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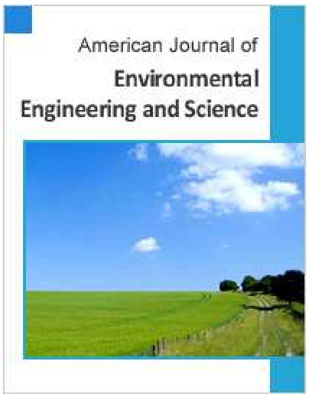

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Experimental Investigation for a Hybrid Arrangement of Steam Condensers

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Abstract
This work represents an experimental investigation to assess the thermal performance of a hybrid condensation system at atmospheric pressure. A combination of the air cooled (ACC) and shell and tube (WCC) heat exchangers were used to show the steam distribution percentage and capability of each of these heat exchangers. When the air flow rate was doubled and its entering dry bulb temperature reduced from (42) to (21) °C, the (ACC) average steam loading and thermal load were increased by (22%). Pre-cooling of air gave an increase in (ACC) steam mass flow rate of (0.58-0.66) kg/hr per each degree reduction of air dry bulb temperature from (37.5°C) to (27°C) for constant air mass flow rate and surface area. The corresponding values of condensation thermal load variations were within (1.28-1.33) times.

1. Introduction

The steam condenser is one of the major components of the steam cycle in steam power plant. It condenses the exhaust steam from the turbine and thus recovers the high-quality feed water for return to the boiler and increases the cycle efficiency, Matthew (2002) [1]. The condensate being recycled back through the system by three major condensation systems, water cooled, air cooled and alternative condensing systems. Larinoff and Moles (1978) [2] investigated the effect of the ambient air temperature on the turbine backpressure and recommended the range from (16) to (43) ^{o}C as limitation for the turbine corresponding back pressure. Jabardo and Mamani (2003) [3], investigated experimentally and theoretical the condensation process for automotive cooling application. The condenser simulation model assumes that the heat transfer surface is divided into three regions: desuperheating, condensing and sub-cooling, each region was treated as an independent heat exchanger. The discrepancy between the experimental data and the simulation results has shown a good agreement.

Tarrad et. al., (2008) [4], and Tarrad et. al., (2009) [5], investigated experimentally and theoretically the performance of air cooled heat exchangers. Their results showed that the load was increased by approximating (3) times with air velocity raised from (1.2) to (4.6) m/s for the test range. Further, the overall performance of the heat exchanger is a dependent measure on the tube bank geometry and arrangement and operating conditions. Tarrad (2010) [6], built a computer code that depends on the idea of using the row by row technique for estimating the heat transfer coefficient , air temperature and air physical properties distribution in the air flow direction from row to row. The model results showed an improvement in the condensation load up to (23%) when air pre-cooling mode applied to inflow air to the ACC to lower the dry-bulb temperature from (45) to (28) ^{o}C.

Tarrad and Kemal (2001) [7] and Tarrad and Majeed (2007) [8] investigated a step by step method for thermal and hydraulic design of steam power plant condensers for a single and two passes. A good agreement was shown between the field data of thermal power plants in Baghdad city and the simulation model. Tarrad (2007) [9], provided a model as a computer code for the thermal-hydraulic design of a single shell-single pass of enhanced tube bundle heat exchanger using the step by step technique. The model basic philosophy depends on the thermal and economical aspects by selecting the low finned tube geometry. This was to avoid the space restriction of the equipment layout in the field application and to minimize the cost of manufacturing machining. Heyns (2008) [10], Studied the performance characteristics of a power plant incorporating a steam turbine and a direct air cooled dry/wet condenser operating at different ambient temperatures. The study showed that for the same turbine power output, the water consumed by an air cooled condenser with a hybrid arrangement is at least (20 %) less than an air cooled condenser with spray cooling of the inlet air.

NREL (2011) [11], identified and evaluate methods by which the net power output of an air-cooled geothermal power plant can be enhanced during hot ambient conditions using minimal amounts of water. Geothermal power plants that use air-cooled heat rejection systems experience a decrease in power production during hot periods of the day, therefore hybrid cooling options, which use both air and water, have been studied to assess how they might mitigate the net power decrease in hot periods. Baweja and Bartaria (2013) [12] have concluded in their review the most important factors which control the performance of the air cooled condenser. It is found that there is degradation in performance of air cooled condenser under high ambient temperatures and windy conditions. The heat rejection rate of ACC also depends on surface condition of fins and thus its performance is reduced due to external fouling of finned tubes due to weather conditions and by internal fouling from condensate (Ammonia corrosion).

More recently, Tarrad and Khudor (2015) [13] have published quite a simplified correlation in this field. They accomplished a correlation for the air side heat transfer coefficient of compact heat exchangers in the form of a dimensional analysis. They concluded that their correlation has estimated the heat duty and overall heat transfer coefficient of the case study heat exchangers with total mean absolute errors of (13%) and (10%) respectively. Tarrad and Al-Nadawi (2015) [14] suggested a model for the rating of air cooled condenser in a refrigeration unit. The tube by tube technique was implemented in a window type air conditioning unit circulating different refrigerant such as R-22, R-407C and R-407A. They postulated that the predicted heat duty of these refrigerants by their model has showed excellent agreement and was within the range of (-5%) and (+1.7%).

In the present study, an experimental investigation will be described for the use of a hybrid condensation system arrangement. The implementation of this system within a frame work of steam condensation in both of the service fluids types will be clarified with the aid of the collected data. Two vertically oriented condensers were combined in parallel with saturated steam supplied at atmospheric pressure. Each condenser can work individually with a specified steam loading regarding to the steam generator feeding. The combined mode for the water cooled condenser under a low cooling water flow rate is considered as a new proposed method to investigate the air cooled condenser capability in a hot ambient condition. Also evaporative cooling for air inflow is used to lower the air dry-bulb temperature occurs during summer sever environment.

2. Experimental Rig and Procedure

2.1. Test Rig

An experimental facility was constructed to allow two types of condensing systems worked as a test arrangement. Each one represents a separate unit having all the specifications and instruments which allows condensation data to be collected over a range of operating conditions. Air cooled condenser, finned tube heat exchanger, and shell and tube heat exchanger were implemented as a single or in a hybrid arrangement as shown in figure 1.

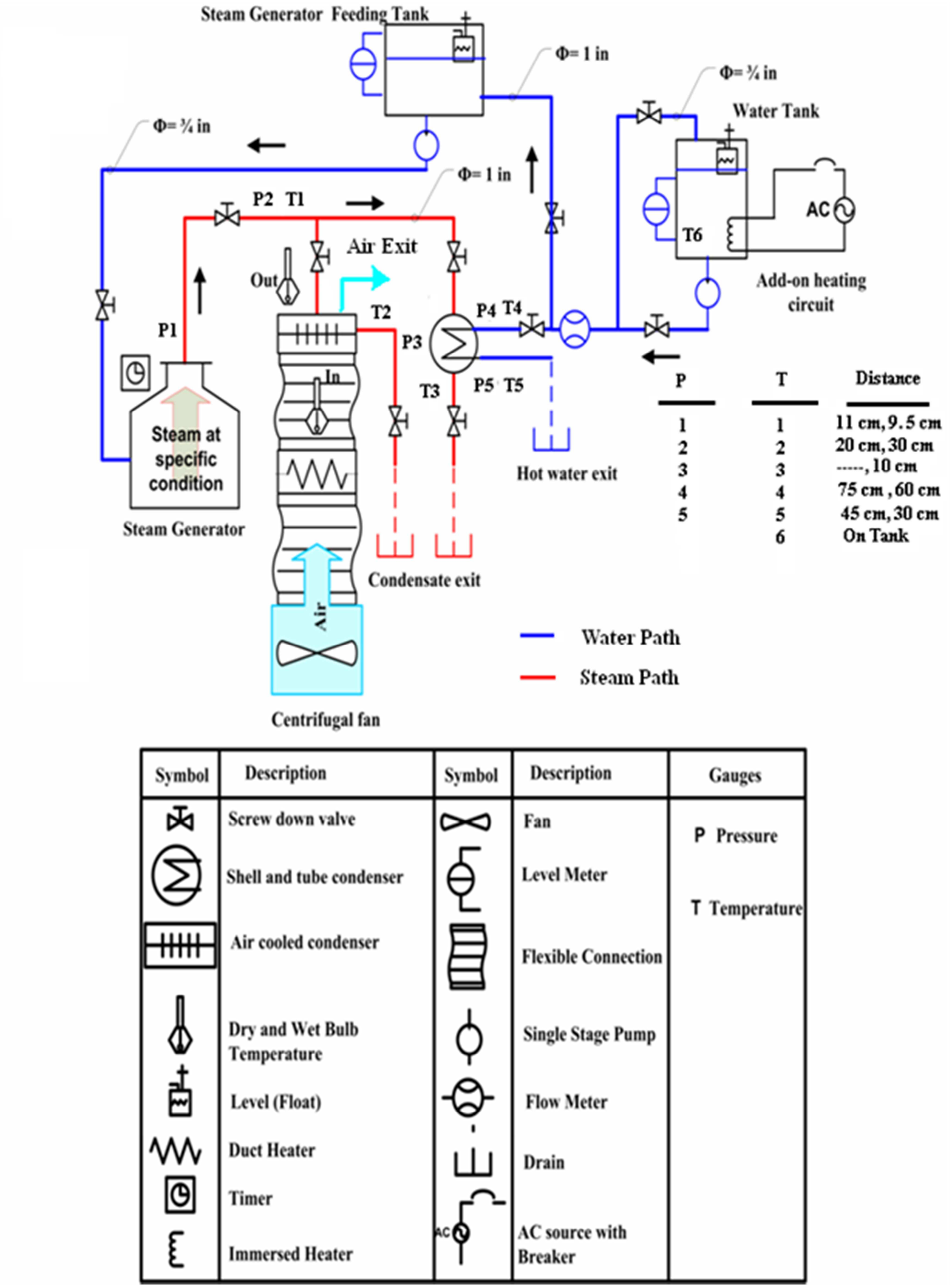

Figure (1). *A Schematic Diagram for the Test Rig.*

2.2. Steam Generator (SG)

The steam generator is constructed in a way that it has the ability to provide wet steam at atmospheric pressure. It was made of galvanized steel sheet of (3) mm wall thickness to ensure operating under high pressure safely. However, the pressure did not exceed (2) bar in the present study. The steam generator core capacity is about (120) litter. Water is pumped to the (SG) with a specific quantity, which must be suitably covering (at height of 50 to 60 cm) the heating element inside (SG). This can be checked by the sight glass level (high temperature proof) installed on the (SG), with a continuous steam flow production at low pressure of (1) atm. A safety valve is fixed at the top of vessel and pressure gauge is installed for pressure measurement and monitoring. Figure (2) illustrates the construction of the steam generator.

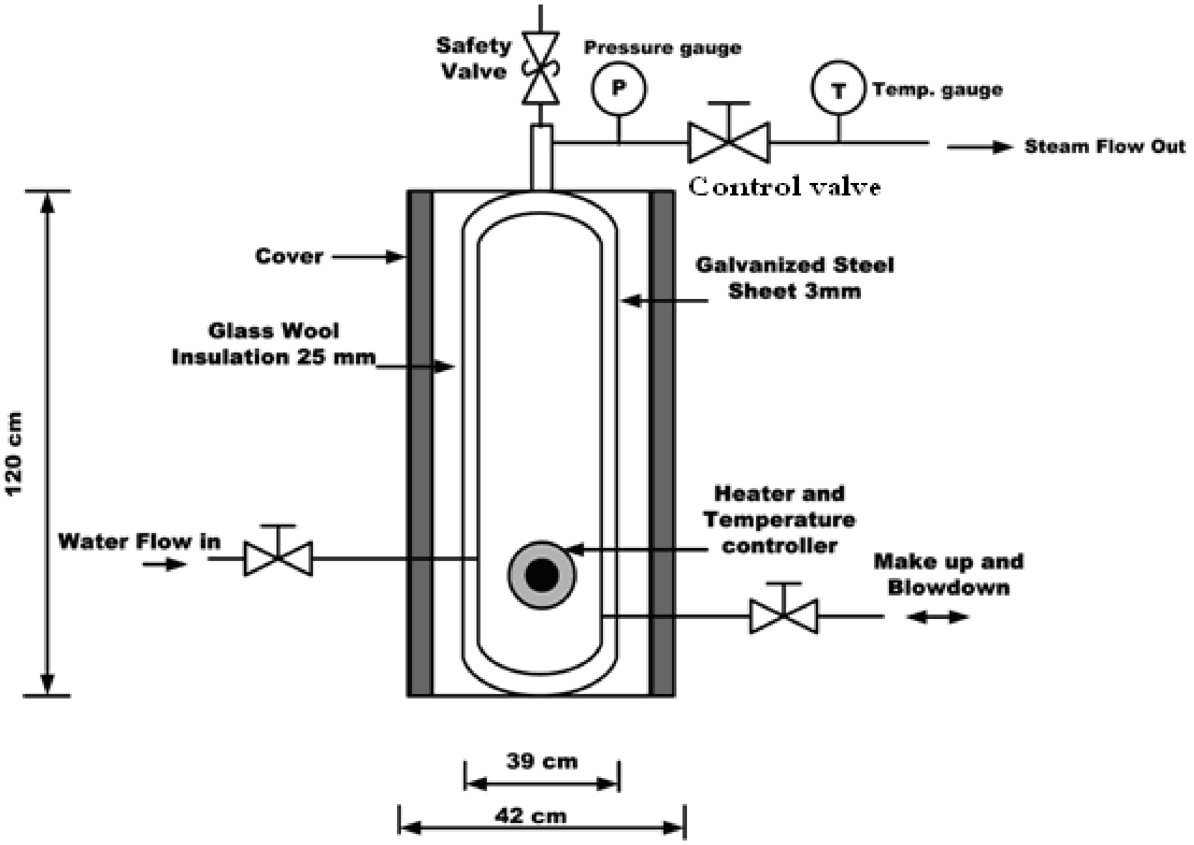

Figure (2). *The Steam Generator Used in the Present Experimental Set-up.*

2.3. Shell and Tube Condenser

The vertical orientation shell side condenser is made of (325) mm tube effective length occupied by (28) stainless steel tube. Each tube is (9.45 mm) outer diameter and (8.68 mm) inner diameter, as shown in figure (3). The tubes are distributed as a rotated square of (45^o) tube layout. The clearance between two adjacent tubes is (3) mm, and the tubes pitch is (12.94) mm. The shell diameter is (81.64) mm. The baffle space is (70) mm and a baffle cut of (25%). The shell side inlet and outlet nozzles are of (42.5) mm, the tube side inlet and outlet connections are of (50.8) mm, and the total heat transfer area is (0.270) m^2. Instruments for temperature and pressure are connected to the tubes and shell sections. The gauges are fitted in prepared pockets on the required locations.

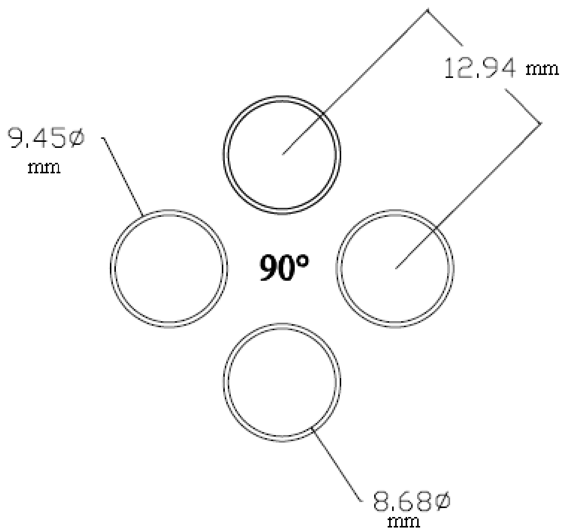

Figure (3). *Shell and Tube Heat Exchanger Tube layout.*

2.4. Air Cooled Condenser (ACC)

The ACC implemented in this work was a vertical type of a single pass and two rows with tubes having flat sides occupied each row with equal numbers, as shown in figure (4). The centrifugal fan installed in front of the ACC to move air towards finned tube bundle. The flat sides of the tube geometry used to enhance heat transfer inside tube with extra heat transfer area and reduce pressure drop outside tube. The tubes are made of a brass metal which has excellent physical properties and copper fins. The fins are constructed to the tubes so each fins row contains (70) fins attached to tube in equal pitch about (4.18) mm. The fins physical characteristic and heat exchanger dimensions are illustrated in table (1). The whole assembly is mounted on legs with a rubber and fasteners to well-set during operation.

Table (1). *ACC Geometry and Physical Characteristic.*

Parameter		Dimension
Core	Width (W)	590 mm
	Depth (D)	30 mm
	Height (H)	320 mm
	No. of Tubes	110
	No. of Tubes/Row	55
	No. of Rows	2
	No. of fins/ Tube (on both sides)	140
	Transverse Distance (X_T) to flow	11.25 mm
	Longitudinal Distance (X_L) to flow	15 mm
	Frontal Area (A_{face})	0.1888 m^2
Fin	Pitch (f_P)	4.18 mm
	Length (l_f)	8.05 mm
	Thickness(t_f)	0.18 mm
	Area of a Single Fin (A_f)	36.12 mm^2
	Material	Copper
	Thermal Conductivity (W/m.C), [15]	388
Tube	Height (H_t)	2.35 mm
	Depth (D_t)	12 mm
	Thickness (t_t)	0.24 mm
	Material	Brass
	Inner Tube Surface	Smooth
	Thermal Conductivity (W/m.ºC), [15]	119
Area	Total Surface Area (A_{total})	3.935 m^2
	Total Fin Surface Area	3.029 m^2
	Total Bare Tube Area	0.906 m^2

The air supplied to the ACC through a flexible ducting system. As a facility to control the air inlet temperature, the duct heating part of (165 x 310) mm^2 is added to the ductwork system. It has (10) kW coiled heating element operates at (220-240) volt, situated inside the entering air construction side with suspended screws to adjust heater to the duct center. The sides in the front of the heater were covered with special design wings to prevent air by-passing and to direct air toward the heating section. This technique was used to heat up the air to the desired test value and represents the effect of temperature rise during summer hot environment.

The ACC fan is of the centrifugal type with forward blade configuration. This is a part of evaporative air cooler construction commercially available in the market. The delivered air volumetric flow rate has two different values either (1200) or (2400) *cfm*, which can be controlled through a variable speed electrical motor. This air cooler was also used to control and reduce the entering air temperature to the ACC as required during the tests.

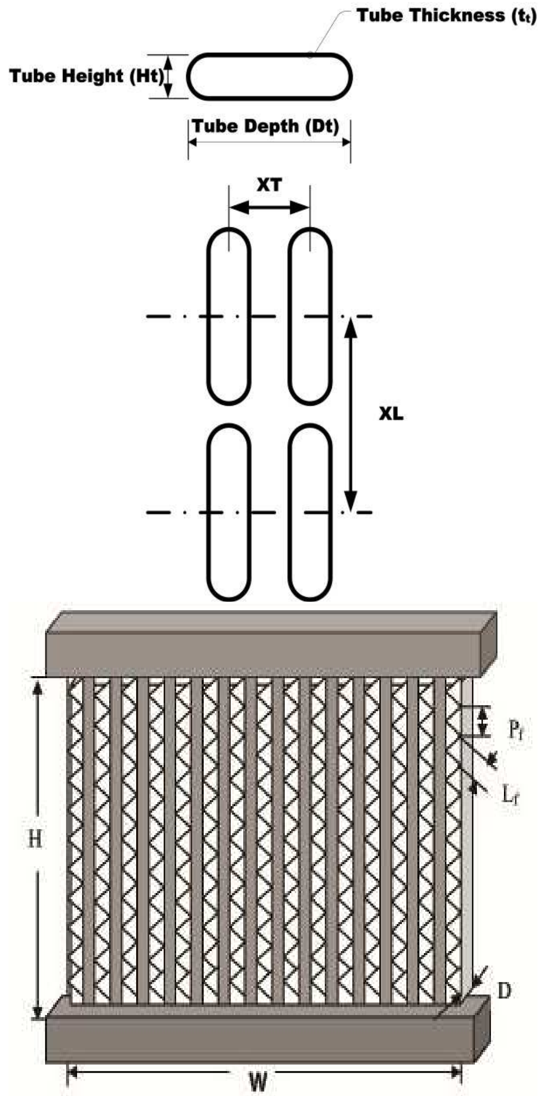

Figure (4). *Air Cooled Heat Exchanger Characteristics.*

2.5. Water Loops

There are two main water loops used in the present work arrangement. The first loop represents the cooling service fluid flow through the tube side of the WCC while the other was used to feed the steam generator.

On the WCC side, the water was used as a cooling medium removing the latent heat from the condensing steam. Water was pumped by a centrifugal pump having an electrical motor of (370) watt having (2900) *rpm* at (220-240) volt. It has a maximum capacity of (30) *l/min* and (30) m of water head. The pump takes the water from main reservoir with (200) liter capacity. Another reservoir equipped with heater (3000-6000) watts was used with ability of heating to operate as an emergency tank when water cut-off occurs and to give warm water depending on the experiments conducted, figure

(1). On the steam generator feeding side, the water was pumped to the intake portion of steam generator with a limited quantity. This measure was taken to insure the safety of the heating element and will minimize the time required for steam generation.

A hand controlled valve is used to adjust the water flow rate through the test section WCC and ACC. The flow rate of the water through the test sections was measured with variable area flow meter. Water that leaves the WCC section was drained to sewage or was returned to the emergency tank. Over a range of experimental sets, water in both steam generator and water supply tanks must be blown down to prevent scale forming and corrosion reactions.

The water cooled condenser, condensate tank, and the hot side piping system are all closely insulated with (1) inch glass wool insulation having (k=0.035) W/m.^{o}C. This measure was taken to minimize the heat loss to the surrounding and human safety.

2.6. Instrumentation

2.6.1. Temperature Measurements

Five stem glass thermometers were installed in the water and steam side to measure the temperature of steam (process fluid) and water (service fluid) at different locations. The technique is used by installing the thermometers with a prepared jacket to immerse the thermometer through the flow by using screwed fittings. This method confirms the accurate measured values for temperature measurements that it provides a direct contact between the bulb and fluid.

Temperature gauges have a range of (0 – 120) ^{o}C at a division of (1) ^{o}C, maximum and minimum errors are corresponding to (2%) and (-1.2%) respectively. Digital thermocouples of the K-type were used to measure the temperature on the inlet and exit sides of ACC condenser. The wet bulb temperature of air entering the condenser was measured by using the wet cotton wick at the bulb of thermometer sensor.

2.6.2. Pressure Measurements

Five stem glass pressure gauges were used to measure the pressure at each exit and inlet ports of service fluid (water) and also at the inlet side of process fluid (steam) of ACC, and on WCC. No pressure gauges were needed for the exit side of steam loops at the condensate side because the system is opened to atmosphere. One gauge was installed on the steam generator to control the buildup of the inside pressure with presence of safety valve at steam generator exit point. Two of the pressure gauges on the water side have ranges of (0-2.5) bar. The division (0.05) bar, maximum and minimum errors are (0) and (-0.3) respectively, which were connected to the tube side of the water cooled condenser. The last one was installed on the inlet to the air cooled condenser and has a range of (0 – 2.5) bar with a division (0.05) bar and maximum error is about (2 %).

2.6.3. Water Flow Rate Measurements

A Rotameter (KDG 2000) was used to measure the flow rate of the circulating water in (*l/hr*) has a range of (200 – 1800). Calibration of the Rotameter has shown an error of about (1 %) *l/hr*.

2.6.4. Uncertainty of Measurements

The uncertainty percentage for the air side heat transfer mode of the measurement was estimated to be within (± 2 %) for the whole tests range in this work. Whereas, the corresponding uncertainty for the parameters measured of the water side tests was within (± 1.5 %). This will produce a mean total uncertainty of the thermal category in measurements for the hybrid system arrangement during the present work to be within (± 2 %). For the hydrodynamic side, the corresponding mean uncertainty in the measured parameters was within (± 1 %).

More details about the experimental rig, instrumentation and electrical board installations can be found form Altameemi (2011) [16].

3. Experimental Results and Discussion

3.1. Air Cooled Condenser Mode

Figure (5) shows the steam loading variation for the ACC at different entering air temperature for two different air flow rates namely (1200) and (2400) *cfm*. It is obvious that the load capacity of the condenser increases with entering temperature reduction or increasing air flow rate. This is a result of increasing the temperature driving force (ΔT_m) or improving the air side heat transfer coefficient which in turn increases the overall heat transfer coefficient of the heat exchanger.

Similarly the thermal load variation with operating conditions of the air side of the condenser is illustrated in figure (6). The same behavior was observed for these measures as those of the steam loading. The thermal load of the condenser showed an enhancement as the entering air temperature reduced or the air flow rate was doubled. The results revealed that at air velocity of (3) m/s, the highest expected load from the air cooled condenser at $T_{a,i}$ of (31) °C was (16.5) kW. The heat transfer rate experienced a gradual reduction as the $T_{a,i}$ was increased to (42) °C at which the achieved thermal load approached (12.3) kW.

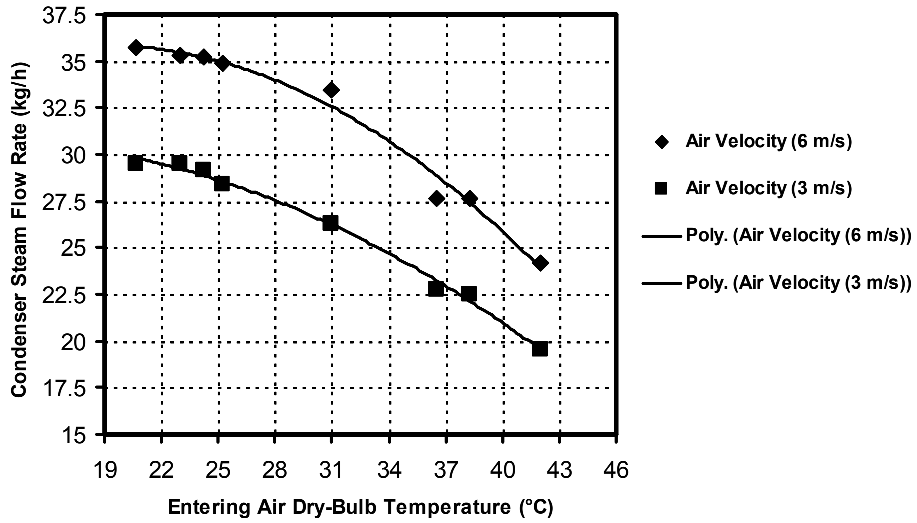

Figure (5). *The Condenser Steam Loading Variation with Entering Air Dry-Bulb Temperature.*

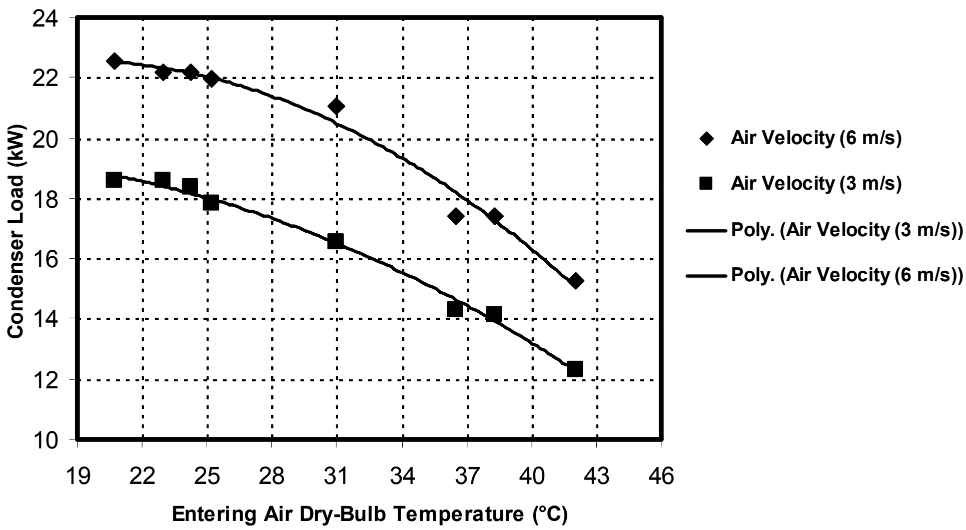

Figure (6). *Air Cooled Condenser Load Variation with Entering Air Dry-Bulb Temperature.*

The reproducibility of the experimental data was established by taking a number of readings at the same operating conditions for the air side as shown in figure (7) at air velocity of (3) *m/s*. The condensate rate measured during these test showed a sub-cooling condition by a value of (2) °C. However, the data revealed an acceptable consistency and can be represented by a straight line as shown in the figure.

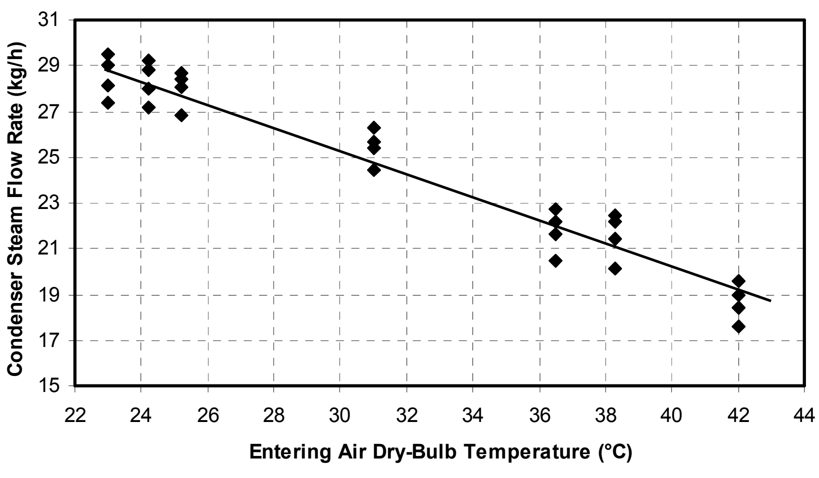

Figure (7). *The Air Cooled Condenser Steam Loading Variation with Entering Air Dry-Bulb Temperature.*

3.2. Water Cooled Condenser Mode

In the WCC mode, the increase in cooling water mass flow rate causes an increase in both the steam loading and the condenser thermal load. The steam condensation rate was increased by (2.2, 1.84 and 1.95) times with increasing cooling water flow rate from (200) to (1000) *l/hr*, at water inlet temperatures of (15, 19 and 23) °C respectively as shown in figure (8).

Figure (9) shows the variation of the condenser thermal load with entering water temperature. It emphasizes that the thermal load experienced an increase of (49.4 %), (43.3%) and (38.2%) when the entering temperature was (15), (19) and (23) °C respectively as water flow rate was increased from (200) to (1000) *l/hr*. It is obvious that the steam loading and condenser thermal load increase as the cooling water temperature reduces or its flow rate increases. This is mainly proportional to the effective temperature driving force and augmentation in the overall heat transfer coefficient of the condenser.

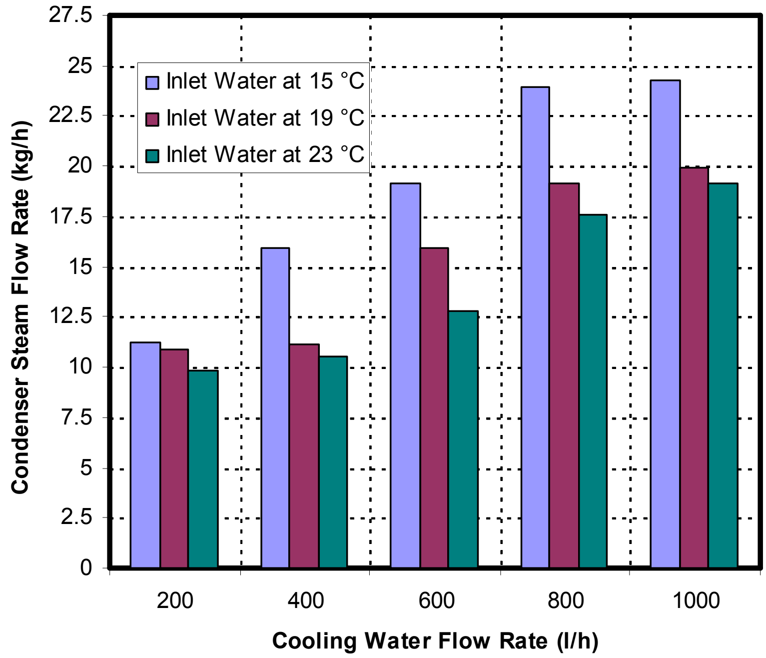

Figure (8). *The Water Cooled Condenser Steam Loading Variation with Cooling Water Flow Rate at Different Entering Temperatures.*

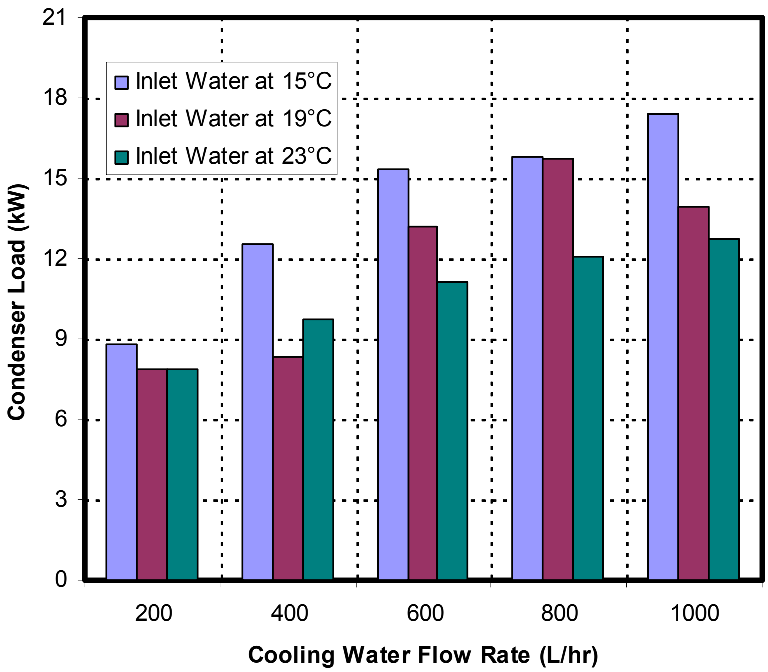

Figure (9). *The Water Cooled Condenser Thermal Load Variation with Cooling Water Flow Rate at Different Entering Temperatures.*

3.3. Hybrid System Mode

The experimental conditions for the hybrid system arrangement were constricted upon the summer environment nature and water shortage. Therefore, the entering cooling water temperature to the shell and tube condenser was limited to (30) ^oC as recommended by Kemal [17] and Woodruff [18]. The water flow rate such as (200) *l/hr* was considered as the lowest possible available flow for the simulation object as a water shortage condition. The latter revealed a low value of Reynolds number at the range of (1000) and hence to be in the laminar flow regime where a low heat transfer rate is expected.

A flexible entering air temperature for the air cooled condenser was considered in the range (31-42) ^oC with a fixed air flow rate of (1200) *cfm*. This air flow rate revealed a face velocity of (3) *m/s* which is recommended for the design of air cooled heat exchangers. The air velocity is an important character in the overall design of such type of heat exchangers. It is usually bounded within the range of (1.5) to (4) *m/s*, Tarrad (2010) [6]. It was recommended by Wilber and Zammit (2005) [19] that the air face velocity should not exceed (3) *m/s*, for pressure drop consideration.

Table (2) shows a summary for the test conditions conducted for the hybrid system at (30) ^oC entering temperature and (200) *l/hr* for water side and (1200) *cfm* air flow operating conditions.

Table (2). *Tests Operating Conditions and Load Distribution in the Hybrid System.*

Case No.	$T_{a,i}$ (^oC)	$T_{w,i}$ (^oC)	$\dot{Q}_{ACC}$ (kW)	$\dot{Q}_{WCC}$ (kW)	$\dot{Q}_t$ (kW)	η_{ACC}	η_{WCC}	$\dot{m}_t$ (*kg/hr*)	$\dot{Q}_{ACS}$ (kW)
1-	31	30	12.6	4.9	17.5	72%	28%	26.36	16.5
2-	36.5	30	11.4	5.3	16.7	68%	32%	25	14.3
3-	38	30	11.6	4.9	16.5	70%	30%	24.9	14
4-	42	30	10.3	5.3	15.8	65%	35%	22.8	12.3

In this table the thermal load and load percentage were calculated from:

$$\dot{Q}_{ACC} = \dot{m}_{ACC} \times (h_g - h_l) \quad (1)$$

$$\dot{Q}_{WCC} = \dot{m}_{WCC} \times (h_g - h_l) \quad (2)$$

$$\eta_{ACC} = \frac{\dot{Q}_{ACC}}{\dot{Q}_{total}} \quad (3)$$

And similarly

$$\eta_{WCC} = \frac{\dot{Q}_{WCC}}{\dot{Q}_{total}} \quad (4)$$

Here $\dot{Q}_{total}$ is the total load of the hybrid system. The condensate flow rate $\dot{m}_t$ was measured experimentally as the sum of the partial condensate of both condensers in the hybrid system arrangement.

The data showed that a general trend exhibited an increase of the total condensation load with reducing of the entering air temperature of the air cooled condenser. The load revealed by the ACC was higher than that of the WCC regardless of the entering air temperature and approached a maximum at (31) ^oC. The same trend was reflected on the condensation flow rate of each condenser where it was higher for the ACC than that of the WCC as well.

Figure (10) shows the variation of the partial load contribution of both condensers with entering air temperature. It is obvious that the ACC load decreases with the air temperature rise where it is compensated by the WCC mode. The corresponding contribution percentage for the water cooled condenser were about (35%) and (28%) at the entering air dry-bulb temperature of (42) °C and (31) °C respectively. The increase of the water cooled condenser load during the high ambient temperature was due to the self-balancing of the system. Therefore, a hybrid system acts as self-control for the steam loading through each condenser as the surrounding conditions are altered. The condensate loading of both condensers is illustrated in figure (11) as a bar chart.

A test was conducted to show the effect of increasing of the cooling water flow rate on the total performance with keeping the other operating conditions fixed as that for (200) *l/hr* water flow rate. Figure (12) shows the steam loading distribution among the hybrid system condensers for (400) *l/hr* water flow rate. This measure didn't show a significant impact on the whole process as revealed from the bar chart.

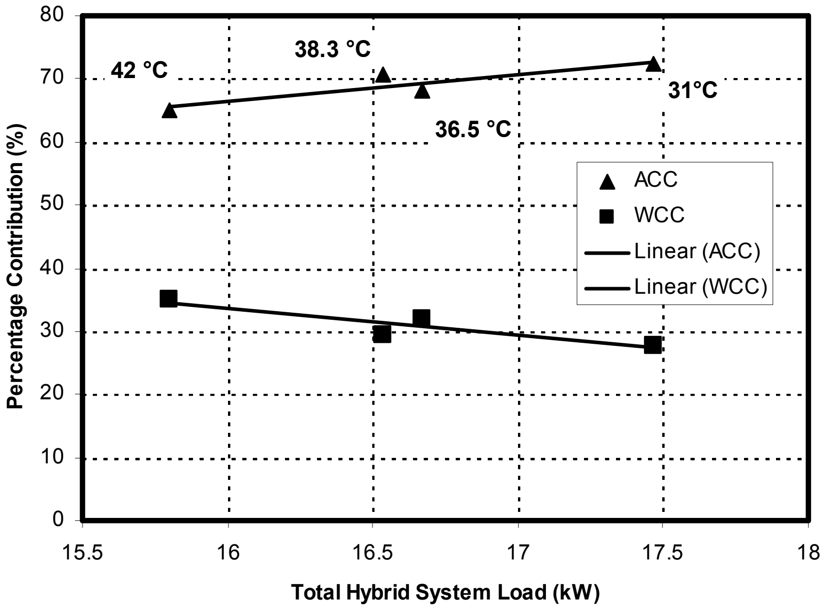

Figure (10). *Thermal Load Contribution of the ACC and WCC in the Hybrid System Arrangement at Water Flow Rate of (200) l/hr.*

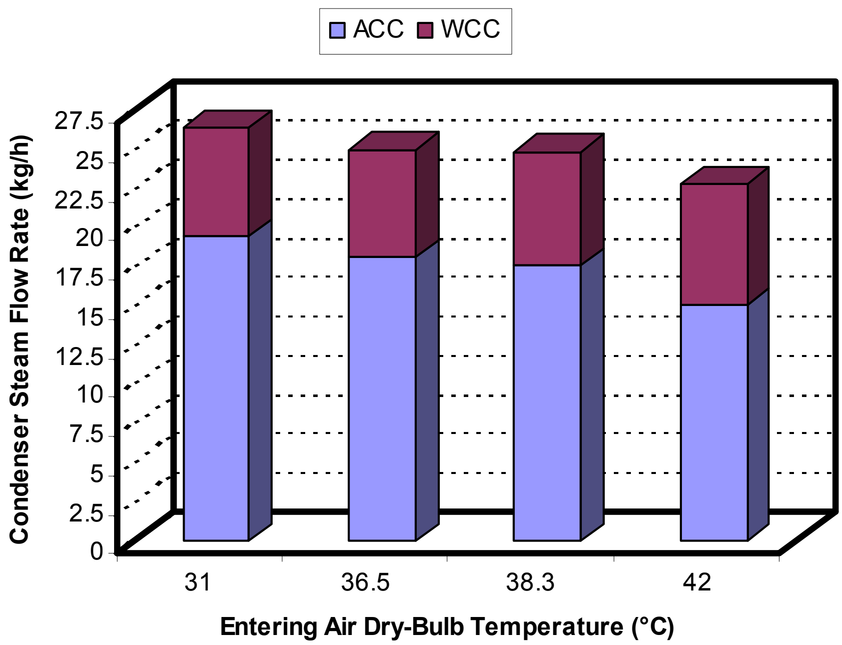

Figure (11). *Steam Load Contribution of the ACC and WCC in the Hybrid System Arrangement at Water Flow Rate of (200) l/hr.*

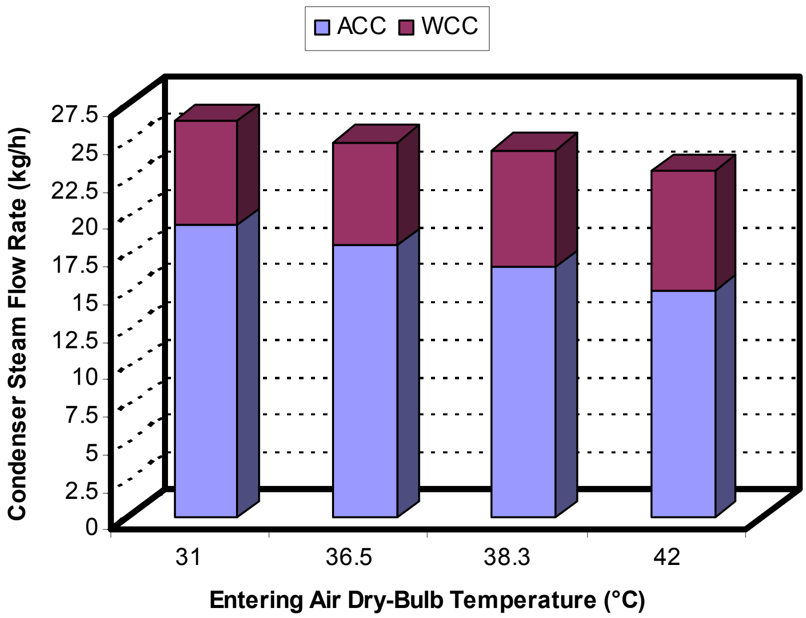

Figure (12). *Steam Load Contribution of the ACC and WCC in the Hybrid System Arrangement at Water Flow Rate of (400) l/hr.*

4. Conclusions

The following findings may be withdrawn from the present work:

1. The air cooled condenser can be implemented with a great confidence to stand alone for steam condensation purpose in winter or cold environment. If it was designed properly then will have the capability to possess the total unit load.
2. Increasing of the air flow rate from (1200) *cfm* to (2400) *cfm* through the heat exchanger improved the overall performance of the condenser. It showed an increase of (3) and (4) kW at entering air temperature of (42) and (21) °C respectively. These values correspond to (23%) and (22%) higher than those at (1200) *cfm*.
3. The hybrid arrangement performs better at the lowest test air temperature (31) °C where the ACC possesses about (72%) of the total unit load. It contributed to more than 65% of the total unit condensation thermal load at the highest test temperature (42) °C.
4. The air cooled condenser has the ability to self-control and acts as a balance mechanism for the possibility of excess steam production of the unit to keep the pressure of the unit constant.
5. The severe operating conditions regarding to the ambient air temperature and water supply shortage was overcome with a great confidence by implementing the hybrid system arrangement.

Nomenclature

A	Area (m^2)
D	Heat Exchanger Depth (mm)
D_t	Tube Section Width (mm)
f_p	Fin Pitch (mm)
h	Fluid Enthalpy (kJ/kg)
H	Heat Exchanger Height (mm)
H_t	Tube Section Height (mm)
L	Length (mm)
$\dot{m}$	Mass Flow Rate (kg/s)
$\dot{Q}$	Heat Load (kW)
t	Thickness (mm)
T	Temperature (°C)
ΔT	Temperature Driving Force (°C)
W	Heat Exchanger Width (mm)
X_L	Longitudinal Distance to Flow Direction (mm)
X_T	Transverse Distance to Flow Direction (mm)

Subscripts

a	Air
ACC	Air Cooled Condenser
ACS	Air Cooled Condenser Alone
f	Fin
Face	Face or Frontal value
g	Gas State
i	Inlet
l	Liquid State
t	Tube
total	Total value
w	Water
WCC	Water Cooled Condenser

References

[1] Matthew S. Layton and Joseph O.Hagan, "Comparison of Alternate Cooling Technologies for California Power Plants", Electric power research institute, Report (2002).

[2] Larinoff M.W. and Moles W.E., "Design and Specification of Air Cooled Steam Condensers", Chemical Engineering, Vol.110, pp 143-156, (1978).

[3] Saiz Jabarrdo, J.M., and Mamani ,W.G., "Modeling and Experimental Evaluation of Parallel Flow Micro Channel Condensers", J.Braz., Soc., Mech., Sci and Eng., Vol.25, No.2, Reo de Janeiro, Apr./June, 2003.

[4] Tarrad, A.H., Khudor D.S and Abdul Wahed M.," A Simplified Model for the Prediction of The Thermal Performance for Cross Flow Air Cooled Heat Exchangers with A new Air Side Thermal Correlation", Engineering and Development Journal, Vol.12, No.3, September 2008.

[5] Tarrad, A.H., Saleh, F.A., and Abdulrasool, A.A.," A Simplified Numerical Model for a Flat Continuous Triangle Fins Air Cooled Heat Exchanger Using a Step by Step Technique", Journal of Engineering and Development, Baghdad, Iraq, (2009).

[6] Tarrad, A.H., "A Numerical Model for Performance Prediction of Dry Cooling Conditions of Air Cooled Condensers in Thermal Power Plant Stations", Journal of Engineering and Technology, Vol.28, No.16, Baghdad, Iraq, (2010).

[7] Tarrad, A.H. and Kemal H.M., "A Model for Prediction of Surface Condenser Performance in Thermal Power Plants", Journal of Engineering and Development, vol.8, No.3, December, 2004.

[8] Tarrad, A.H. and Majeed L.M., "The Application of Step by Step Technique for the Performance Prediction of Thermal Power Plant Surface Condensers ", Journal of Engineering, Vol.16, No.1, March, 2010.

[9] Tarrad, A.H., "A Numerical Model for Thermal-Hydraulic Design of a Shell and Single Pass Low Finned Tube Bundle Heat Exchanger", Journal of Engineering and Technology, Vol.25, No.4, 2007.

[10] Heyns, J.A., and Kroger D.G.," Performance Characteristics of an Air Cooled Steam Condenser Incorporating a hybrid (dry/wet) Dephlegmator", Mechanical Engineering Department, Stellenbosch University, December 2008.

[11] National Renewable Energy Laboratory (NREL)," Hybrid Cooling System for Low-Temperature Geothermal Power Production", NREL Report, No.ARGT.0910, 2011.

[12] Baweja, M. and Bartaria, V. N.," A Review on Performance Analysis of Air-Cooled Condenser under Various Atmospheric Conditions", International Journal of Modern Engineering Research (IJMER), Vol.3, Issue.1, pp-411-414, Jan-Feb. 2013

[13] Tarrad, A. H. and Khudor, D. S., "A Correlation for the Air Side Heat Transfer Coefficient Assessment in Continuous Flat Plate Finned Heat Exchangers", Transactions of the ASME, Journal of Thermal Science and Application, Vol. (7), No. 2, Paper No. TSEA-14-1194, DOI: 10.11151/1.4029459 2015.

[14] Tarrad, A. H. and Al-Nadawi, A. K., "A Rating Model of Finned-Tube Condenser using Pure and Zeotropic Blend Refrigerants", accepted for publication in German Journal of Mechanical Engineering Research, August 2015.

[15] Thermal Conductivity of Metals, in SI unit, available from the World Wide Web (www) site: www.engineeringtoolbox.com.

[16] Altameemi, A. F., "Study and Evaluation of the Operation Characteristics for the Condensation Load Distribution in Hybrid Systems on the Condenser Side", MSc. Thesis, Al-Mustansiriya University, Baghdad, June 2011.

[17] Kemal, H.M., "Performance Prediction of Power Plant Condensers Using Step By Step Method", M.Sc. Thesis, Mech. Eng. Dept., Al-Mustansiriya University, Baghdad, Iraq, June,(2001).

[18] Woodruff, E.B., Lammers, H.B and Lammers, T.F.," Steam Plant Operation", 8^{th} Edition, 2005.

[19] Karl R. Wilber, Kent Zammit, "Development of Procurement Guidelines for Air-Cooled Condensers", CEC/EPRI, Advanced Cooling Strategies/Technologies Conference, Sacramento, California, (2005).