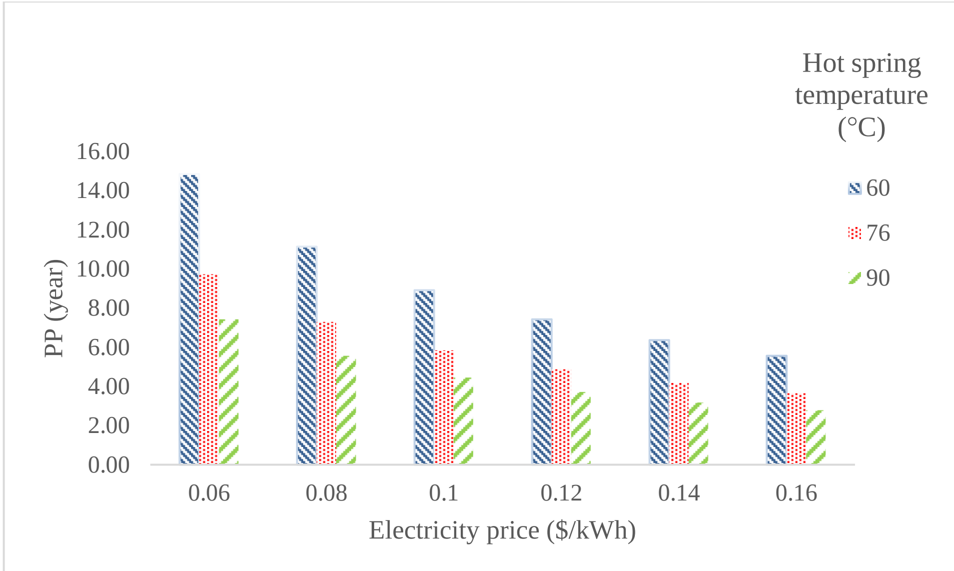

Figure 4. 14:The impact of electricity price on the Payback Period

4.8. Validation

To validate the outcomes of five power generations with ORC geothermal energy plants in different parts of the globe are compared with the power generation estimation through the present study. The actual outcomes of the past studies besides the present study estimations are illustrated in Table 4.6. In the first study the hot spring was located in Indonesia, a double-stage ORC was employed with a different working fluid therefore the output power was slightly more than the present study estimation. An American hot spring was the second study, in which a condenser with 4.4 ^{o}C water-medium and R134a as working fluid in ORC were used, while the condenser of the present study designed with 18 ^{o}C air-medium and R227ea as working fluid. Consequently, the second case study could capture more energy in ORC and showed a higher output power and energy efficiency compared with the present study. For the third study in Taiwan, there was a slight difference in the power output and thermal efficiency between the presented guideline and the outcome of the study. It was based on the difference in the working fluid types. The fourth study was in Poland, since the hot water mass flow rate was one-third of the present study, the power output was lower almost with the same ratio in comparison with the present study. In the fifth study which was located in China, the difference between the hot spring temperature, mass flow rate, and the working fluid with the present study data has made the output differences. More details about the hot springs in those studies are presented in Table 4.6.

Table 4. 6: Validation of the model with the previously published data

Case number	Application	Hot source temperature (°C)	Hot source mass flow rate (kg/s)	Working fluid	Net power output (kW)	Thermal efficiency (%)	Reference /Location
1	Geothermal power generation plant	62.2	4	R123yf	15.7	-	[224] Indonesia
	Present Study	63	5	R227ea	13.5	4.2	
2	Geothermal power generation plant	73	35	R134a	210	8.2	[170] USA
	Present Study	73	35	R227ea	110.8	5.9	
3	Geothermal power generation plant	80.4	7.5	R245fa	16.8	6	[225] Taiwan
	Present Study	81	8	R227ea	24.4	7.2	
4	Geothermal power generation plant	88	1.7	R227ea	9	7.8	[213] Poland
	Present study	87	5	R227ea	26	7.2	
5	Geothermal power generation plant	110	55	R245fa	500	-	[226] China
	Present Study	90	50	R227ea	303	7.2	

The overall comparison shows in past studies the outcomes are close enough for validation of the presented guideline. It should be noted that a present study is a tool for "estimation" of power generation capacities and other economic and environmental characteristics by considering the mass flow rate and temperature of the hot spring for an ORC energy plant when the working fluid is R227ea. There are many variables such as working fluid type, evaporator and condenser pressure, cold source medium and its temperature, and the efficiency of the ORC components that impact the outcomes. Consequently, the proposed guideline is an estimation tool not a sizing or designing approach.

4.9. Conclusion

Renewable power generation plants with lower environmental impact support energy sustainability. Hot springs as a source of heat and a natural energy storage can be used for electricity generation under the umbrella of renewable power generation. Having a techno-enviro-economical tool is greatly beneficial to approximate the technical, economic, and environmental aspects of power plant projects including power generation capacity, initial investment, and possible income from the power plant. The present study provides a guideline for estimation of the power generation capacity for hot springs with various temperatures and mass flow rates while the economic aspects of the power generation can be predicted in terms of PP and LEC. The proposed tool in this study is capable to estimate the environmental business cost based on the carbon tax, electricity cost, and tax benefit for different projects depend on the location of the project and that will be compared with similar power generation plants with fossil fuels.

ORC is the most suitable cycle for power generating in the hot spring with a low-temperature energy source. Therefore, in this study a thermodynamic model of the power generation cycle has modeled and simulated in Aspen Plus V9. The proposed guideline covers the hot springs temperature in the range of 60 °C to 90 °C and discharges a mass flow rate of 5 kg/s to 50 kg/s. The power generation capacity for those range of temperature and mass flow rate is from 9.3 kW to 303 kW depends on the temperature and mass flow rate of the hot spring. Study about the developed technical model for the ORC energy plants reveals:

- The hot springs with a larger discharge water mass flow and a higher temperature result in the higher power generation capacity which are economically more profitable,
- The temperature of hot spring and working fluid heat capacity affect the energy plant thermal efficiency,
- The hot springs with a larger discharge water mass flow and a higher temperature need larger and more costly heat exchangers.

The investigation around the proposed economic model with the capability of implementing on a wide range of hot springs shows:

- The increase of electricity price makes the hot spring energy plant projects more desirable economically,
- The hot springs with a larger discharge water mass flow and a higher temperature lead to a smaller LEC which makes the power generation plant projects more profitable.

It can be concluded that the hot springs with higher temperatures and/or larger discharge mass flow rates are financially ideal candidates for launching the power generation plant projects. The common practice of power generation by fossil fuel affects the environment immensely. Replacing geothermal energy plants with the old fashion power generation plant when it is possible, is an effective solution to reduce air pollutions.

Hot spring temperature and discharge mass flow rate are essential parameters in power generation rate in power generation plants. The presented guideline assists engineers and investors in estimation of the power generation capacity and efficiency of possible plant projects for any hot springs. The economic and environmental aspects of the energy plant can be approximated as well. Essentially the

Techno-Enviro-Economic assessment is a quick tool for a preliminary feasibility study of geothermal power plant for any hot springs.

Credit authorship contribution statement

Saeed Ghoddousi: Conceptualization, Methodology, Software, Formal analysis, Investigation, Resources, Data curation, Writing - original draft, Visualization. Behnaz Rezaie: Conceptualization, Methodology, Validation, Writing - review & editing, Supervision, Project administration. Samane Ghandehariun: Writing - review & editing.

Chapter 5: Conclusion

5.1 Conclusion

This thesis has presented novel approaches to save water and energy in cooling towers and power generating from a low-temperature renewable energy resource regarding water-energy nexus. Each study has discussed each application technically, economically, and environmentally to improve its performances, water usage, energy consumption and generation, and cots-effectiveness. The analysis shows the possibility of improving the efficiency of the designed systems that sometimes hiddenly wastes resources.

In addition to saving water and energy resources, the financial aspect of the plan needs to be considered as a key factor for investors that motivates them to implement the proposed tool. It has been shown that economic conditions play an important role in implementing any energy system improvement plan. Interestingly, the cooling tower improvement approach does not need any investment and can make benefits continuously over its operation time. This fact shows that cost saving without any expenses is applicable by proper modeling and modifying the cooling tower, to save resources.

While developing renewable energy systems has become a goal in the present time, the financial matter is also a key aspect for making decisions. Proposing a new source for generating power on a small scale has been done in the present thesis to reduce the investment cost for performing the energy plant. This thesis clarified the financial aspect of implanting new energy plants with real industry quotes provide a practical guideline for investors and researchers to develop renewable energy systems in the future.

The present thesis elaborated on the roles of cooling towers in cooling cycles while narrating the history of water-energy modeling methods of cooling towers through the Markel, the Poppe, Braun, and the Effectiveness–(NTU) Models since water and energy are deeply connected in cooling towers. The major classifications and configurations of cooling towers showed the wet cooling towers consume the highest water while dry cooling towers use the minimum water in a trade-off with performance. The hybrid cooling towers have the advantages of both while they are more costly. There are some practical approaches to reducing the environmental impact of cooling towers, such as using VFD, adding a dry section, and employing water filtration to reduce water and energy consumption.

Applying an improvement approach on the existed mechanical draft wet cooling tower at the UI, Moscow campus by using DAQ and ambient air condition actual data has been shown in the present thesis. It has been shown that instead of working 100% load of cooling tower, the operation must be performed by considering ambient air conditions. Operating 100% load of cooling tower wasted a considerable amount of water and energy and provided extra cooling for the selected HVAC system. By adjusting outlet water temperature in the cooling tower developed model, the cooling towers' load can be adjusted, reducing its water and energy consumption. Consequently, the improvement plan reduces cooling tower fans' energy consumption and water loss due to evaporation. For one month of operation of the improved technique, up to 28035.50 kWh energy saving, 179.49 m3 of water saving, and $1125.17 cost savings have been recorded.

Hot spring as a low-temperature geothermal energy source has been investigated in the present thesis. A techno-enviro-economical tool has been developed to approximate the technical, economic, and environmental aspects of hot spring power plant projects, including power generation capacity, initial investment, and possible income from the power plant. The proposed tool in this study can estimate the environmental business cost based on the carbon tax, electricity cost, and tax benefit for different projects depending on the location of the project. The proposed guideline covered the hot springs temperature in the range of 60 °C to 90 °C and discharged mass flow rate of 5 kg/s to 50 kg/s. The power generation capacity for those ranges of temperature and mass flow rate has been calculated from 9.3 kW to 303 kW depending on the hot spring's temperature and mass flow rate. It has been shown that hot springs with a larger discharge water mass flow and a higher temperature result in higher power generation capacity, which is economically more profitable. The payback period of investing on a hot spring energy plant can be as low as six years, which is better cost-effective than other geothermal energy plants. In addition, replacing hot spring energy plants with the old-fashioned power plant is an effective solution for reducing air pollutions.

5.2 Future Study

Since this thesis has focused on applications that have not gotten much attention before, significant amounts of additional research and development can continue towards the water-energy nexus.

Implementing cooling towers in different industries that inefficiently work and hiddenly consume water and energy needs to be stopped for resource conservation. This water-intensive application can be improved in different ways to work more efficiently, which can make a path for future works. Proposing the new auxiliary equipment, finding methods of decreasing make-up water, and optimizing the energy models are the major paths for future investigations to reduce energy consumption in cooling towers. Similarly, capturing evaporated water from cooling towers before releasing to ambient air using novel material such as nanomaterial, improving the heat transfer parameters in cooling towers, savings through the pumps for delivering water from the tower to the facility and its related system improvement, findings new make-up water resources instead of fresh-water usage, and implementing new dry cooling towers to the present water-intensive cooling tower are potential investigations for water and energy reduction opportunities in the light of water-energy nexus in cooling towers.

Since hot springs are distributed worldwide in different climates and geographical conditions, applying the provided guideline on the real hot springs in different areas could be great potential for future research. While implementing power generation plants from low-temperature energy resources like hot springs has not been performed frequently, developing new studies focusing on improving the performance of ORC equipment such as heat exchangers, pumps, condensers, and expanders for low-temperature energy sources is another area that needs attention in the future.

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