www.czasopisma.pan.pl PAN www.journals.pan.pl
POLSKA AKADEMIA NAUK

ARCHIVE OF MECHANICAL ENGINEERING

DOI: 10.24425/ame.2022.144076 2023, Vol. 70, No. 1, pp. 129–149

Mohamad Hasan MALEKMOHAMADI[1], Hossein AHMADIKIA[2], Siavash GOLMOHAMADI[3], Hamed KHODADADI[4]

Evaluation of wet cooling tower replacement by Heller cooling tower in a power plant

Received 23 July 2022, **Revised** 27 September 2022, **Accepted** 28 November 2022, **Published online** 8 December 2022

Keywords: Heller cooling tower, wet cooling tower, condenser, power plant

Water resources are the main component of natural systems affected by climate change in the Middle East. Due to a lack of water, steam power plants that use wet cooling towers have inevitably reduced their output power. This article investigates the replacement of wet cooling towers in Isfahan Thermal Power Plant (ITPP) with Heller natural dry draft cooling towers. The thermodynamic cycle of ITPP is simulated and the effect of condenser temperature on efficiency and output power of ITPP is evaluated. For various reasons, the possibility of installing the Heller tower without increasing in condenser temperature and without changing the existing components of the power plant was rejected. The results show an increase in the condenser temperature by removing the last row blades of the low-pressure turbine. However, by replacing the cooling tower without removing the blades of the last row of the turbine, the output power and efficiency of the power plant have decreased about 12.4 MW and 1.68 percent, respectively.

Nomenclature

Symbols

A_{face}	front surface of heat exchanger	C_r	ratio of heat capacity
C	heat capacity	d	diameter
c_p	specific heat capacity	e	roughness

✉ Hossein Ahmadikia, e-mail: ahmadikia@eng.ui.ac.ir
[1]University of Isfahan and Isfahan Thermal Power Plant, Isfahan, Iran
[2]University of Isfahan, Isfahan, Iran; ORCID: 0000-0003-2167-5237
[3]Isfahan Thermal Power Plant, Isfahan, Iran
[4]Department of Electrical Engineering, Khomeinishahr Branch, Islamic Azad University, Isfahan, Iran

f	friction
G	velocity of mass flow rate
g	coefficient of gravity
H	height
h	coefficient of convection heat transfer
$\hat{h}$	enthalpy
h_l	head loss
k	coefficient of conduction heat transfer
L	length
m	mass flow
N	number
NTU	number of heat transfer unit
P	pressure
pass	number of passes in heat exchanger
Q	heat
q	heat transfer
Re	Reynolds number
RH	relative humidity
R_j	other thermal resistance
T	temperature
U	overall heat transfer in the area
W	work

Greek symbols

Δ	difference
δ	thickness
ε	efficiency
η	efficiency
θ	angle
κ	coefficient without dimension
ρ	density

Subscripts

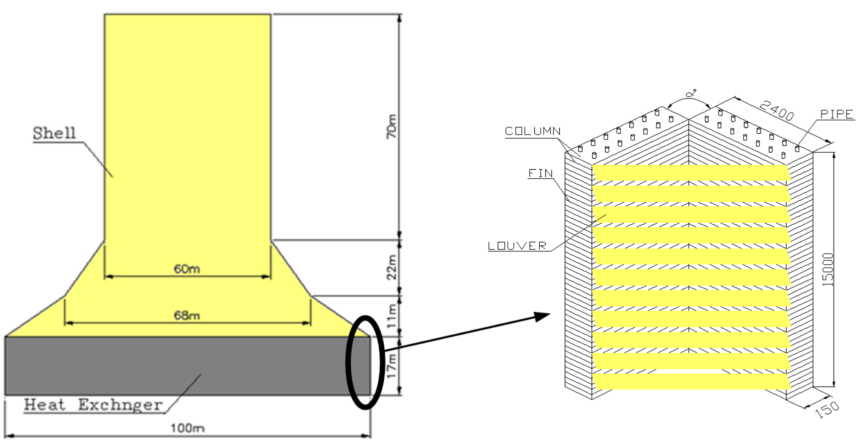
a	air
i	inner or inlet
in	inlet
max	maximum
min	minimum
o	outer or outlet
out	outlet
T	total
t	tube
w	water

1. Introduction

In recent years, climate change and global warming have caused irreparable damage to some countries. Studies have shown that in the Middle East and North Africa average temperature has increased and rainfall has decreased. The water crisis and the drought in Isfahan City, in the central area of Iran, have made the water supply scarce throughout the country. Dziegielewski and Baumann [1] state that valid long-term forecasts of water demand are essential to all types of water-related planning. Without such forecasts, water planners cannot efficiently allocate water resources among competing uses or ensure long-term sustainability. For power plant cooling systems, optimizing and retrofitting the performance of the cooling tower reduces the power plant's internal electricity consumption and water consumption [2]. Isfahan Thermal Power Plant (ITPP) is a natural gas-fired steam power plant with five units and total output power of 835 MW, which uses wet cooling towers. The water consumption of the cooling tower is 2000 m^3/h. Due to the lack of water, it is necessary to adopt methods to reduce the water consumption of the power plant in large volume. For steam power plants, a well-known solution is to change the wet cooling tower to a dry one, which reduces water consumption by more than 95%. In addition, the fog of wet cooling towers and its combination with ITPP combustion products cause acidic rain, which must be avoided and prevented (see Fig. 1). Also, the blowdown of wet towers causes environmental pollution. These problems can be resolved by changing the wet to dry system and installing the Heller tower.

Fig. 1. Wet tower of ITPP and their output plume

One of the indirect dry cooling systems is the Heller cooling tower. This system was first presented in 1965 by Professor Heller at the University of Budapest, Hungary. The heat of the water leaving of the condenser of the power plant is absorbed by the ambient air passing through a number of compact heat exchangers called the Heller-Forgo in the Heller cooling tower. These heat exchangers are in the form of a delta with a vertical position in the circular environment around the cooling tower. The deltas are made of a special type of heat exchanger called Forgo, which consist of 15-meter columns stacked on top of each other. Fig. 2 shows the Heller tower and a view of a delta. The air flow rate is controlled by louvers. The Heller cooling tower does not need compensation water, so it does not



(a) Heller cooling tower (b) Forgo heat exchanger

Fig. 2. Heller cooling tower and delta heat exchanger

pollute the environment and also does not need a fan. Of course, one of the major disadvantages of the high Heller tower is its inefficiency in high wind speeds and high ambient air temperature.

Due to the retrofitting of the cooling tower and the installation of a new tower, many parts of the existing cooling system must be replaced. New circulating water pumps are required to provide additional heat. Therefore, the effect of changing the pumps and circulating water piping system on other parts must be evaluated. Many studies and researches have been done to change the design of the cooling system of various power plants. Burns et al. [3] discussed the retrofit cost estimation of an alternative cooling system. They described the characteristics of the cooling tower, circulating water pumps and pipes, and the changes required in the condenser. Loew et al. [4] investigated the cost of retrofitting combined cycle power plants in Texas with alternative cooling systems, either hybrid towers or air-cooled condensers. They determined the projected annual reductions in water withdrawals and the cost per gallon of water saved by the retrofit. They concluded that replacing the once-through cooling with wet recirculating towers has a lower cost per reduced water withdrawal.

Conradie and Kroger [5] presented the modeling equations of the Heller cooling tower, including mass, momentum, energy, and experimental relations. In another study, they evaluated the optimal performance of the Heller cooling tower in its economic concept and studied the dimensional changes and size of heat exchangers used in the Heller towers [6]. Buyz and Kroger [7] studied the improvement of the shape, dimensions and size of Heller towers. Zou et al. [8] presented a new concept called solar enhanced natural draft dry cooling towers, in which solar collectors are added to traditional natural draft dry cooling towers to increase their performance. Bagheri and Nikkhoo [9] have optimized the location of two new Heller towers that will be added to the Shazand Power Plant (SPP). In their study, the air flow inside and around the old and new towers has been studied numerically, and the best optimal location for the constructing two new towers has been suggested according to wind direction and speed.

In past years, studies that have been done about Heller tower are usually about improving its performance when the weather is hot, or windy [10–13]. Since different weather conditions, especially the ambient temperature, have a great effect on the performance of a dry cooling tower, then the condenser vacuum is also highly dependent on weather changes. Under the influence of this change in condenser vacuum, the amount of output power and thermal efficiency of the power plant will change. Therefore, it is necessary to evaluate the effect of condenser vacuum on the performance of the steam power plant. By performing such an evaluation, the effect of changing the wet tower to the dry tower on the thermodynamic cycle of the power plant will be determined. Alizadeh et al. [14] have evaluated the performance of Heller tower of the Shahid Montazeri Power Plant (SMPP) in different weather conditions for condenser vacuum, output power and thermal efficiency of that power plant. Jahangiri and Rahmani [15] have numerically modeled the Heller tower of SMPP. They studied the effects of wind and ambient temperature on the performance of the Heller tower and the thermodynamic cycle of the power plant. According to them, the output power is reduced by 25% in high wind speeds and ambient temperatures. Samani [16] presented a model for a combined cycle power plant with a Heller dry cooling tower. He checked the output of the power plant in different operating conditions.

The above studies show that replacing a wet cooling tower with Heller requires a lot of knowledge and is more difficult than designing a Heller tower for a new power plant. Changes in turbine, condenser and Heller system installation are some of the challenging aspects of cooling tower replacement. Here, the design of the dry Heller tower and the changes of some components of the power plant are studied, and their qualitative and quantitative effects on the thermal efficiency and output power of the power plant are investigated. In this study, a unit of ITPP with a nominal capacity of 320 MW is modeled according to its real data and at the rated load. The condenser of this unit is of the surface type and a wet cooling tower is designed and installed to dissipate heat from it. Changing each part of the power plant makes it possible to achieve changes in efficiency, fuel consumption and output power. In this study, due to the replacement of the cooling tower, it is possible to determine any changes in the characteristics of the power plant. Also, the replacement of the wet cooling tower with natural draft cooling tower is discussed.

2. Heller cooling tower analysis

In the Heller cooling tower design, the temperature of the air and water entering the heat exchanger are known. Here, using ε – *the NTU* method for analyzing the heat exchanger is preferable to the logarithmic thermal difference method.

2.1. Heat transfer analysis of Heller cooling tower

The maximum allowable heat is obtained through a counter-flow heat exchanger with the maximum allowable thermal change $((T_{w,\mathrm{in}}–T_{a,\mathrm{in}})$. The maximum heat exchange q_{max} and the actual heat exchange q are obtained through the following equations:

$$q_{\max} = C_{\min}\left(T_{w,\mathrm{in}} - T_{a,\mathrm{in}}\right), \tag{1}$$

$$q = \dot{m}_w c_{pw}\left(T_{w,\mathrm{in}} - T_{w,\mathrm{out}}\right) = \dot{m}_a c_{pw}\left(T_{a,\mathrm{out}} - T_{a,\mathrm{in}}\right), \tag{2}$$

$$\varepsilon = q / q_{\max}, \tag{3}$$

where ε, the coefficient of performance defined in the actual heat exchange, is the maximum allowed heat exchange ratio. From Eqs. (1) and (3), the following equation is the result:

$$q = \varepsilon C_{\min}\left(T_{w,\mathrm{in}} - T_{a,\mathrm{in}}\right). \tag{4}$$

The number of transfer units, NTU, is a dimensionless parameter given as:

$$NTU = UA / C_{\min}\,, \tag{5}$$

where UA is the total heat transfer coefficient of the heat exchanger. The following equation, specific to heat exchangers with finned pipes and cross-sectional flow, can be used in the design of heat exchangers in Heller cooling towers [17].

$$\varepsilon = 1 - \exp\left[NTU^{0.22}\,\frac{\exp\left(-C_r\,NTU^{0.78}\right) - 1}{C_r}\right], \tag{6}$$

$$\varepsilon = C_{\min} / C_{\max}\,. \tag{7}$$

The Forgo heat exchanger is used in Heller cooling towers. Louver dampers are installed to regulate the flow of air passing through the heat exchanger. The dependence of the total heat transfer coefficient on the heat transfer coefficient inside and outside the pipe is presented as [6]:

$$\frac{1}{UA} = \frac{1}{\eta_o h_o A_o} + \frac{1}{h_i A_i} + \frac{\delta_t}{k_t A_t} + R_j\,. \tag{8}$$

To have a simple equation, Eq. (8) is rewritten as:

$$\frac{1}{U} = \frac{1}{U_i} + \frac{1}{U_o}\,, \tag{9}$$

where, U_i and U_o are determined by the following experimental relations [18]:

$$U_i = \left(319 + 5.67\frac{T_{w,\text{in}} + T_{w,\text{out}}}{2}\right)\dot{m}_w^{0.8}\,, \tag{10}$$

$$U_o = 1180\left[\frac{\dot{m}_a}{A_{\text{face}}}\left(\frac{2\rho_{0,a}}{\rho_{a,\text{in}} + \rho_{a,\text{out}}}\right)^{0.64}\right]^{0.515}. \tag{11}$$

According to Fig. 3, the specifications of the Forgo type single column heat exchanger are listed in Table 1.

Table 1. Specifications of Forgo heat exchanger

Parameter	Value	Parameter	Value
Width (m)	2.4	Free surface of passing air (m^2)	17.5
Height (m)	15	Fin thickness (mm)	0.33
Depth of airflow (mm)	150	Fin distance (mm)	2.8
Front surface of heat exchanger (m^2)	34.5	Number of tubes	240
Inner tubes diameter (mm)	17.1	Number of passes	2
Outer tubes diameter (mm)	18.25	Number of rows	6

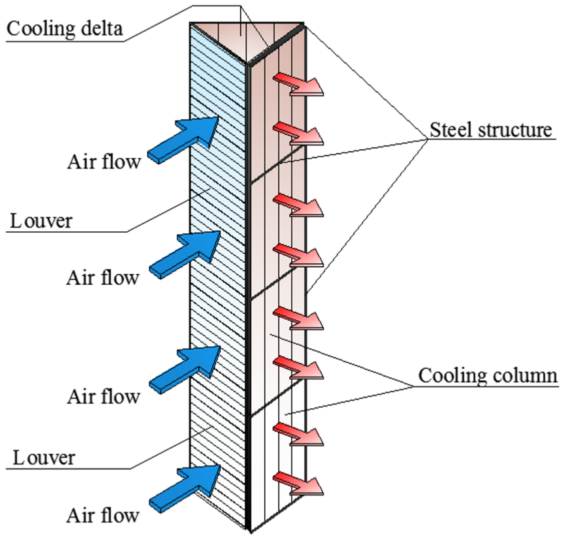


Fig. 3. Forgo heat exchanger

2.2. The Heller tower

The air enters the tower and its upward movement is due to buoyancy forces caused by its heat gain in the heat exchanger. The air suction through the tower is calculated by [9, 18, 20]:

$$\Delta P = gH\Delta\rho_a\,, \tag{12}$$

where, $\Delta\rho_a$ is the air mass density difference between the inside and outside of the Heller tower. The tower must be tall enough to compensate for any possible resistance to air flow. Therefore, the height of the tower is calculated by:

$$\Delta P = \Delta P_{\text{delta}} + \Delta P_{\text{louver}} + \Delta P_{\text{exit}} = gH\Delta\rho_a\,, \tag{13}$$

in which, ΔP_{delta} is the air pressure drop in passing through the deltas around the tower, ΔP_{louver} the air pressure drop in the louvers, and ΔP_{exit} is the air pressure drop exiting the tower shell, which are determined based on experimental relations [18–20]:

$$\Delta P_{\text{louver}} = 0.00548\left(\frac{\dot{m}_a}{A_{\text{face}}}C_k^{0.5}\right)^2, \tag{14}$$

$$\Delta P_{\text{delta}} = \left[0.147 + 0.007\cot^2\left(\frac{\alpha}{2}\right)\right]\left(\frac{\dot{m}_a}{A_{\text{face}}}C_k^{0.5}\right)^{1.76}, \tag{15}$$

$$\Delta P_{\text{exit}} = \frac{\rho_{\text{air,mean}}}{2g}\left[\frac{G_d}{3.6\rho_{\text{air,mean}}}\,\frac{4}{\pi\left(D_2-2\right)^2}\right]^2, \tag{16}$$

where, θ is the delta angle and $C_k = \rho_{a,0}/\rho_{a,\text{mean}}$ where $\rho_{a,0}$ and $\rho_{a,\text{mean}}$ are the standard conditions and average air density, respectively. G_d is the total discharge of air passing through the deltas in tons per hour and D_{out} is the diameter of the outlet of the tower.

The Forgo heat exchangers are placed next to each other with a 40° to 60° delta angle. The diameter of the tower base is determined according to the degree between the delta columns and the number of heat exchangers installed in a delta shape around the tower. In practice, some distance is allowed among the heat exchangers for expansion and contraction, and this automatically increases the diameter of the tower base. The base diameter is calculated as follows (see Fig. 4):

$$D_{\text{base}} = 2\kappa \left[\frac{2.4\sin(\theta/2) + 0.15\cos(\theta/2)}{\tan\beta} + 0.15\sin(\theta/2) \right], \tag{17}$$

$$\beta = 360/N_{\text{delta}} \,. \tag{18}$$

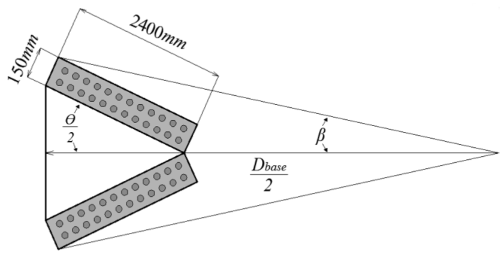


Fig. 4. Delta-type heat exchanger

The diameter of the concrete foundation of the tower is about 1.5 times the diameter of the base of the tower, which must be considered according to the plant site. From the balance of forces, it is concluded that the diameter of the base should be at least 65% of the height and at most equal to the height of the tower [21].

A computer program is developed based on the Heller tower simulation design and the data available at SMPP and SPP according to the data presented in Table 2. Table 3 compares the output data with the results obtained from the two mentioned

Table 2. SMPP and SPP Heller towers data

Parameter	SMPP	SPP
Nominal production (MW)	200	325
Circulating water (m^3/h)	25200	34000
Entrance water temperature (°C)	58	60
Exit water temperature (°C)	48	50
Air humidity (%)	40	35
Entrance air temperature (°C)	30	30
Entrance air pressure (Pa)	86813	81810
Between delta angle (degree)	60	49.03

power plants. Comparing the obtained values with real data validates the developed computer program.

Table 3. Comparison of calculated and real data of SMPP and SPP Heller towers

Parameter	SMPP		SPP	
	calculated	real	calculated	real
Number of heat exchangers	238	238	268	264
Tower base diameter (m)	105.91	109	109.73	110
Upper opening diameter (m)	62.07	62	61.83	62
Tower height (m)	121.32	120	150.12	150

3. Thermodynamic simulation of power plant

The ITPP configuration is shown in Fig. 5. The outlet pressure and temperature of the ITPP boiler are 170 kg/cm^2 and 540°C, respectively. The steam flow rate of the boiler is 1056 Ton/hr. The power plant has six closed heaters, one open heater, and one boiler with a reheating system. The turbines here include one in high pressure, one in medium and one in low pressure. The condenser is also surface type. There is also a flow meter at points 1, 13, and 21 of Fig. 5.

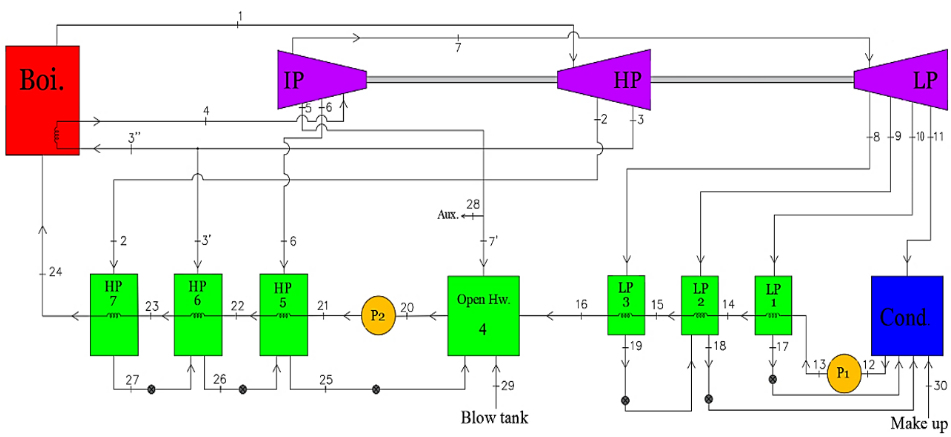


Fig. 5. The schematic of ITPP [22]

Using the law of conservation of mass and thermodynamic laws, the thermodynamic simulation of the power plant has been done. With every change in power plant parts, output power, efficiency, fuel consumption and other characteristics are calculated. Therefore, according to the replacement of the cooling tower and the change of the condenser temperature, the change of the thermal efficiency and power output of the power plant will be determined. The values of the necessary parameters at different points of the power plant cycle and nominal load are listed in Table 4, which are in accordance with the technical documents of the power plant.

Table 4. Thermodynamic characteristics and mass flow rate of different points of the power plant according to Fig. 4 at nominal load

Number	Pressure (MPa)	Temperature (°C)	Enthalpy (kJ/kg)	Flow (kg/s)
1	17.462	540	3395	293.33
2	7.823	428.5	3196	37.56
3	3.708	327.4	3034.65	255.78
3’	3.708	327	3034.64	26
3”	3.708	327	3034.64	229.8
4	3.414	540	3543	229.8
5	1.609	434	3111	19.79
6	0.632	319.8	3334	8.21
7	0.632	319.8	3111	197.4
7’	0.632	319.8	3111	9.23
8	0.164	212.4	2852	8.21
9	0.0745	74.56	2728.6	8.13
10	0.0294	68.65	2602	8.6
11	0.0095	44.75	2447	172.5
12	0.0096	44.85	188.4	205.9
13	1.464	46.52	190.7	205.9
14	1.65	66.3	287.3	205.9
15	1.544	74	383.7	205.9
16	1.29	125.9	478.4	205.9
17	0.0294	68.6	287.3	8.6
18	0.0745	71.6	383.7	16.34
19	0.156	94.4	478.4	8.21
20	0.6102	165.4	679.4	293.33
21	20.01	167.7	705.1	293.33
22	19.74	203	855.4	293.33
23	16.66	245.7	1061	293.33
24	19.37	292	1303	293.33
25	1.558	172.5	859.9	76.09
26	3.56	207.9	1066	63.56
27	7.58	251	1308	37.56
28	0.632	319.8	3111	10.56
29	0.618	166	2772	2.14
30	0.098	34	142.5	8.42

The temperature and pressure of different points of the cycle according to Fig. 4 and Table 4, as well as the flow rate of points 1, 13 and 21 (where there is a flow meter in these three points), are the input parameters of the simulation program. The control system automatically adjusts the air flow and fuel flow rate. The thermodynamic properties of all cycle points are shown in Table 4. Continuity and energy laws in the various components of the power plant cycle and empirical relationships are used by Engineering Equation Solver (EES) software. In order to validate the simulation, parameters such as pressure, temperature and flow at different points of the cycle and turbines output power, pumps consumption, etc., at different loads have been matched with real values like ITPP. According to Fig. 5, the energy flow diagram at nominal load is shown in Fig. 6. This diagram shows the flow rate at the nominal load. In Fig. 6, the amount of heat given to the boiler is the heat absorbed by the water and steam inside the boiler, and was calculated using the laws of conservation of energy and mass.

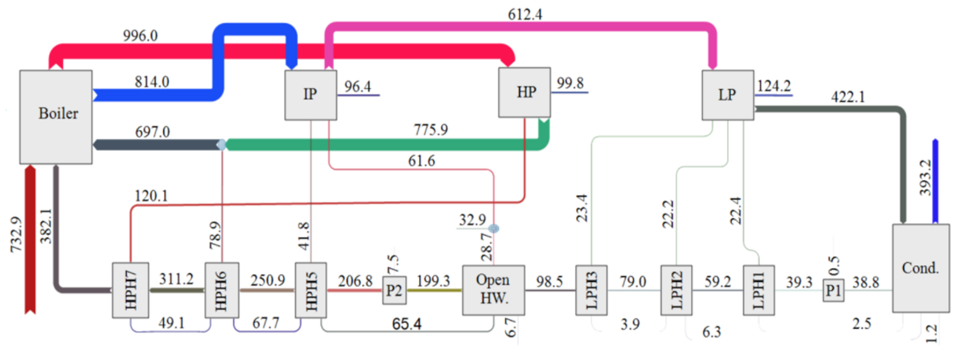


Fig. 6. ITPP power cycle diagram at rated load (numbers are rounded and in MW)

3.1. Effect of condenser temperature on efficiency and output power of ITPP

According to the data related to ITPP, the output power and thermal efficiency of the ITPP steam cycle are computed based on different condenser temperatures. The obtained results are presented in Fig. 7, where it can be seen that the increase

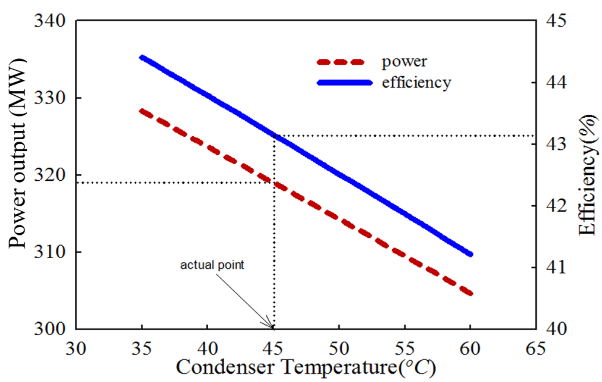


Fig. 7. Effect of condenser temperature on output power and thermal efficiency in ITPP steam cycle

in condenser temperature can reduce the output power and thermal efficiency of the power plant units.

3.2. Effect of condenser temperature on Heller tower size

The ambient temperature severely affects the efficiency of the Heller tower, resulting in changes in the temperature of the water leaving the tower, leading to wide variations in condenser pressure. Due to climatic change, the water temperature of the cooling tower changes seasonally and as a result the pressure and temperature of the condenser changes. Fig. 8 shows the effect of the ambient temperature on the outlet water temperature of the Heller tower of the SMPP, where this power plant is located close to the ITPP.

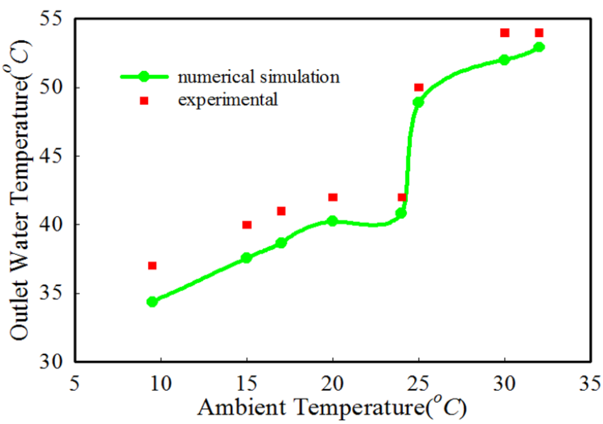


Fig. 8. The change in the outlet water temperature of the Heller tower based on the change in the ambient temperature and comparison with the experimental data of SMPP [19]

Heat is dissipated in natural draft dry cooling towers due to the temperature difference between the water and the ambient temperature, with great dependence on climatic and wind conditions. In order to compensate for the increase in ambient temperature and wind blow, a high condenser temperature should be considered in the design of the condenser. This fact, in addition to installation costs, significantly reduces the dimensions of the cooling tower.

The specifications of a Heller cooling tower for a 7 m^3/s flow, 30°C of ambient temperature and 25% RH to have a temperature difference of 7.8°C in inlet and outlet water temperatures (according to data from ITPP), are shown in Fig. 9. As the temperature of the condenser increases, the dimensions, the number of heat exchangers and the size of the Heller cooling tower are drastically reduced.

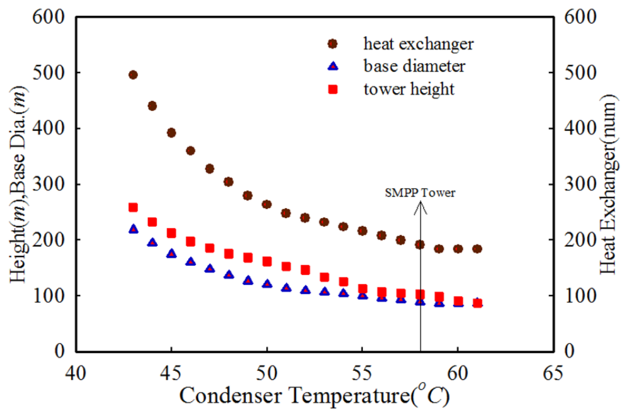


Fig. 9. Effect of condenser temperature on the Heller cooling tower dimensions

4. Heller cooling tower design without changes in turbine and pumping system

First, the pressure drop coefficients should be determined according to the second power of the mass flow rate in the condenser. Then, the head drop of the riser water pipes of the wet cooling tower should be reduced. The pressure drop levels of the condenser, pipes and Heller tower are calculated based on the water flow rate. The pressure drop curve in terms of system flow rate (system resistance curve) is compared with the characteristic curve of the existing pump, and the new operating point of the system and pump is determined.

4.1. Tower design without increasing condenser temperature

Based on ITPP data, a Heller tower is designed at 30°C, 25% relative humidity, and is related to the atmospheric pressure of Isfahan. The circulating water flow is considered to be 13.167 m^3/s. The new inlet and outlet water temperatures of the cooling system are 45°C and 37.2°C, respectively, according to the current conditions of ITPP. The Heller tower computer program has been developed for one or more towers with different conditions with variety of water flow rates in a single or parallel arrangement of towers. Some similar data of SMPP and SPP without increasing condenser temperature are presented in Tables 5 and 6.

Tables 5 and 6 show that the obtained dimensions do not correspond to what might be manufacturable, and this is because the design conditions have deviated from their true path. Due to the small temperature difference between the water and the environment, the determined number of heat exchangers is very high. Therefore, the lower water speed in heat exchangers reduces the heat transfer coefficient on the water side. Considering the mentioned drawbacks, the installation of the Heller tower assuming no increase in the temperature of the condenser temperature is

Table 5. Data for a tower similar to SMPP towers without increasing condenser temperature at a delta angle of 60°

Flow (m^3/s)	Number of heat exchangers	Base diameter (m)	Upper opening diameter (m)	Height (m)
12	720	304.9	179.9	165.7
14	840	355.7	209.9	209.5
16	960	406.5	239.8	239.8

Table 6. Designed data for two parallel towers similar to SPP towers without increasing in condenser temperature at a delta angle 49°

Flow (m^3/s)	Number of heat exchangers	Base diameter (m)	Upper opening diameter (m)	Height (m)
6	380	155.2	87.64	171. 1
7	442	180.7	101.92	166.65
8	406	207.4	116.67	163.25

neither logical nor justifiable. As a result, the condenser must operate at a higher temperature despite reduced power and thermal efficiency.

4.2. Tower design by increasing condenser temperature

For Heller cooling tower design, the water flow rate, inlet and outlet water temperature, and conditions of the environment must be known. With an estimated flow rate at 58°C intake water temperature, the design ambient air temperature entering the Heller tower for SMPP and SPP is assumed to be 30°C. By determining the number of heat exchangers, the pressure drop curve of the new system is compared with the characteristic curve of the existing pumps and the crossing point shows the flow rate. This process is repeated several times with different numbers of towers and arrangements until all the designed and simulated towers are acceptable and the pump efficiency is kept as high as possible. To calculate the friction coefficient on the water side of the Forgo heat exchanger, the following equation is applied [23]:

$$f = 0.25 \left[\log \left(\frac{e}{3.7 d_i} + \frac{5.74}{\mathrm{Re}_w^{0.9}} \right) \right]^{-2} . \tag{19}$$

Using the friction coefficient obtained through Eq. (19), the pressure drop of the Forgo heat exchanger is calculated from Eq. (20), which is then added to the pressure drops associated with the condenser and other parts of the water pipes.

$$h_{l,\mathrm{Forgo}} = \frac{8 f\ L\,\mathrm{pass} \left(\dot{m}_{T,w} / 120 N_{\mathrm{delta}} \right)^2}{\left(\pi^2 g d_i^5 \right)} . \tag{20}$$

Here $m_{T,w}$ is the water flow rate entering to Heller tower. The new head loss system is as follows:

$$h_{l,\text{new}} = h_{l,\text{other}} + h_{l,\text{Forgo}} . \tag{21}$$

Since the riser pipes in the existing wet tower are eliminated in the new system, the slope of the pressure drop curve will be lower than the current one. Another thing is that the temperature difference between the environment and the system increases. This means that the contribution of water flow in heat dissipation is reduced. Reducing the flow in this way also causes a further reduction in the slope of the pressure drop curve. As a result, the flow is higher than before. This contradiction means that the existing pumps are larger than necessary and must be replaced with smaller pumps. This issue is clear in Fig. 10. By using the current pumps and increasing the flow rate, the number of heat exchangers increases and finally the designed tower is large and structurally unacceptable. As a result, it cannot be designed using the existing pumps.

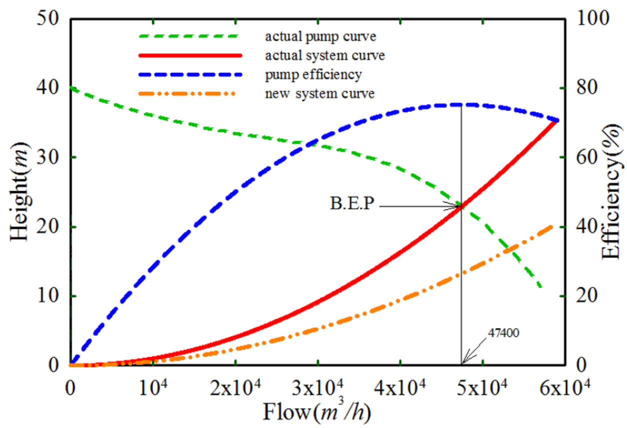


Fig. 10. System resistance curves and characteristic curve of pumps

5. Effect of temperature increase on turbine blades

As the power of the turbines increases, the size of the rotors, discs and blades increases proportionally, which causes problems. Because the last rows of low-pressure turbines are long, pressure, temperature, and steam quality affect efficiency and blade life. Increasing the dimensions increases the tension and vibration of the blades. Reducing the temperature of the turbine exhaust increases the speed of the steam, which reaches the speed of sound, and the generated sonic waves create cyclic stresses. Since the longer blades change cross-section in a revolving manner (due to a change from impulsing to reactive conditions), they are exposed to twisting and rotating forces. Due to the large dimensions and different hardness at disc connection points, they have different natural vibrating frequencies. In order to reduce vibration (self-imposed vibration), a group of a few blades is connected through a lashing wire. Such groups consist of eight blades in ITPP. The temperature increase at the last row causes the breakage of the lashing wire, which is the first factor that intensifies the cycle fatigue caused by the separation of the grouped blades. In a sense, high temperature causes low cycle fatigue through high centrifuge force in long blades, and in a case where this cycle is rated as overspeed, the condition is worsened, especially when the chemical environment of the steam is unsuitable. It causes cracks in the blades that eventually break and cause drastic vibration in the adjacent blades [24–26].

The increase in exhaust pressure disturbs the steam flow pattern on the blades. This pattern depends on the steam speed in the turbine stages. Hence, a different kinetic energy is is converted to work. The increased pressure causes thermal friction, also known as thermal windage. It should be noted that this thermal condition is due to the high speed of the blade (about 500 m/s at the top of the blades). In this condition, the steam flow pattern due to the pressure increase in the exhaust is such that the steam speed is behind the rotor movement and the blades fan the steam using the energy released from the other primary stage blades. This phenomenon not only reduces drastically the efficiency, but also intensifies friction at the top of the blades.

An increase in the condenser temperature of a low-pressure turbine has a destructive effect on the blades of the last stage. As the steam is ejected through the nozzles and passes through the moving blades, it causes the moving blades to vibrate. The high turbine speed assists this phenomenon, and as a result, the vibration frequency of the blade increases. A few natural frequencies are inevitable for these blades. The condition where the steam flow pattern causes the blades to vibrate is called “stall flutter” and is caused by low vacuum.

6. Design based on increasing condenser temperature and changing the pump system

The characteristics of ITPP are similar to those of SPP, and we used Heller-type SPP cooling tower for the new ITPP system. The last row of blades removal slightly affects the output power of the power plant. Here, two cases with and without last-row blades of a low-pressure turbine are evaluated for design. In both cases, the output power and thermal efficiency reduction conditions are compared and discussed with the existing situation (condenser temperature of 45°C, see Fig. 7).

6.1. Cooling tower design using existing last row turbine blades

The inlet and outlet water temperature difference in the current condenser is 7.8°C, which is estimated to be 10°C through the new system, which is related to SMPP and SPP. It can reach 14°C for ITPP [22]. An increase in condenser temperature results in a decrease in output power and thermal efficiency (see Fig. 7) compared to the existing 45°C and output power and thermal efficiency decrease by 12.68 MW and 1.68%, respectively at the modified 58°C temperature. Based on the SPP data, the tower design is based on an ambient temperature of 30°C, a circulating water flow rate of 34000 m^3/h and a delta angle of 49°. Considering the low vacuum of the power plant condenser in the summer and the lack of water spraying in the ITPP condenser, the speed of the air passing through ITPP heat exchangers was considered lower than SPP, which increased the number of heat exchangers. The geometric specifications of the new design are determined and compared with SPP specifications in Table 7.

Table 7. Specifications of the proposed cooling tower without blades

Parameter	Flow rate (m^3/h)	Number of heat exchangers	Base diameter (m)	Upper opening diameter (m)	Height (m)
calculated	34000	286	116.05	65.4	158.8
suggested	34000	286	116	65	160
SPP	34000	264	110	62	150

6.2. Cooling tower design by removing the last row of blades

The low-pressure turbine in the ITPP is symmetrical, which technically allows the elimination of the last row blades. The calculations related to the design of a condenser with a temperature of 58°C are repeated assuming the removal of the last row blades. In this case, the low-pressure water heater (heater No. 1 in Fig. 5) and the steam extraction located near the last row of blades are also removed. Calculations of the new conditions of the ITPP thermodynamic cycle show that the output power, thermal efficiency and condenser temperature will be 301.5MW, 40.79% and 58°C, respectively. In other words, according to Fig. 7, the output power is reduced by 17.5 MW and the thermal efficiency is reduced by 2.36%.

By removing the last row of blades, the increase in heat loss in the turbine without removing the last row of blades is observed at the condenser temperature of 58°C. The design conditions are calculated according to the specifications for SPP. There are two possibilities to consider: Increasing the temperature difference in the cooling tower up to 13°C or increasing the circulating water flow rate up to 44.36 m^3/h in the existing temperature difference of 10°C. A slight increase in temperature or water flow rate confirms that at high condenser temperatures, the efficiency of the last row blades is low compared to low condenser temperature.

The geometric specifications of the proposed Heller cooling tower are presented in Table 8. In both of the above assumed cases, if the blades are not removed, there is no significant change in the number of heat exchangers and the base diameter of the Heller tower. A significant reduction in output power and thermal efficiency of the power plant is evident compared to the current situation.

Table 8. Suggested cooling tower specifications with removed blades

Parameter	Flow rate (m^3/h)	Number of heat exchangers	Based on diameter (m)	Upper opening diameter (m)	Height (m)
Increasing temperature	34000	290	117.68	66.31	158.34
Increasing flow	34044.36	286	116.05	65.4	158.8

7. Heller cooling tower plant layout study

There is insufficient space to install Heller cooling towers in ITPP, even if the wet towers are removed. The power plant is located on the eastern side of the Zayandehrood River and these areas are used for agriculture, so the Heller towers can be installed there. In Fig. 11, the dashed lines represent the water intake of the tower while crossing the river, and the circles of the tower and the solid lines represent the water exiting from the tower. The rectangular section represents the proposed location of the new pump house and the new pipelines will be on the internal streets of the plant. The distance between the units and the installation site of the proposed tower is 250 m. The advantage of this location is that the plant does not shut down while the new system is being installed.

Fig. 11. Heller tower installation location in ITPP

8. Conclusions

The replacement of Heller tower with ITPP wet towers was evaluated. In all the different methods of investigation, it was assumed that there is no change in the structure of the condenser because it was not economical at all. First, the thermal performance of Heller Tower was analyzed and simulated by a computer program. The results of the computer program were compared with the Heller Tower of SMPP and the Heller Tower of SPP and the correctness of the program was confirmed.

The ideal thermodynamic cycle of the power plant was simulated based on the characteristics of the 320 MW ITPP units, and the output power and thermal efficiency of the cycle were obtained based on the condenser temperature in the ideal state. Then, the effect of the condenser temperature on the size of the Heller tower was investigated. It was observed that the amount of output power and efficiency has an inverse relationship with the size of the Heller tower and an agreement should be made between these two issues. The design was investigated assuming no change in the temperature of the condenser, turbine and current pumping system and several modes and arrangements. These tower were large in terms of its dimensions and size and most of them were structurally inappropriate.

The design was done with the current system and assuming an increase in condenser temperature and two factors can reduce the head losses in the system and the reduction of the system head losses leads to an increase in the flow rate. One is to remove the rising pipes of the wet tower, and the other is to reduce the necessary flow of circulating water due to the increase in temperature difference between the tower and the environment. This contradiction is due to the large size of the existing pumps for the new system, so it was concluded that the new pumps should be smaller than the existing pumps. After that, the effect of temperature increase on the turbine blades was investigated and it was stated that the increase in temperature leads to the creation of various stresses in the blades of the last row of the low-pressure turbine. Then, the design was carried out assuming an increase in the condenser temperature and with the specifications of the SPP due to the closeness of its rated output power to units of ITPP. This design was initially carried out with the assumption of not removing the blades of the last row of the low-pressure turbine, as the designed tower was very close to the SPP Heller tower. In this case, the output power decreased by 12.4 MW and the thermal efficiency decreased by 1.68% compared to the current state. In the next step, the design was carried out assuming the removal of the blades of the last row in two cases. One mode is to increase the temperature difference in the tower, and the other mode is to increase the flow of circulating water. It was shown that the case of increasing the temperature of the proposed tower was not significantly different from the case without removing the last row of blades. The case with increased flow rate was the same as the case without removing the last row of blades. The decrease in output power and the decrease in thermal efficiency were calculated to be 17.5 MW and 2.36%, respectively, in this mode compared to the current mode. At the end, the placement of the designed Heller towers and the new pumping system was discussed according to their required and available space.

Acknowledgements

The authors thank and support the management staff of Isfahan, Shahid Montazeri, and Shazand power plants.

References

[1] B. Dziegielewski and D. Baumann. Tapping alternatives: The benefits of managing urban water demands. *Environment: Science and Policy for Sustainable Development*, 34(9):6–41, 2010. doi: 10.1080/00139157.1992.9930929.

[2] D. Marmer. Water conservation equals energy conservation. *Energy Engineering*, 115(5):48–63, 2018. doi: 10.1080/01998595.2018.12027708.

[3] J.M. Burns, D.C. Burns, and J.S. Burns. Retrofitting cooling towers: estimates required to achieve the next level of CWA 316(b) compliance. In *Proceedings of the ASME Power Conference*, pages 25–33, 2004. doi: 10.1115/POWER2004-52051.

[4] A. Loew, P. Jaramillo, and H. Zhai. Marginal costs of water savings from cooling system retrofits: a case study for Texas power plants. *Environmental Research Letters*, 11(10):104004, 2016. doi: 10.1088/1748-9326/11/10/104004.

[5] A.E. Conradie and D.G. Kröger. Performance evaluation of dry-cooling systems for power plant applications. *Applied Thermal Engineering*, 16(3):219–232, 1996. doi: 10.1016/1359-4311(95)00068-2.

[6] A.E. Conradie, J.D. Buys, and D.G. Kröger. Performance optimization of dry-cooling systems for power plants through SQP methods. *Applied Thermal Engineering*, 18(1-2):25–45, 1998. doi: 10.1016/S1359-4311(97)00020-3.

[7] J.D. Buys and D.G. Kröger. Dimensioning heat exchangers for existing dry cooling towers. *Energy Conversion and Management*, 29(1):63–71, 1989. doi: 10.1016/0196-8904(89)90014-9.

[8] Z. Zou, Z. Guan, H. Gurgenci, and Y. Lu. Solar enhanced natural draft dry cooling tower for geothermal power applications. *Solar Energy*, 86(9):2686–2694, 2012. doi: 10.1016/j.solener.2012.06.003.

[9] S. Bagheri and M. Nikkhoo. Investigation of the optimum location for adding two extra Heller-type cooling towers in Shazand power plant. *Proceedings of the 17th IAHR International Conference on Cooling Tower and Heat*, pages. 74–83, Australia, 2015.

[10] W. Peng and O.K. Sadaghiani. Presentation of an integrated cooling system for enhancement of cooling capability in Heller cooling tower with thermodynamic analyses and optimization. *International Journal of Refrigeration*, 131:786–802, 2021. doi: 10.1016/j.ijrefrig.2021.07.016.

[11] M.A. Ardekani, F. Farhani, and M. Mazidi. Effects of cross wind conditions on efficiency of Heller dry cooling tower. *Experimental Heat Transfer*, 28(4):344–353, 2015. doi: 10.1080/08916152.2014.883449.

[12] A. Jahangiri, A. Borzooee, and E. Armoudli. Thermal performance improvement of the three aligned natural draft dry cooling towers by wind breaking walls and flue gas injection under different crosswind conditions. *International Journal of Thermal Sciences*, 137:288–298, 2019. doi: 10.1016/j.ijthermalsci.2018.11.028.

[13] A.R. Seifi, O.A. Akbari, A.A. Alrashed, F. Afshari, G.A.S. Shabani, R. Seifi, M. Goodarzi, and F. Pourfattah. Effects of external wind breakers of Heller dry cooling system in power plants. *Applied Thermal Engineering*, 129: 1124–1134, 2018. doi: 10.1016/j.applthermaleng.2017.10.118.

[14] R.A. Kheneslu, A. Jahangiri, and M. Ameri. Interaction effects of natural draft dry cooling tower (NDDCT) performance and 4E (energy, exergy, economic and environmental) analysis of steam power plant under different climatic conditions. *Sustainable Energy Technologies and Assessments*, 37:100599, 2020. doi: 10.1016/j.seta.2019.100599.

[15] A. Jahangiri and F. Rahmani. Power production limitations due to the environmental effects on the thermal effectiveness of NDDCT in an operating powerplant. *Applied Thermal Engineering*, 141:444–455, 2018. doi: 10.1016/j.applthermaleng.2018.05.108.

[16] A.D. Samani. Combined cycle power plant with indirect dry cooling tower forecasting using artificial neural network. *Decision Science Letters*, 7:131–142, 2018. doi: 10.5267/j.dsl.2017.6.004.

[17] T.L. Bergman, F.P. Incropera, D.P. DeWitt, and A.S. Lavine. *Fundamentals of Heat and Mass Transfer*. John Wiley & Sons, 2011.

[18] Archive of Isfahan Mohammad Montazeri Power Station. Isfahan, Iran, 1984.

[19] H. Ahmadikia and G. Iravani. Numerical and analytical study of natural dry cooling tower in a steam power plant. *Journal of Advanced Materials in Engineering (Esteghlal)*, 26(1):183–195, 2007. (in Persian).

[20] H.G. Zavaragh, M.A. Ceviz, and M.T.S. Tabar. Analysis of windbreaker combinations on steam power plant natural draft dry cooling towers. *Applied Thermal Engineering*, 99:550–559, 2016. doi: 10.1016/j.applthermaleng.2016.01.103.

[21] K.F. Reinschmidt and R. Narayanan. The optimum shape of cooling towers. *Computers & Structures*, 5(5-6):321–325, 1975. doi: 10.1016/0045-7949(75)90039-5.

[22] Isfahan Thermal Power Plant documents, No. C.583 and C.749, Islam Abad Power Plant, Isfahan, Iran, 1988.

[23] I.H. Shames. *Mechanics of Fluids*. 4th ed. McGraw-Hill, New York, 2003.

[24] C.R.F. Azevedo and A. Sinátora. Erosion-fatigue of steam turbine blades. *Engineering Failure Analysis*, 16(2):2290–2303, 2009. doi: 10.1016/j.engfailanal.2009.03.007.

[25] H. Kim. Crack evaluation of the fourth stage blade in a low-pressure steam turbine. *Engineering Failure Analysis*, 18(3):907–913, 2011. doi: 10.1016/j.engfailanal.2010.11.004.

[26] L.K. Bhagi, P. Gupta, and V. Rastogi. Fractographic investigations of the failure of L-1 pressure steam turbine blade. *Case Studies in Engineering Failure Analysis*, 1(2):72–78, 2013. doi: 10.1016/j.csefa.2013.04.007.