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DEVELOPMENT OF THE CONCEPT OF DOUBLE REFRIGERANT AND AIR COOLED STEAM POWER PLANT

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ABSTRACT

In the present paper a system is proposed to improve the performance of air cooled steam power plant. In this system, two steam turbines are used rather than one turbine. The exhaust steam of the one turbine is cooled in an air condenser, while the outlet steam of the other turbine is cooled in a refrigerant cooled condenser. Two refrigeration machines are employed with this system. The first one serves to produce cooling for condensing the steam in the air condenser. The other machine produces cooling during the off-peak loads to be stored in a cooling storage container to be used for cooling the refrigerant cooled condenser through the period of the peak-loads. The refrigerating machines have three different operating periods; i.e. direct cooling of the refrigerant cooled condenser, charging the cooling storage container and discharging it. Simple energy analyses have been developed for predicting the energy output of the proposed combined steam power plant system. On an example of nearly realistic steam plant data the developed analyses were used to study the energy characteristics of the proposed system. In this study, the temperature in the refrigerant cooled condenser was maintained at a design value of 20^{o}C. The results of this work has led to inferring that the use of the proposed combined system effects achieving an increase of 6, 14, 22, and 32% of plant power with only air condenser during peak-loads for ambient temperature of 20, 30, 40 and 40 ^{o}C respectively, when all steam exhaust of the two steam plants is cooled in a refrigerant cooled condenser.

INTRODUCTION

Steam power plants represent the largest segment of the world electricity production. These plants rely on a fuel source (fossil or nuclear) to heat water to steam that is used to drive a turbine-generator. Steam exhausted from the turbine is condensed and recycled to the steam generator or boiler. The steam condensation occurs in a steam condenser. Cooling water mass flow rates of greater than 25 times the steam mass flow rate are necessary depending on the allowable temperature rise of the cooling water – typically 8-14^{o}C. Cooling water is the major source of wastewater generated by most thermal power plants. For each MWe of a facility about 7.5 m^3 per day of waste water are generated, with about 70% of this wastewater coming from cooling tower blow-down [1].

Due to enhanced concern about water supplies and water use priorities, air cooling systems for thermal power plants are receiving increased consideration, even though electric power from air cooled power plants currently costs 10 to 15 per cent more than power from wet-cooled plants [2]. The steam condensing pressures and temperatures of air cooled unit are significantly higher than a wet cooled unit, due to the low heat transfer rates of air cooling and operation at the dry bulb temperature rather than the wet bulb one. There is little current research and development work being reported in the literature on air cooling systems for power plants. A few important exceptions include improved heat exchanger geometries for finned tube bundles in air-cooled condensers [3, 4]; enhancement of air-cooled condenser performance with the use of limited water [5, 6]; the use of evaporative condenser [7]; optimization techniques [8]; and using double wet and air cooled condenser, where the heat of the wet condenser is dissipated into a cooling storage container [9].

The concept of ammonia dry cooling system is reported in [10]. The system is an indirect type, in which the usual circulating water loop is replaced by a phase-change ammonia loop, where the ammonia is evaporated in the tubes of the steam condenser and condensed in an air-cooled condenser. The of condensation absorbed by the ammonia is rejected in an air cooled condenser of a refrigeration machine into the surrounding atmosphere. This concept was tested and well documented [10-13], with the participation of several major equipment vendors (Baltimore Air Coil, The Trane Company, Curtiss-Wright, CB&I, and Union Carbide). This concept has the advantage of high heat transfer rates from the steam to the ammonia liquid. Meanwhile, low condensation temperature of the steam can be achieved since it is controlled by the ammonia refrigeration machine. However, most of the power generated by the steam plant is used for driving ammonia compressor of the refrigeration machine during peak-loads.

A cooling system for steam plant is proposed in the current paper that has potential to take advantage of high performance of refrigerant cooled steam condenser and increase plant output at peak-loads. This system is composed of Air cooled steam condenser, refrigerant cooled steam condenser and cooling storage container. A portion of the exhaust steam of the steam plant is condensed in air cooled condenser, while the rest is condensed in the refrigerant cooled condenser. The condensation heat absorbed by the refrigerant is dissipated over part loads period directly to the atmosphere through an air-cooled refrigerant condenser of the refrigeration machine. During the period of off peak loads, the refrigeration machine produces, beside the direct cooling necessary for condensing the steam of the refrigerant cooled steam condenser, cooling to be stored in the storage container. During the period of peak loads, the refrigeration machine is stopped and the stored cooling is for dissipating the heat absorbed by the refrigerant. In this way the power of the refrigerant compressor is saved so that the whole power generated by the steam plant can be sent to the grid.

NOMENCLATURE

COP	refrigeration machine coefficient of performance	-
E	energy	kJ
h	specific enthalpy	kJ/kg
p	pressure	bar
P	power generated by the steam plant	kW
Pr	Power of the refrigerant compressor	kW
Q	heat rate	W
R	Power fraction defined by Eq. (1)	-

Greek letters

Δt	time period duration	s
η_o	power plant overall efficiency	-
η_{th}	steam cycle thermal efficiency	-
ζ	Cooling loss coefficient	-

Subscripts

a	operation with air cooled condenser
b	boiler
$b1, b2$	turbine designation
c	Condenser
dc	direct cooling of the steam condenser
ev	Evaporator
n	net power of the steam plant
r	operation with refrigerant cooled condenser
st	cooling storage refrigeration machine
t	total power
0	condenser design temperature
$1, 2, 3$	refrigeration machines operating periods
$1,..,15$	refers to the steam states (Figures 1,2)
$'$	isentropic expansion

DESCRIPTION Of THE PROPOSED SYSTEM

The configuration of the proposed double refrigerant and air cooled steam plant system is shown schematically in Figure 1. This system consists of a steam power plant cycle and two refrigeration cycles. The steam cycle comprises two steam turbines and other auxiliaries (boiler, feed water heaters, condensers…), which constitute a modern steam power plant. Of these constituents only the boiler (a), steam turbines (b1) and (b2), air cooled condenser (c), refrigerant-cooled condenser (d), feed water pumps (e1) and (e2) and generator (f) are shown in fig. 1. The air-cooled condenser (c) is used to condense the exhaust steam of the turbine (b1), while the water-cooled condenser (d) serves to condense the steam outlet from the turbine (b2). The corresponding *T-s* diagram of this cycle is shown in Figure 2. The numeric data of Fig. 1 correspond to the points 1 to 7 given in Fig. 2.

The first refrigeration cycle is referred to as (dc). It is made up of a compressor (g1) driven by an electric motor (h1), a refrigerant condenser (i1), a throttling valve (j1) and an evaporator (d) which acts at the same time as the steam condenser of the steam plant. In this cycle, the low pressure refrigerant vapor is compressed by the compressor (g1) and transferred to the refrigerant condenser (i1). The heat absorbed from the condensing steam of the evaporator condenser (d) and the heat of compression is removed by the air-cooled refrigerant condenser (i1), where the high pressure vapor refrigerant is condensed. The condensed refrigerant liquid is transferred to the refrigerant expansion valve (j1). This valve has the function to reduce the pressure of the liquid refrigerant, and in turn, the boiling point of the refrigerant is lowered. The low pressure liquid refrigerant flows through the tubes of the evaporator condenser (d), absorbing heat, changing into vapor, in turn condensing the steam. The water condensate and low pressure vapor refrigerant leaving the evaporator condenser (d) complete the steam and the first refrigeration cycles respectively.

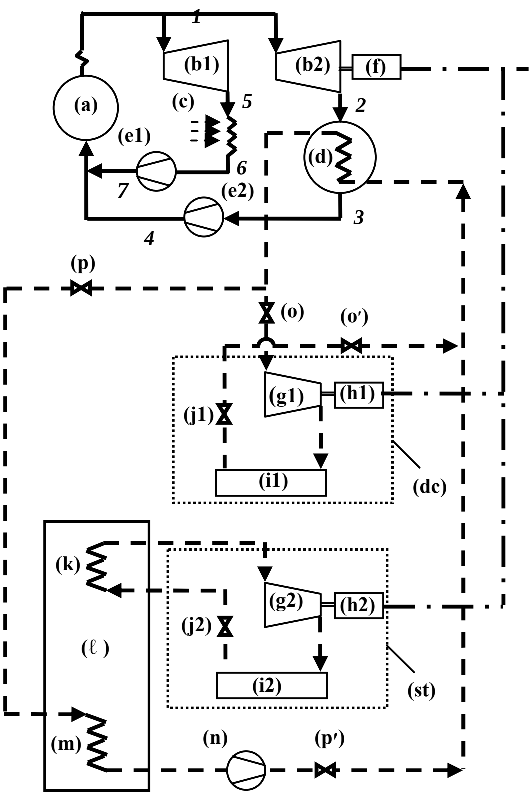

Fig. 1. Schematic diagram of the proposed combined steam cycle

(a) boiler
(b1), (b2) steam turbines
(c) steam condenser
(d) evaporator/ steam condenser
(e1), (e2) feed water pump
(f) generator
(g1), (g2) refrigerant Compressors
(h1), (h2) electric motors
(i1), (i2) refrigerant condensers
(j1), (j2) throttling valves
(k) evaporator
(ℓ) cooling storage container
(m) heat exchanger
(n) refrigerant Circulating pump
(p), (o′), (p), (p′) valves
water/steam refrigerant
electricity

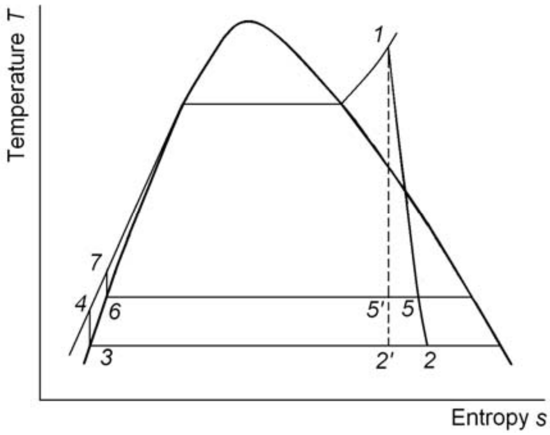

Fig. 2 Temperature - entropy diagram of the studied steam cycle

The second refrigeration cycle is designated by (st). It consists of a compressor (g2), operated by an electric motor (h2), a refrigerant condenser (i2), a throttling valve (j2) and an evaporator (k). This cycle functions quite similar to the first refrigeration cycle except that the low pressure liquid refrigerant leaving the throttle valve (j2) evaporates as it flows through the evaporator (k) by absorbing heat from the cooling storage container (ℓ), where the cooling is stored.

The two refrigeration cycles have three different operating periods along the time of a day: the period of exclusive direct cooling of the steam condenser (d); the period of charging the cooling storage container (ℓ) and the period of discharging the cooling storage container. The first period occurs at part-loads when the steam plant can generate excess power over the demand of the end users which is adequate or larger than that needed for running the refrigerant compressor of the first refrigeration cycle but less than the power required to operate the compressors of both refrigeration cycles. Over this period, the compressor of the second refrigeration cycle is stopped, while that of the first cycle runs to produce cooling that is used to directly condense the exhaust steam of the turbine (b2). During this period, the valves (o) and (o′) are opened and the valves (p) and (p′) are closed. The cool refrigerant liquid getting out of the throttle valve (j1) flows through the tubes of the evaporator condenser (d), absorbs the heat of vaporization of the steam coming out of the steam turbine (b2) and is vaporized. The refrigerant vapor exiting the evaporator condenser (d) is sent to the refrigerant compressor (g1), where it completes the first refrigeration cycle.

The second period (charging the cooling storage container) takes place at part-loads when the excess power of the steam plant suffices to drive the compressors of both refrigeration cycles. The cooling produced by the first cycle serves to directly condense the exhaust steam of the turbine (b2) at this period, while the cooling produced by the second cycle is stored in the cooling storage container (ℓ). During this period, the valves (p) and (p′) are closed, whereas the valves (o) and (o′) are opened. The refrigerant vapor coming out of the evaporator condenser (d) and evaporator (k) are compressed in the compressors (g1) and (g2) respectively, and complete the refrigeration cycles.

The third period (discharging the cooling storage container) comes about throughout the peak-loads of the steam plant. During this period, the refrigeration compressors (g1) and (g2) are set out of operation, and the valves (o) and (o′) are closed, while the valves (p) and (p′) are opened. The cooling refrigerant of the steam condenser (d) changes its loop, in which it no longer flows through the first refrigeration cycle. Rather, the refrigerant vapor exiting the evaporator condenser (d) flows through the heat exchanger (m), where it rejects the heat absorbed from the condenser steam to the cooling storage container (ℓ) and is condensed. It is then re-pumped by the circulating pump (n) to the evaporator condenser (d) for condensing the exhaust steam of turbine (b2). The refrigerant is vaporized in the evaporator condenser (d) and repeats this cooling loop until the demand of the end users gets low enough, so that the first refrigeration cycle is re-activated.

Optimal operation of the steam plant is achieved when the turbine exhaust steam is cooled to its design point which is in the range of 15 -30 ^{o}C. Accordingly the temperature of the steam condenser cooling refrigerant can be in the range 10 – 25 ^{o}C. As a result, the evaporator temperature of the first refrigeration cycle (used for direct cooling of the steam condenser) is high enough to effect large coefficient of performance for direct cooling.

As to the second refrigeration cycle (st) which serves to charge the cooling storage container, it is worthy mentioning that cooling is stored as ice, chilled water or some other materials. As a tremendous amount of heat must be removed from the steam condenser, ice storing is preferred in this case to limit the volume of the storing container. Production of ice forces the second refrigerating machine to operate at inevitably low evaporator temperature and cause low coefficient of performance. Fortunately, the storing process takes place when the end users demand is minimum, which occurs at night hours and early morning, where the ambient temperature has its lowest value. This makes the temperature of the condenser of the refrigerating machine relatively low that partially compensate for the low temperature of the evaporator by fairly improving the coefficient of performance. As the condensers of the both refrigeration cycles (dc) and (st) are air cooled their coefficients of performances are 77 – 85 % lower than those of the refrigeration cycles with water cooled condensers [14]. Accordingly and concerning the performance of the large industrial cooling refrigeration cycles given in [15], it is expected for the coefficient of performance of the refrigeration cycle (dc) to be greater than 4 and for the coefficient of performance for the refrigeration cycle (st) to lie in the range 1.5 – 3.

ENERGY ANALYSES OF THE PROPOSED SYSTEM

The objective of the energy analyses presented in this section is to predict the energy output of the proposed combined refrigerant and air cooled steam power plant under different operating conditions. For this purpose, it is necessary to know the characteristics of the steam cycle implemented by each turbine. Generally, modern steam plants are using sophisticated steam cycles which include feed water heaters, re-heaters, etc. to assure high thermal efficiency. The details of these cycles are beyond the scope of the present paper. Therefore, the simple steam Rankin cycle, as shown in Figs. 1 and 2 is considered for the following analyses. The thermal and overall efficiency of the simple steam cycle and turbine respectively, are modified to take into account this assumption.

Referring to Figs. 1 and 2, the steam exiting the boiler of the steam plant with state *1* is expanded in the turbine (b1) to point *5* and turbine (b2) to point *2*. If it is assumed that the exhaust steam temperature of both turbines (b1) and (b2) is at a standard condition (design condition) denoted by the subscript *0*, then the power generated by turbines (b1) and (b2) at this condition are $P_{b1,0}$ and $P_{b2,0}$ respectively. Hence, the power fraction R is introduced by the following relation:

$$R = \frac{P_{b2,0}}{P_{b1,0} + P_{b2,0}} = \frac{P_{b2,0}}{P_{t,0}} \tag{1}$$

Where $P_{t,0}$ is the total power ($P_{b1,0}+P_{b2,0}$) generated by the power plant at the design condition of the exhaust steam temperature.

Obviously, the temperature of the exhaust steam of turbine (b2) can be cooled to the design temperature, since the cooling refrigerant temperature can be controlled by the first refrigeration cycle (dc) and/ or storage container. Accordingly, it is assumed throughout the following analyses that the power $P_{b2,w}$ of the turbine (b2) with refrigerant-cooled condenser is equal to $P_{b2,0}$ at the design condition. Moreover, it is assumed that the turbine internal efficiency, mechanical and generator efficiencies are the same for both turbines (b1) and (b2) and remain unchanged during the operation of the plant. Considering the above mentioned assumptions and neglecting powers of the pumps (e1) and (e2), it follows from Fig. 2. that:

$$\frac{\eta_{o,0}}{\eta_{o,a}} = \frac{\eta_{th,0}}{\eta_{th,a}} = \left(\frac{h_1 - h_{2'}}{h_1 - h_{5'}}\right)\left(\frac{h_1 - h_6}{h_1 - h_3}\right) \tag{2}$$

It is to be mentioned here that fan powers of air cooled refrigerant condenser and steam condenser are included in the total efficiencies $\eta_{o,0}$ and $\eta_{o,a}$ respectively. In air cooled steam condenser, heat of steam condensation is removed by the air cooling. For the refrigerant cooled condenser heat of steam condensation absorbed by the cooling refrigerant and heat of compression are rejected by the cooling air of air cooled refrigerant condenser. Thus, the heat to be get red of to the surrounding is greater for refrigerant cooled steam condenser than that of air cooled steam condenser. However, the temperature rise o cooling air is higher for the former condenser than that for the latest one. Accordingly, it is assumed here that fan powers required for both condensers are equal.

From Fig. 2 and by aid of Eq. (2), the following equation for power ratios can be derived:

$$\frac{P_{b1,0}}{P_{b1,a}} = \frac{P_{b2,0}}{P_{b2,a}} = \frac{P_{b2,r}}{P_{b2,a}} = \frac{P_{t,0}}{P_{t,a}} = \left(\frac{h_1 - h_{2'}}{h_1 - h_{5'}}\right)^2 \left(\frac{h_1 - h_6}{h_1 - h_3}\right) \tag{3}$$

In order to be able to calculate the compressor power of the refrigeration cycle (dc) during the first period (exclusive direct cooling of the steam condenser), it is necessary to determine the rate of heat it removes from the steam condenser (d). The overall efficiency $\eta_{o,0}$ of the turbine (b2) and the thermal efficiency of its steam cycle are defined as:

$$\eta_{o,0} = \frac{P_{b2,w}}{Q_b} \tag{4}$$

$$\eta_{th,0} = \frac{Q_b - Q_c}{Q_b} \tag{5}$$

Solving Eqs. (4) and (5) to get Q_c results in:

$$Q_c = \frac{P_{b2,w}}{\eta_{o,0}} \left(1 - \eta_{th,0}\right) \tag{6}$$

Given the coefficient of cooling loss ξ_{dc} for direct cooling, the rate of heat $Q_{ev,dc}$ to be removed by the evaporator/ condenser (d) of the refrigeration machine (dc) during this period may be given in combination with Eq. (3) by:

$$Q_{ev,1} = Q_{ev,dc} = \frac{P_{b2,r}}{\eta_{o,0}} (1 - \eta_{th}) (1 + \xi_{dc}) \tag{7}$$

As only the refrigerant compressor (g1) of the first refrigeration cycle (dc) is operating during this period, the total power $P_{r,1}$ of the refrigerant compression is equal to the power $Pr_{,dc,1}$ of the refrigerant compressor (g1). Accordingly, the coefficient of performance $COP_{dc,1}$ of the refrigeration machine (dc) during this period is defined as:

$$COP_{dc,1} = \frac{Q_{ev,dc}}{Pr_{dc,1}} = \frac{Q_{ev,dc}}{Pr_1} \tag{8}$$

Combination of Eqs. (7) and (8) results in:

$$\frac{Pr_1}{P_{b2,r}} = \frac{Pr_{dc,1}}{P_{b2,r}} = \frac{(1 - \eta_{th,0}) (1 + \xi_{dc})}{COP_{dc,1}\ \eta_{o,0}} \tag{9}$$

The net power $P_{n,1}$ generated by the power plant and is available for the end users during the first period is given by:

$$P_{n,1} = P_{b2,r} + P_{b1,a} - Pr_1 \tag{10}$$

Dividing both sides of Eq. (7) by $P_{t,d}$ and then dividing both the nominator and dominator of the right hand side of the resulted equation by $P_{t,0}$ and combining it with Eqs. (1), (3) and (9), the following equation can be derived:

$$\frac{P_{n,1}}{P_{t,a}} = R \left\{ \begin{array}{c} \left[\left(\frac{h_1 - h_{2'}}{h_1 - h_{5'}}\right)^2 \left(\frac{h_1 - h_6}{h_1 - h_3}\right)\right] \\ \left[1 - \frac{(1 - \eta_{th,0}) (1 + \xi_{dc})}{\eta_{o,0}\ COP_{dc,1}}\right] - 1 \end{array} \right\} + 1 \tag{11}$$

During the second period (charging the cooling storage container), both refrigeration machines (dc) and (st) are operating in unison. The cooling generated by the refrigeration cycle (dc) serves to cool the steam condenser directly, whereas the cooling produced by the refrigeration cycle (st) is stored in the storing container (ℓ) to be used during the third period. Accordingly, the total heat rate $Q_{ev,2}$ to be removed by the evaporators of the refrigeration cycles (dc) and (st), and the total power Pr_2 of their refrigeration compressors are given by:

$$Q_{ev,2} = Q_{ev,dc} + Q_{ev,st} \tag{12}$$

$$Pr_2 = Pr_{dc,2} + Pr_{st} \tag{13}$$

where $Q_{ev,dc}$ is the same as that given by Eq. (7), and $Pr_{dc,2}$ can be expressed by analog with Eq. (9) as:

$$\frac{Pr_{dc,2}}{P_{b2,r}} = \frac{(1 - \eta_{th,0}) (1 + \xi_{dc})}{COP_{dc,2}\ \eta_{o,0}} \tag{14}$$

The coefficients of performance $COP_{dc,2}$ and COP_{st} of the refrigeration machines (dc) and (st) respectively, during the second period is given as:

$$COP_{dc,2} = \frac{Q_{ev,dc}}{Pr_{dc,2}} \tag{15}$$

$$COP_{st} = \frac{Q_{ev,st}}{P_{r,st}} \tag{16}$$

Given that the duration time of the second period is Δt_2, the total thermal energy $E_{ev,2}$ to be removed by the evaporators of the refrigeration machines through this period can be calculated as:

$$E_{ev,2} = E_{ev,dc} + E_{ev,st} = Q_{ev,2}\ \Delta t_2 = (Q_{ev,dc} + Q_{ev,st})\ \Delta t_2 \tag{17}$$

Insertion of Eqs. (12), (13), (15) and (16) into Eq. (17), it follows that:

$$E_{ev,2} = [COP_{st}\ (Pr_2 - Pr_{dc,2}) + COP_{dc,2}\ Pr_{dc,2}]\ \Delta t_2 \tag{18}$$

$E_{ev,dc}$ is equal to the heat energy removed by the refrigerant from the steam condenser (d) and the heat added to the cool refrigerant of the refrigeration machine (dc) from the surroundings over the second period. $E_{ev,dc}$ is determined with help of Eq. (7) by:

$$E_{ev,dc} = Q_{ev,dc}\ \Delta t_2 = \frac{P_{b2,r}}{\eta_{o,0}} (1 - \eta_{th,0}) (1 + \xi_{dc})\ \Delta t_2 \tag{19}$$

$E_{ev,st}$ equals the sum of the thermal energy removed from the steam condenser (d) through the third period and the heat transferred from the surroundings to the cooling refrigerant and the cooling storage container. Given that the duration time of the third period is Δt_3, $E_{ev,st}$ is calculated similarly to Eq. (19) by:

$$E_{ev,st} = \frac{P_{b2,r}}{\eta_{o,0}} (1 - \eta_{th,0})\,(1 + \xi_{st})\,\Delta t_3 \qquad (20)$$

where ξ_{st} is the cooling loss coefficient during storing the cooling and conveying it to the steam condenser (d).

Since $E_{ev,2}$ is the sum of $E_{ev,dc}$ and $E_{ev,st}$, it follows from Eqs. (18) and (19) that:

$$E_{ev,2} = \frac{P_{b2,r}}{\eta_{o,0}} [(1-\eta_{th,0})(1+\xi_{dc})\Delta t_2 + (1-\eta_{th})(1+\xi_{st})\Delta t_3] \qquad (21)$$

Equating the right hand sides of Eqs. (18) and (21) and solving for the power ratio $Pr_2/P_{b2,r}$, the following equation is obtained:

$$\frac{Pr_2}{P_{b2,r}} = \left\{\frac{1-\eta_{th,0}}{\eta_{o,0}}\right\} \left\{ \left(\frac{1+\xi_{dc}}{COP_{dc,2}}\right)\left(1-\frac{COP_{dc,2}}{COP_{st}}\right) + \frac{1}{COP_{st}}\left[(1+\xi_{dc}) + \frac{\Delta t_3}{\Delta t_2}(1+\xi_{st})\right] \right\} \qquad (22)$$

The net power $P_{n,2}$ of the steam plant during the second period is given by:

$$P_{n,2} = P_{b2,r} + P_{b1,a} - Pr_2 \qquad (23)$$

Using Eqs. (22) and (23) and following the same method of deriving Eq. (11), leads to obtaining the following equation is obtained:

$$\frac{P_{n,2}}{P_{t,d}} = \left\langle R \left(\frac{h_1 - h_{2'}}{h_1 - h_{5'}}\right)^2 \left(\frac{h_1 - h_6}{h_1 - h_3}\right)\right\rangle \left\langle 1 - \left\{\frac{1-\eta_{th,0}}{\eta_{0,0}}\right\} \left\{ \left(\frac{1+\xi_{dc}}{COP_{dc}}\right)\left(1-\frac{COP_{dc}}{COP_{st}}\right) + \frac{1}{COP_{st}}\left[(1+\xi_{dc}) + \frac{\Delta t_3}{\Delta t_2}(1+\xi_{st})\right] \right\} \right\rangle - R + 1 \qquad (24)$$

Concerning the third period (discharging the cooling storage container), it is worth mentioning that no energy is consumed for driving the refrigerant compressors. As a result, the net power output of the steam plant $P_{n,3}$ for this period is equal to the sum of the power developed by both turbines (b1) and (b2). Hence, it follows that:

$$P_{n,3} = P_{b2,r} + P_{b1,a} \qquad (25)$$

Eq. (25) can be worked out in an analogous method of deriving Eq. (11) for getting the power ratio $P_{n,3}/P_{t,a}$. Hence, the following equation is obtained:

$$\frac{P_{n,3}}{P_{t,a}} = R\left[\left(\frac{h_1 - h_{2'}}{h_1 - h_{5'}}\right)^2 \left(\frac{h_1 - h_6}{h_1 - h_3}\right) - 1\right] + 1 \qquad (26)$$

RESULTS AND DISCUSSION

The analyses described in the previous section were used to predict the energy output of the different configurations of combined cooling thermal storage and refrigerant and air cooled steam power plant under different ambient temperatures. The results presented hereafter are based on the basic design parameters of the steam power plant listed Table 1. It is to be noticed that the values of $\eta_{o,0}$ and $\eta_{th,0}$ given in table 1 are approximately close to those of a real modern steam plant. Also, the value of ξ_{st} in is valid for ice storage container used for inlet air cooling of gas turbines [16]. ξ_{dc} is assumed to be half the value of of ξ_{st}, since heat transferred from the surroundings to the cooling refrigerant during direct cooling is considerably less than that when the cooling is stored. The data given in table 1 was kept unchanged for al calculations carried out in this study. The thermal properties of water and steam were determined as a function of pressure and temperature by aid of the soft ware associated with the text book [17].

Table 1: Basic design data of the studied steam power plant

Boiler pressure	p_b	bar	150
Steam temperature at inlet to the turbines	T_1	^oC	500
Steam temperature in the refrigerant cooled condenser	T_c	^oC	20
Rise of steam temperature in the air cooled condenser over ambient temperature	ΔT_c	^oC	8
Steam plant overall efficiency at design condition	$\eta_{o,0}$	-	0.35
Thermal efficiency of Steam cycle at design condition	$\eta_{th,0}$	-	0.5
Cooling loss coefficient for direct cooling	ξ_{dc}	-	0.05
Cooling loss coefficient for cooling through cooling storage container	ξ_{st}	-	0.025

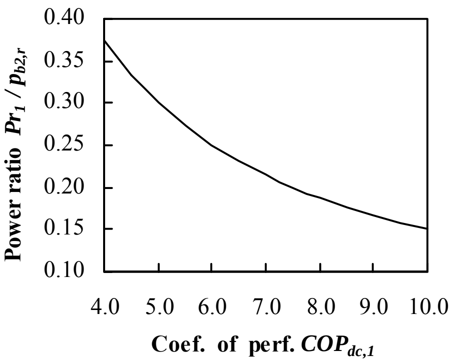

Fig. 3 Power of the direct cooling refrigerant compressor as a function of coefficient of performance

In Figure 3, the power ratio $Pr_1 / P_{b2,r}$ determined using Eq. (9) is plotted against the coefficient of performance $COP_{dc,1}$. It is to be noticed that $Pr_1 / P_{b2,r}$ is dependent on neither power fraction R nor ambient temperature T_a as pointed out by Eq. (9). Fig. 3 shows obviously that $Pr_1/P_{b2,r}$ is receding sharply with growing $COP_{dc,1}$, especially at low $COP_{dc,1}$. $Pr_1 / P_{b2,r}$ drops from 0.38 to 0.25, 0.19 and 0.15 when $COP_{dc,1}$ is decreased from 4.0, to 6.0, 8.0 and 10.0 respectively.

Eq. (11) suggests clearly that the power ratio $P_{n,1}/P_{t,a}$ is dependent on the $COP_{dc,1}$, T_a and R. $P_{n,1}/P_{t,a}$ is plotted in Figure 4 versus $COP_{dc,1}$ for R of 0.25, 0.5, 0.75 and 1.0. The diagrams of Fig. 4 represent T_a of 25, 35 and 45°C. Fig. 4 indicates clearly that $P_{n,1}/P_{t,a}$ ranges for the studied values of $COP_{dc,1}$, T_a and R from 0.69 to 1.08. Also, it indicates that the effect of R on $P_{n,1}/P_{t,a}$ is relatively small (it varies in a range less than 10%) at ambient temperatures lower than 35°C ($COP_{dc,1}$ anticipated to be higher than 7). For ambient temperature greater than 35°C ($COP_{dc,1}$ is expected to be less than 7), the deviation in $P_{n,1}/P_{t,a}$ for the different values of R diminishes. Accordingly, the value of R has relatively little effect on the plant net power output to be sent to the grid at the first period for practically possible operating conditions.

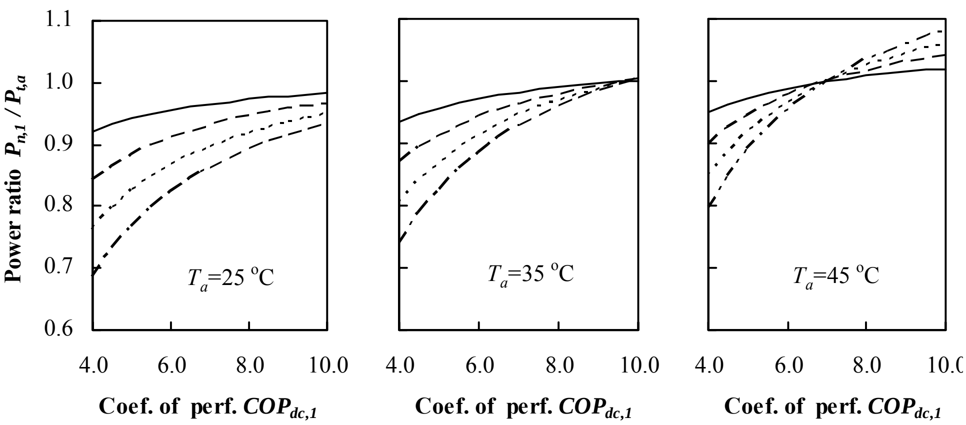

Fig. 4 Effect of the coefficient of performance on the net power output of the steam plant during direct cooling of the refrigerant-cooled condenser

——— R=0.25 — — — R=0.5 .... R=0.75 —.—.— R=1.0

In Figure 5, the power ratio $Pr_2 / P_{b2,r}$ calculated by aid of Eq. (22) is plotted versus COP_{st} for $\Delta t_2 / \Delta t_3$ of 1.0, 2.0, and 3.0 and for $COP_{dc,2}$ of 4, 7 and 10. $Pr_2 / P_{b2,r}$ is independent of both power fraction R and ambient temperature T_a, as denoted by Eq. (22). It is to be noticed here that the cooling generated by the two refrigeration cycles is used for two purposes; i.e. direct cooling of the refrigerant condenser and storing the cooling thermally for later use at peak loads. Accordingly, the power of the refrigerant compressors is relatively high for this period and it may exceed the power of the refrigerant cooled turbine at low values of COP_{st} and $\Delta t_2 / \Delta t_3$. To make this clearer, the power ratio $P_{n,2} / P_{t,a}$ calculated by using Eq. (24) is drawn in Figure 6 against COP_{st} for $\Delta t_2 / \Delta t_3$ of 1.0, 2.0 and 3.0 and for R of 0.25, 0.5, 0.75 and 1.0 at T_a of 40°C and for $COP_{dc,2}$ of 6. Fig. 6 shows that $P_{n,2} / P_{t,a}$ is raised sharply with increasing COP_{st} for all values of R and $\Delta t_2 / \Delta t_3$. To assure the availability of enough portion of the plant power for end users, R should not exceed 0.5, and $\Delta t_2/\Delta t_3$ has to be greater than 2.0, when COP_{st} is less than 2. Should R be selected greater than 0.5, $\Delta t_2 / \Delta t_3$ has to be greater than 2, when COP is less than 2. This makes more than 50% of the generated power available for end users.

It is worthy mentioning here that cooling is stored as ice, chilled water or some other materials. As a tremendous amount of heat must be removed from the refrigerant cooled condenser, ice storing is preferred to limit the volume of the storing container. Production of ice forces the cooling storage refrigerating machine to operate at relatively low evaporator temperature which effects low coefficient of performance COP_{st}. Fortunately, the storing process takes place when the end users demand is minimum that occurs at night hours and early morning, where the ambient temperature has its lowest value. This makes the temperature of the condenser of the refrigerating machine relatively low that partially compensate for the low temperature of the evaporator by remarkably improving the coefficient of performance COP_{st}.

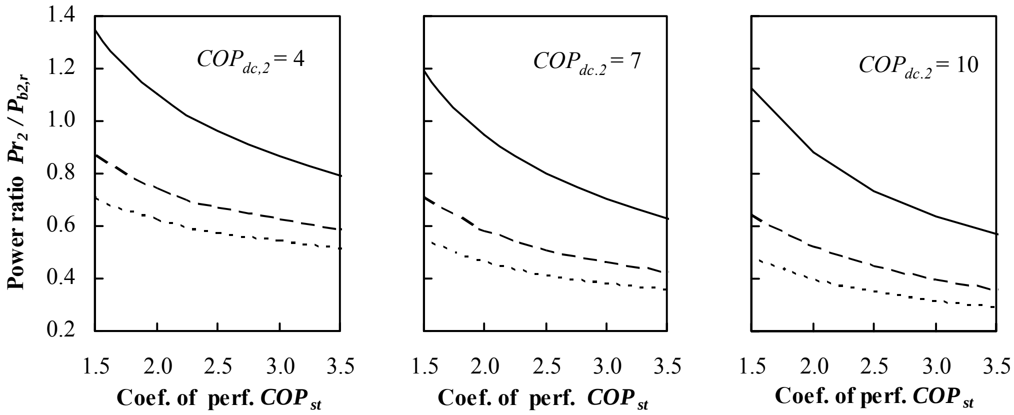

Fig. 5 Average power of the refrigerant compressors during cooling thermal storage

——— $\Delta t_2/\Delta t_3$=1.0 – – – $\Delta t_2/\Delta t_3$=2.0 .... $\Delta t_2/\Delta t_3$=3.0

As for the capacity of the cooling storage container, it has been estimated on storing ice to be Ca. 16 m^3 for each MW of the refrigerant cooled turbine and one hour of the peak-loads. This estimation has been based on calculating the heat energy to be removed from the cooling storage container by using Eq. (21) and considering the storage time is zero and the peak-load time equals one hour. From this energy the amount of ice formed is calculated.

For showing the effect of using cooling storage container with double refrigerant and air cooled steam plant, the power ratio $P_{n,3}/P_{t,a}$ determined using Eq. (26) is depicted in Figure 7 versus the ambient temperature T_a. The curves in Fig. 7 represent power fraction R of 0.25, 0.5, 0.75 and 1.0. Fig. 7 reveals that $P_{n,3}/P_{t,a}$ is increased with rising ambient temperature T_a for all values of R. This can be interpreted as follows: As T_a rises, both the condensation temperature and pressure in the air cooled condenser are elevated and the turbine power $P_{t,a}$ is reduced as indicated by Eq. (3). This results in raising the ratio $P_{n,3}/P_{t,a}$. Also, Fig. 7 shows that the rise in $P_{n,3}/P_{t,a}$ with T_a as R is increased is enhanced. This is ascribed to the increasing power $P_{b2,r}$ of the refrigerant cooled turbine, and in turn, growing power $P_{n,3}$ of the plant. The power ratio $P_{n,3}/P_{t,a}$ is raised from 1.0 at T_a equals 12^{o}C to 1.034, 1.069, 1.10 and 1.14 at T_a of 30^{o}C for R equals 0.25, 0.5, 0.75 and 1.0 respectively. $P_{n,3}/P_{t,a}$ is further increased to 1.08, 1.16, 1.24 and 1.32 as T_a reaches 50^{o}C for R having the value of 0.25, 0.5, 0.75 and 1.0 respectively.

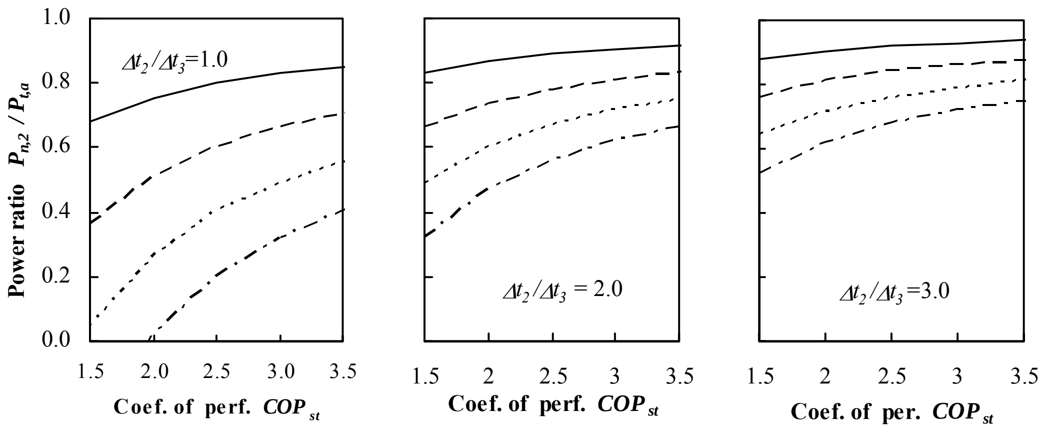

Fig. 6 Average net total power of the steam plant during cooling thermal storage ($COP_{dc,2}$ = 6 and T_a = 40 ^{o}C)

——— R= 0.25 – – – R=0.5 .... R=0.75 –.–.– R=1.0

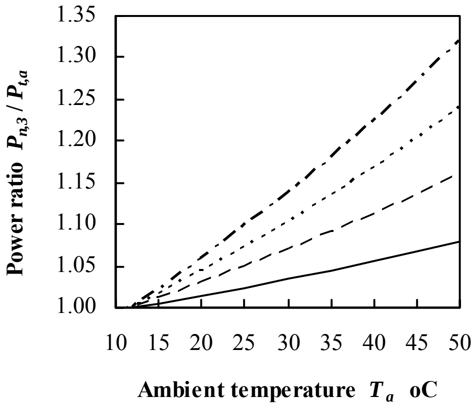

Fig. 7 Plant net power output during discharging the cooling storage container (peak-loads) as a function of ambient temperature

——— R= 0.25   – – – R=0.5
………… R=0.75   –.–.– R=1.0

CONCLUDING REMARKS

In the current paper, energy analyses for combined cooling storage container and double refrigerant and air cooled steam power plant are presented. The results of these analyses for constant temperature of 20°C in the refrigerant cooled condenser led to drawing the following Conclusions:

1. The net power output of the different configurations of the proposed system is always greater at peak loads than using air cooled condenser alone.
2. The increase in the power is dependent on the ambient temperature and power factor which represents the ratio of the powers of the refrigerant cooled turbine to air cooled condenser at condenser design temperature.
3. The maximal increase in the plant is achieved when the whole exhaust steam of both turbines is condensed in a refrigerant cooled condenser. In this case, the plant power rises by 6, 14, 22, and 32% of plant power with only air condenser during peak-loads for ambient temperature of 20, 30, 40 and 50°C respectively.
4. During the exclusive direct cooling of the refrigerant cooled condenser, the plant net power output is almost greater than 80 % of the total power generated by the two steam turbines for most of practically possible operating conditions.
5. The power fraction should be chosen less than 0.5 and the time ratio of charging to discharging the cooling storage container has to be at least 2 to assure more than 50% of the plant generated power available for end users if the coefficient of performance of the cooling storage refrigeration machine is less than 2.
6. For increasing the power fraction, when the coefficient of performance of the cooling storage refrigeration machine is less than 2, the time ratio should be made greater than 2.

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