The dry and adiabatic fluid cooler as an alternative to cooling towers: An experimental view

M. Lucas, P.J. Martínez, J. Ruiz

Dpto. Ingeniería de Sistemas Industriales. Universidad Miguel Hernández de Elche
mlucas@umh.es

A.S. Káiser, A. Viedma, B. Zamora

Departamento de Ingeniería Térmica y de Fluidos. Universidad Politécnica de Cartagena

Abstract

Energy and environmental implications of a refrigeration cycle are largely conditioned by the choice of condensing system. Conventional solutions transfer heat to water, and recirculated through cooling towers or to atmospheric air through a dry condenser. While the use of cooling towers means less energy consumption due to lower pressure in the condenser, a number of environmental implications are questioning their installation. Mainly, it represents an emission of chemicals or microorganisms to the atmosphere as Legionella. The dry and adiabatic fluid cooler works as a standard fluid dry cooler enhancing the dry cooler's capacity with adiabatic pre-cooling of the air intake. The ambient dry bulb temperature is reduced as the air passes through an evaporative pad especially designed to humidify and cool the air (Figure.1). The main objective of this study is to experimentally investigate the thermal performance of a dry and adiabatic fluid cooler. With the experimental data, a thermal model will be developed in order to compare the energy implications arising from the replacement of traditional systems (dry condenser and cooling towers) by condensing air coolers with adiabatic pre-cooling.

INTRODUCTION

The use of air conditioning systems in buildings is gaining importance caused by the quality of life and levels of comfort in society today. Directive 2002/91/EC of the European Parliament and of the Council of 16 December on the energy efficiency performance of buildings includes the housing sector and services, consisting mainly of buildings, absorbs more 40% of final energy consumption in the Community and is being expanded. This trend will likely increase energy consumption and thus emissions of carbon dioxide. In view of this directive, we understand the importance the EU attaches to energy savings in the tertiary sector. The air conditioning contributes to increased energy demand in the tertiary sector.

The increase in installed capacity for air conditioning has caused in countries like Spain increased peak electricity consumption in the summer period reaching a value similar to the consumption peak in winter [1]. The current trend to replace the condensed water systems for air-cooled systems in centralized facilities has increased this fact. As a guide, it may be noted that the refrigeration equipment that can be condensed by water covering a power range between 120 and 5000 kW, and that within that range are competing with the possibility of being air cooled up to 1500 kW [2].

A fundamental difference between the water-cooled and air condensation is the condensation of water use lower levels of temperature in the refrigeration system, so thereby maintaining the other operating conditions, energy consumption and cost equipment performance is lower

. The effect of temperature variation of condensation on the power absorbed by the compressor can be from 1.8 to 4% per degree Celsius [3], depending on the cycle under consideration and the refrigerant used. Associated with the worst energy efficiency of air-condensed systems is the increase in emissions of CO2 to the atmosphere.

Cooling towers are equipment devices commonly used to dissipate heat from water-cooled refrigeration, air conditioning and industrial processes. The principle of operation of cooling towers requires distributing or spraying water over a heat transfer surface across or through which a stream of air is passing [4], [5]. As a result, water droplets are incorporated in the air stream and, depending on the velocity of the air, will be carried out of the unit. This is known as drift and it is independent of water lost by evaporation.

Cooling tower drift is objectionable for several reasons [6]. Mainly, it represents an emission of chemicals or microorganisms to the atmosphere. In addition, corrosion problems can result on equipment, piping and structural steel, and can be the source of electrical systems' failure. In the case of cooling towers, undoubtedly the most well known pathogens are the multiple species of bacteria collectively known as Legionella. These bacteria tend to thrive at the range of water temperatures frequently found in these cooling systems. Hence, workers or other persons near a cooling tower may be exposed to drift, may inhale aerosols containing the Legionella bacteria, and may become infected with the illness. Several Legionella outbreaks have been linked to cooling towers [7], [8].

In Spain, some local Governments tend to restrict the installation of cooling towers due to several severe outbreaks of Legionella [9], [10]. For example, the local government of the city of Murcia has forbidden the installation of cooling tower in the metropolitan area [11]. The regional government of Valencia has published public financial aids [12]to replace cooling towers with a more safety alternative, standard dry coolers. Following this tendency, some companies, owners of commercial buildings with large air conditioning systems (thousands of kilowatts), have replaced cooling towers with dry coolers. This situation is not only about human health but also about energy efficiency and sustainability.

The answer that the market has offered to administrative pressure exerted on the cooling towers is the search for alternatives to dissipate heat from industrial plants, refrigeration and air conditioning. The commercial alternatives to replace conventional cooling towers are air cooled heat exchangers. If from a public health approach air heat exchangers have an advantage because they are not risk facilities, from an energy point of view creates more consumption and increased cost of operation. Associated with the worst energy efficiency of air-condensed systems is the increase in emissions of CO2 to the atmosphere.

In addition to traditional solutions such as condensation from cooling towers, evaporative condensers and air condensers, commercially hybrid devices are emerging that seek a compromise between environmental impact and energy consumption. The objective of the dry and adiabatic fluid coolers is to cool water but with a different operation principle. The dry and adiabatic fluid cooler works as a standard fluid dry cooler enhancing the dry cooler's capacity with adiabatic pre-cooling of the air intake. The ambient dry bulb temperature is reduced as the air passes through an evaporative pad especially designed to humidify and cool the air (Fig.1). Part of the water is evaporated, while the excess water leaves the adiabatic section via a gutter system to the sewer.

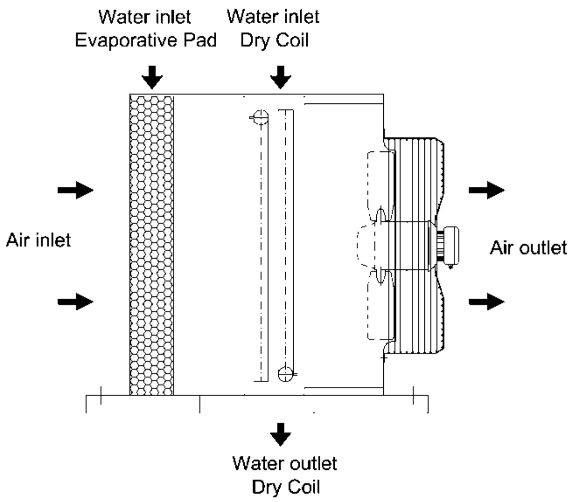

Fig. 1. Schematic diagram of the dry and adiabatic fluid cooler

The catalog data supplied by the manufacturers of aero dry and adiabatic fluid cooler usually coincide with those standard fluid cooler. This does not reflect the actual physical operation of these devices and prevents evaluating energy improvements are achieved by installing a dry and adiabatic fluid cooler instead of a standard fluid cooler. The objective of this work is to evaluate experimentally the thermal performance of a dry and adiabatic fluid cooler in order to verify the improvements they offer compared to the standard fluid cooler. In addition, to determine the influence that the main system operating variables cause both on the efficiency of the evaporative section, and on the global coefficient of heat exchange of the dry section.

METHOD

Analysis of the heat and mass transfer process in a direct evaporative cooler.

Spray devices using directly recirculated water have been in use for many years with different objectives. They have been used for conditioning air being called terms such as evaporative coolers, adiabatic plants or humidifiers. Fig. (2) shows a schematic diagram that will be use for the analysis. The major assumptions that are used to derive the basic modeling equations may be summarized as, Kuehn et al. [12]:

- The rate at which make-up water is added to the sump is negligibly small compared to the rate of water flow through the nozzles
- Heat transfer through the walls of the chamber from the ambient surroundings is ignored
- The small addition of energy to the water by the pump has a negligibly effect upon the water temperature
- There is no temperature change of the water droplets as they pass through the device
- The mass flow rate of water is much greater than the mass rate of air

From steady-state energy and mass balances on an incremental volume (Fig. 2), we get

$$\dot{m}_a \cdot dh = \dot{m}_a \cdot dW \cdot h_{f,w} \qquad (1)$$

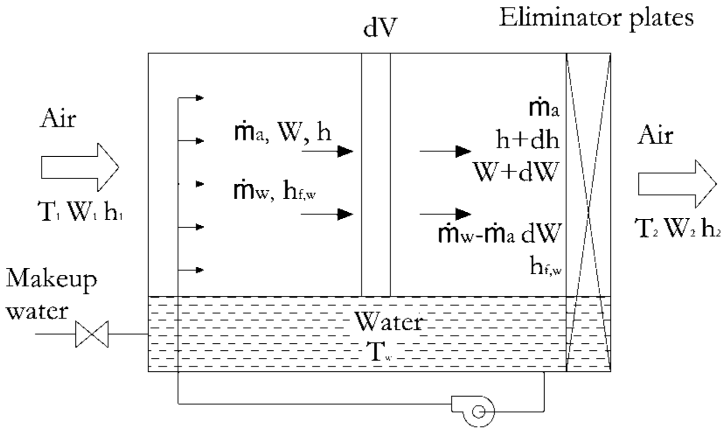

Fig. 2. Schematic diagram of evaporative cooler using directly recirculated water

We may also write the water energy balance in terms of the heat- and mass-transfer coefficients, h_C and h_D; respectively. Heat transfer for evaporation of water added must come from cooling of the air stream. Thus,

$$\underbrace{h_C \cdot A_V \cdot dV \cdot (T - T_w)}_{\text{Sensible Heat}} = \underbrace{h_D \cdot A_V \cdot dV \cdot (W_{s,w} - W) \cdot h_{fg,w}}_{\text{Latent Heat}} \qquad (2)$$

where h_C is transfer coefficient of air, (kW/m^2 K); A_V is surface area of water droplets per unit volume of tower, (m^2/m^3); V is contact volume (m^3); T is dry bulb temperature of moist air (K) and T_w is water temperature, (ºC) (K); h_D is convective mass transfer coefficient, (kg$_a$/m^2 s); W is humidity ratio of moist air, (kg$_w$/kg$_a$); $W_{s,w}$ is humidity ratio of saturated moist air evaluated at t_w, (kg$_w$/kg$_a$) and h_{fg}, change of phase enthalpy($h_{fg,w=}$ $h_{g,w-}$ $h_{f,w)}$, (kJ/kg$_w$).

Substitution of Lewis $Le = h_C/h_D cp_a$ in Eq (2) and assuming that Le must be unity for a device using directly recirculated sprayed water gives:

$$\eta_w = \frac{(T_2 - T_1)}{(T_1 - T_{bh})} \qquad (2)$$

Analysis of the dry heat exchanger.

To model the behavior of a heat exchanger is essential to relate the total heat transfer to the inlet and outlet temperatures of the fluid, the global heat transfer coefficient and the total surface area for heat transfer. Modeling the dry section is performed using the widely known model of heat exchangers of Log mean temperature difference (LMTD). Two starting equations can be obtained by applying global energy balances to the hot and cold fluids.

$$q = \dot{m}_a \cdot dh \tag{3}$$

$$q = \dot{m}_w \cdot c_{p,w} \cdot dT_w \tag{4}$$

It can be obtained another expression relating the total heat transfer q to the temperature difference between hot and cold fluids, the extent of Newton's law of cooling, using the global coefficient of heat transfer U. However, as ΔT varies with position in the heat exchanger, is necessary to work with a flow equation of the form:

$$q = U \cdot A \cdot \Delta T_m \tag{5}$$

Where ΔT_m is an appropriate mean temperature difference. This can be determined by applying an energy balance for different elements in the hot and cold fluids. The general assumptions for the problem solution are as follows:

- Heat exchanger is insulated from its surroundings, in which case the only heat exchange is between hot and cold fluids.
- Conduction along the walls is negligible.
- Potential and kinetic energy changes are negligible.
- Overall heat transfer coefficient changes over the heat surfaces are negligible.

Under these assumptions the Log mean temperature difference can be expressed by:

$$\Delta T_m = \frac{\Delta T_{in} - \Delta T_{out}}{\ln(\Delta T_{in} / \Delta T_{outl})} \tag{6}$$

Experimental set-up

Experiments were carried out on a test facility assembled on the roof of a laboratory at the Universidad Miguel Hernández in the city of Elche in the southeast of Spain. Since the dry and adiabatic fluid coolers found in the market are of great powers, it was decided to build a prototype composed of commercial elements. The main modules that make up the pilot plant are: evaporating section, coupling section and dry heat exchanger. The adiabatic panel is the commercial Control and Ventilation HUMIBAT L-20 model with dimensions of 650x2000x1250 mm and weighs 150 kg. Packing is a plastic mesh and nominal pressure at the nozzles is 10 m.c.a. Nominal water mass flow is 1,75 m^3/h. Dry heat exchanger is the commercial BTU EAA66-023011.4/H model. It's a dry heat exchanger with copper pipes and aluminum fins. The dimensions of coupling are practically the same as the evaporative section. It has two fans of 1.9 kW of electrical power consumed to produce a nominal airflow of 24000 m3/h. The fan's motor is equipped with a variable speed control, which allows the change of the air mass flow rate. The data for thermal design of dry heat exchanger without considering adiabatic section are: Thermal Power 30 kW, for an inlet temperature of water 40°C, an outlet temperature of 35ºC and an ambient air temperature of 30°C. The heat load is an electrical heater Stiebel Eltron DHKW45 model, made up of 15 3-kW electric heating elements.

Fig. 3. View of pilot test facility assembled at Universidad Miguel Hernandez, Elche (Spain)

A general-purpose data-acquisition system was set up to carry out the experimental tests. All data were monitored with an HP 34970A Data Acquisition Unit. A specific program was written and compiled for the system in HP BenchLink Data Logger Software, supporting up to 36 inputs, with 16 bits A/D, 9600 bauds transmission speed and programmable gain for individual channels. The physical principia of the sensors used during the experiment and the specifications of the measuring devices are shown in Table 1.

Experimental procedure

This section describes the test conditions carried out to characterize thermodynamically the dry and adiabatic fluid cooler. The aim of the tests is to obtain the evaporative section efficiency and the product of area and the global heat transfer coefficient, UA, of the dry section. The tests are structured by selecting the variables to study and setting two levels for each one: Fan speed (50 Hz and 25 Hz), evaporative section water flow (1.75 m^3/h - 0.88 m^3/h), dry section water flow (5.2 m^3/h - 2.6 m^3/h) and Thermal Load (30 kW and 15 kW). This makes a total of 2^4 tests that were performed in random order.

At the moment of conducting the test, the pilot plant should be within acceptable operating conditions. That is, the water distribution system should be clean and free of foreign materials that may obstruct the flow. Mechanical equipment must be in good working order. Fans should turn in the right direction. The filling of the evaporative section must be free of dirt, scale, residual oil, tar or straw. The water level in the evaporative section collection tank should remain constant during the test to ensure a proper flow.

In order to achieve steady operating conditions is considered a start-up period of 30 minutes. From that moment starts the data collection for a period of 2 hr. After processing the data, it checks whether it has achieved a time interval greater than 30 minutes, where the variables are within the limits defined by the stationary conditions on the Standard UNE 13741 [14].

Table1. Sensor devices specifications

Parameter	Model	Sensor	Range	Accuracy	Output
Air temperature	E+E (EE20-FT6B51)	Capacitive Sensor	-20ºC a 80ºC	±0,3ºC	4 - 20 mA
Air Humidity	E+E (EE20-FT6B51)	Capacitive Sensor	0% a 100%	±2%	4 - 20 mA
Wind velocity	Young (05103L)	Cup anemometer	0 a 50 m/s	±0,3 m/s	0 - 10 V
Air Velocity	Testo	Vane anemometer	0,5 a 20 m/s	0,1m/s ±1,5% m.v.	4 - 20 mA
Evaporative Water flow rate	Krohne Optiflux 2000	Electromagnetic	DN 25-150	±0,3% m.v.	4 - 20 mA
Heat Exchanger Water flow rate	Contacesa	Oval wheel flow meter	2 – 20 m3/h	0,4 % f.s.	4-20 mA
Water temperature	Desin	Pt 100 type RTD	-200 a 600oC	±0,08ºC	4 wires

RESULTS

This section first describes the results of a test in detail and then shows the summary results for all the 16 trials. After starting up the installation and configuration of sensors, a test setting as operating conditions the nominal conditions of the equipment was carried out (Test 15 in Table 2). That is, a water flow in the recirculation circuit of the evaporative section of 1.75 m^3/hr, a water flow in the dry section of 5.2 m^3/hr, a thermal load of 30 kW and fans fed by turning the drive frequency to 50 Hz were set. The initial values of water flows and thermal load will progress throughout the test as a result of the dynamics of the installation with which the measured values will approach but not match them exactly. The air velocity sensor was installed in the middle section for being the location where a more stable measure is obtained. The pilot facility is operated for 2 hr and data collected from the selected time period to comply with the stationary conditions. In this test, after the first half hour following 30 min comply with these conditions.

Fig (4) shows the main variables collected during the test in the evaporative section. It can be seen the reduction of dry air temperature input from a mean value of 30.87°C to about 25.36ºC, which means that evaporative section reduces 5.51°C the dry temperature of the air entering the dry exchanger. In the period of measurement, the wet bulb temperature of inlet air is 23.96 º C and this causes an efficiency of evaporative section η = 79.7%, with an uncertainty of 8.2% calculated according to the uncertainty ISO guide [15], with a confidence level of 95%. An interesting practical data to assess the environmental implications of this device is the amount of evaporated water. For its determination raises a mass balance in the evaporative section in which the amount of water evaporated is defined by the product of air mass flow and absolute humidity difference between the middle section and the input, calculating a value of 6,42 10-3 kg/s. The air mass flow is determined by a balance of energy in the dry section and the absolute humidity is derived from the direct reading of dry bulb temperature and relative humidity at the input and in the middle section.

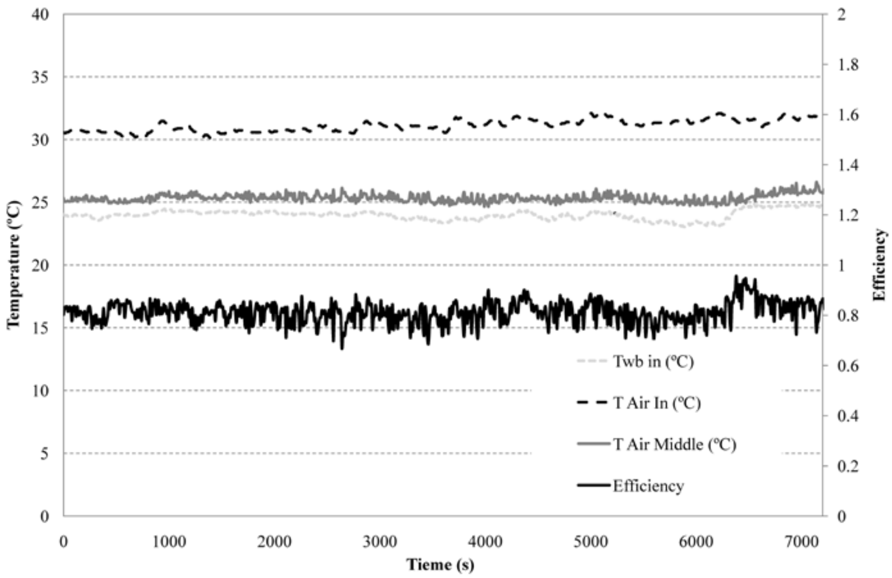

Fig. 4 Variables measured at the evaporative section

The temperatures of the heat exchanger section are: Inlet water temperature, 39.72ºC is cooled to 34.48ºC. Whereas the circulating water flow at the heat exchanger is 5.09 m3/hr leads to a exchanged thermal load of 30.988 kW. Air side if it comes to 25.36ºC recorded in the middle section, comes out to 35.31°C. Air enters at 25.36ºC recorded in the middle section, going out at 35.31ºC. With these data a Log mean temperature difference Δ Tm = 6.485ºC is obtained and the product of area and the global heat transfer coefficient is UA = 4750 W/°C.

A representation on a psychrometric chart show how the transformation of the inlet air evaporative section remains roughly constant enthalpy line, which is indicative of an adiabatic process. In the dry section the transformation undergone by the heating air is a constant absolute humidity as there is no water supply or removed.

Once the test results for the nominal operating conditions are described, Table 2 shows the main results obtained for the data set of 16 tests.

To know the influence that any of the variables have on efficiency and UA, 2-2 tests are represented where only the displayed variable changes. That is, for example in Fig (5) between Test 1 and 14 only changes the fan velocity while the water flows and the thermal load is the same and so on. Fig (5) shows how decreases the efficiency of evaporative section with increasing velocity of air flow. This effect is already described in some references [16] and is justified by the fact that increasing the air flow velocity decreases the contact time with the film of water filling the evaporative section, which results in a lower wetting himself and, and consequently a lower efficiency.

Table 2. Medium values measured during the tests

Test	Water flow dry section (m^3/h)	Inlet Water T (ºC)	Outlet Water T (ºC))	Water flow Evap. section (m^3/h)	Air Vel (m/s)	Air Inlet Tª (ºC)	Air Wet Bulb T (ºC)	Air Middle T (ºC)	Air Outlet T (ºC)	Efic.	Thermal Load (kW)	LMTD (ºC)	UA (W/ºC)
Test 1	2.45	43.29	33.16	0.97	1.16	28.40	22.85	23.69	34.69	0.85	28.86	9.03	3.20
Test 2	2.81	36.39	31.45	1.76	0.92	31.40	24.54	25.57	32.65	0.85	16.11	4.73	3.41
Test 3	5.19	29.42	26.82	1.69	1.64	25.23	21.41	22.06	28.42	0.83	15.63	2.41	6.49
Test 4	5.32	32.94	30.35	0.98	1.22	30.23	23.09	23.85	31.66	0.89	15.99	3.21	4.98
Test 5	2.69	43.99	34.58	1.74	0.92	30.35	24.22	24.89	35.27	0.89	29.39	9.19	3.20
Test 6	5.17	33.63	30.97	0.89	1.52	31.12	24.83	25.71	32.38	0.86	15.99	2.79	5.73
Test 7	2.55	34.56	29.36	0.88	1.22	32.36	23.71	24.05	31.83	0.96	15.41	3.88	3.97
Test 8	5.23	39.02	34.05	0.87	1.12	27.76	21.91	22.41	32.46	0.92	30.14	8.86	3.40
Test 9	5.22	31.12	28.52	1.71	0.93	27.29	21.71	22.22	28.35	0.91	15.71	4.30	3.65
Test 10	2.38	30.55	24.65	1.62	1.56	31.39	18.19	20.33	28.08	0.84	16.35	3.30	4.95
Test 11	5.50	35.85	31.18	1.71	1.04	28.38	17.46	19.62	31.65	0.80	29.82	7.26	4.10
Test 12	2.59	30.21	25.04	1.05	1.68	25.37	20.45	21.38	26.32	0.81	15.59	3.77	4.13
Test 13	2.91	38.94	30.04	1.70	1.45	31.12	20.08	21.81	32.23	0.84	30.05	7.45	4.04
Test 14	2.74	42.14	32.75	0.92	1.58	29.82	23.90	25.11	34.85	0.79	29.84	7.46	4.00
Test 15	5.09	39.72	34.48	1.68	1.60	30.87	23.96	25.36	35.31	0.80	30.99	6.48	4.75
Test 16	5.24	39.84	34.92	0.93	1.68	33.03	24.95	25.72	35.71	0.91	29.93	6.33	4.73

Fig (6) shows the influence of evaporative section water flow on efficiency. In view of the results, we cannot get a clear conclusion. This does not correspond with what was expected since water to air mass flow ratio appears to be a determining factor in the evaporative cooling equipment. The absence of a clear trend can be justified because the range of work studied can moisturize enough the filling of the evaporative section, without changing thus the area of exchange between both streams.

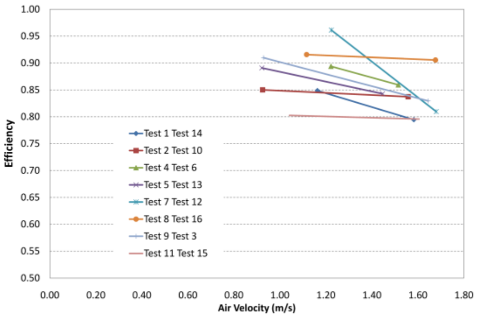

Fig. 5. Influence of air velocity on efficiency

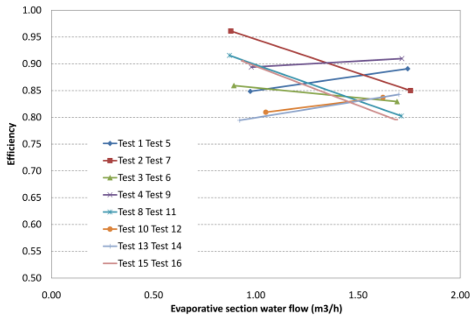

Fig. 6. Influence of evaporative section water flow on efficiency

The influence that the studied variables have in the product of area and the global heat transfer coefficient from the dry heat exchanger is discussed below. Figs (7) and Fig (8) show how the heat transfer increases by increasing the velocity of air through the outside of the pipes and water inside. This is justified by the increase in both external convection coefficient and internal respectively. Concerning the influence of thermal load, in Fig (9) can be seen that when it increases, the heat transfer per unit temperature decreases. This shows that with increasing power, although the overall coefficient of heat transfer increases, it is not so much how does the LMTD. As for the influence of water flow in the evaporative section, Fig (10) shows no clear trend, something that it was expected after analyzing the results of the adiabatic section.

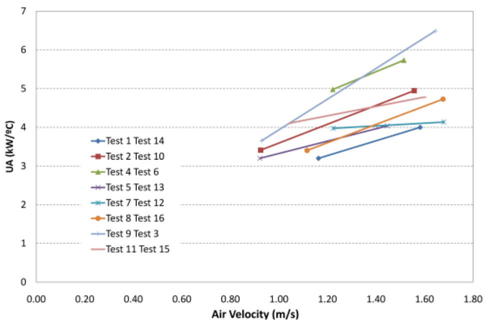

Fig. 7. Influence of air velocity on UA

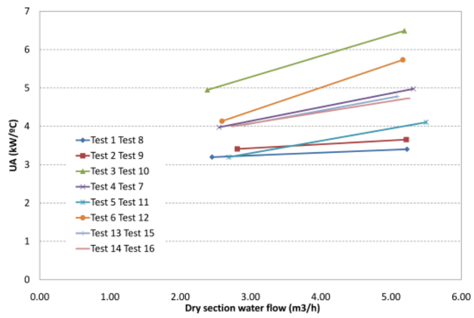

Fig. 8. Influence of dry section water flow on UA

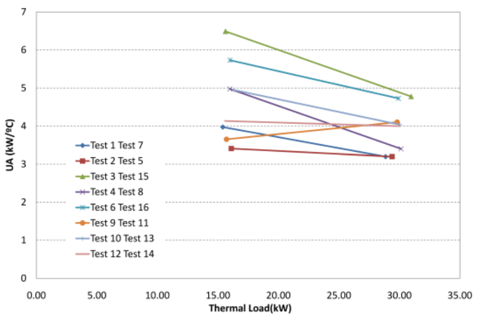

Fig. 9. Influence of thermal load on UA

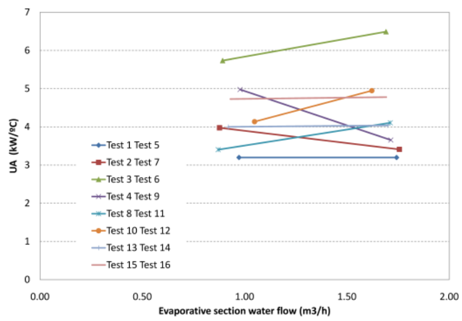

Fig. 10. Influence of evaporative section water flow on UA

CONCLUSIONS AND FUTURE WORK

This paper has described the design and construction of a dry and adiabatic fluid cooler prototype. The final objective of building the prototype is to have an installation that will generate an experimental model with which to describe the thermal behavior of the dry and adiabatic fluid cooler. As a working hypothesis, the prototype was expected to obtain more favorable conditions of operation that can be expected of dry fluid cooler.

The study of the evaporative section has shown that the air temperature decreases to pass through the filling. In nominal conditions, it has been obtained that the temperature of inlet air to the dry section after passing through the adiabatic section is 5.51ºC below ambient. For these operating conditions, a cooling efficiency of 79.7% it has been obtained. At the dry section, a thermal load of 30.988 kW is exchanged and the product of area and the global heat transfer coefficient is UA = 4750 W/°C.

The parametric study has shown how decreases the efficiency of evaporative section with increasing velocity of air flow. On the other side, to modify the flow in the evaporative section does not show a clear trend in efficiency over the range studied. With respect to the product of area and the global heat transfer coefficient air cooler is shown how the product of area and the global heat transfer coefficient increases by increasing the velocity of air through the outside of the pipes and water inside. Concerning the influence of thermal load, the heat transfer per unit temperature decreases.

With all this information it is intended to build a model to assess the influence of the installation of a dry and adiabatic fluid cooler in place of a conventional fluid cooler, paying special attention to energy consumption of air conditioning facilities in different locations.

ACKNOWLEDGEMENTS

Construction of prototype has been sponsored by the Regional Government of the Comunidad Valenciana, through the Project GV/2011/072.

The instrumentation used to conduct experimental work has been funded through the Project "Estudio energético de un sistema alternativo para la condensación en ciclos de refrigeración: el aero-refrigerador con preenfriamiento adiabático" from *Convocatoria 2009 de Proyectos Bancaja-UMH para grupos precompetitivos*.

REFERENCES

[1] www.ree.es/cap03/pdf/MIBEL/Seguimiento_demanda_MIBEL_SEP2006.pdf
[2] M.D. Pugh (CTI), Benefits of Water-Cooled Systems vs Air-Cooled Systems for Air-Conditioning Applications. AHR/ASHRAE/ARI Expo, 2005.
[3] F.W.H. Yik, J. Burnett, I. Prescott, Predicting air-conditioning energy consumption of a group of buildings using different heat rejection methods. Energy and Buildings 33 (2) (2001) 151-166.
[4] J.C. Kloppers, D.G. Kröger Cooling Tower Performance Evaluation: Merkel, Poppe, and e-NTU Methods of Analysis. Journal of Engineering for Gas Turbines and Power, 127 (1) (2005) 1–7.
[5] J.C. Kloppers, D.G. Kröger A critical investigation into the heat and mass transfer analysis of counterflow wet-cooling towers. Int. J. of Heat & Mass Transfer, 48 (3-4) (2005) 765-777.
[6] B.G. Lewis, On the question of airborne transmission of pathogenic organisms in cooling tower drift. Cooling Tower Institute, Technical Paper-T-124A, 1974.
[7] R. Isozumi, Y. Ito, I. Ito, M. Osawa, T. Hirai, S. Takakura, Y. Iinuma, S. Ichiyama, K. Tateda, K. Yamaguchi, M. Mishima, An outbreak of Legionella pneumonia originating from a cooling tower. Scand J Infect Dis. 37(10) (2005) 709-11.
[8] R.H. Bentham and C.R. Broadbent, A model autumn outbreaks of Legionnaries' disease associate with cooling towers, linked to system operation and size. Epidemiol Infect. 111 (1993) 287-95.
[9] A. García-Fulgeiras, C. Navarro, D. Fenoll, J. García, P. Gonzalez-Diego, Legionnaires' Disease Outbreak in Murcia, Spain. Emerging Infectious Diseases. 9 (8) (2003) 915-921.
[10] J. A. Fernández, P. López, D. Orozco, J. Merino, Clinical Study of an Outbreak of Legionnaire's Disease in Alcoy, Southeastern Spain. Eur J Clin Microbiol Infect Dis (2002) 21:729–735
[11] Ordenanza de protección de la atmósfera. Disposición Transitoria única. Ayto de Murcia BORM. 20/6/2006
[12] Orden de la Consellería de Empresa, Universidad y Ciencia, sobre concesión de ayudas para instalaciones industriales de riesgo para la transmisión de la legionelosis.
[13] Kuehn T., Threlkeld J.L. y Ramsey J.W. (1998), Thermal environmental engineering. Prentice Hall.
[14] UNE-EN 13741 "Ensayos de recepción de las prestaciones térmicas de las torres de refrigeración húmedas de tiro mecánico fabricadas en serie (2004)
[15] ISO Guide "Guide to the expression of uncertainty in measurement" (1993), ISBN: 92-67-10188-9.
[16] J.M. Wu, X. Huang, H. Zhang Theoretical analysis on heat and mass transfer in a direct evaporative cooler Applied Thermal Engineering 29 (2009) 980–984