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Effect of pre-cooling of inlet air to condensers of air-conditioning units

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SUMMARY

Energy conservation and increase in performance of air-conditioning systems could be achieved by pre-cooling the air intake of the condensers. This paper experiments three different methods of pre-cooling the condenser air; the cooling pad (CP) setup, the cooling mesh (CM) setup and the shading setup. The CP and CM setups are two different methods of evaporatively cooling the air. The three methods have been applied to three identical, 2.8 tons, split air-conditioning units during the peak summer time period in Kuwait, under ambient temperatures ranging from 39 to 45°C. The results yielded a drop in the power consumption ranging from 8.1 to 20.5% and an increase in the cooling load ranging from 6.4 to 7.8% by using the CP and CM setups, which, in turn, resulted in an increase in the coefficient of performance (COP) of the units by 36–59%. The shading setup has resulted in an increase of power consumption due to air trapped below the shaded area, which resulted in heat being generated. Copyright © 2005 John Wiley & Sons, Ltd.

KEY WORDS: air-conditioning; condenser; evaporative cooling; energy conservation; power consumption

1. INTRODUCTION

Due to extreme summer temperatures in Kuwait, 75% of the total power generated is consumed during summertime by air-conditioning. As the outdoor temperature increases, the performance of the condensers decreases, thereby increasing the power required for cooling. In addition to more cooling load, another drawback is the decrease in the compressor life expectancy.

A possible solution to meeting the extremely high demand of electrical power consumption during summertime is to pre-cool the air entering the condenser, resulting in higher performance of air-conditioning units, and hence, lower the power required for cooling.

Improving air-conditioning performance has always been an issue in this part of the world, where the outside temperature may reach 55°C during the summer in the shade. The performance of the unit depends directly on the ambient temperature and is known to decrease with the increase in ambient temperature. More than 20% decrease in A/C performance can be noticed when the ambient temperature increases from 35 to 50°C.

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The objective of this research is to study and experiment three different methods of pre-cooling the air intake of condensers of split air-conditioning units. Two methods involve applying evaporative cooling to the air intake of the condensers, and the third method involves placing the outdoor unit of the air conditioner under shade. Almost all of Kuwait's outdoor units are placed on top of the roofs of buildings without shade. Under shaded conditions, the temperature of the air intake is expected to be lower than that of air under direct sunlight and radiation heat gain will be minimal.

Evaporative cooling could be applied through many different configurations. Two specific configurations are expected to be effective in both heat transfer enhancement and in reducing scaling problems associated with the contact of water with the condenser surface, and therefore, are recommended to be tested and evaluated.

The first configuration, called the cooling pad (CP) system, involves spraying water on top of cellulose CPs located in front of the condenser, thereby, cooling the air prior to passing over the condenser coils. The second configuration, called the cooling mesh (CM) system, introduces the idea of spraying water onto a thin plastic mesh opposite to the direction of air intake, thereby cooling the entering air and at the same time avoiding direct contact of water with the condenser coils and hence, avoiding scaling problems.

The main objectives of this research are to study and compare the performance characteristics of the air conditioner under normal operating conditions and under pre-cooled condenser air conditions, to calculate the electrical power savings and finally, to justify the cost of water consumed in the evaporative cooling methods against the lowered power consumption of the air-conditioning unit.

Guinn and Novell (1981) experimented evaporative cooling on a three-ton split air-conditioning unit, in Alabama, by directly spraying water onto the condenser. The results yielded a 12–19% increase in the energy efficiency ratio (EER), a reduction in electrical consumption by 5–9%, and a reduction in the compressor head pressure by 9–17%.

Guinn *et al.* (1985) conducted a similar experiment through two summers in Alabama on a three-ton packaged heat pump. The reduction in electrical consumption was 8.7%.

In both researches, problems with mineral deposits and scaling on the condenser coils were faced. In an attempt to avoid such problems, a different experimental study by Goswami *et al.* (1993) involved applying indirect evaporative cooling to the condenser of a 2.5-ton residential air-conditioning unit by using a media pad evaporative cooler. The results yielded a reduction in compressor power by 17–20% and an increase in the EER by 22%.

Further experimental analysis has been conducted by Waterbury *et al.* (1999) on evaporative cooling using a media pad evaporative cooler. The control of water flow was a function of the ambient dry bulb temperature. The experiments were performed on a 7.5-ton DX unit. With a yearly water consumption rate of 99 000 gal yr^{-1}, the yearly energy consumption dropped down from 14 334 to 2930 kW h yr^{-1}.

A simple theoretical comparison between air cooled and evaporatively cooled condensers has been presented by James *et al.* (1993) in terms of design conditions, energy consumption, operational cost, ease of maintenance, space requirements, fan noise and health risks of Legionnaire's disease.

Kachhwaha *et al.* (1998) predicted the evaporative cooling performance in spray air-flow systems using a two-dimensional numerical model that included drop size distribution and velocities at nozzle spray angle.

Al-Taqi and Maheshwari (2003) conducted evaporative cooling experiments on a package unit and two air cooled chillers in Kuwait, using a condenser pre-cooling system (CPS) similar to the CM system proposed above. Under average outdoor conditions of 43.8°C, 15% RH (in the city) and 28.6% RH (coastal area), they managed to achieve an average effective power saving of 1.42 kW for the packaged unit (in the city), 4.6 kW for the air-cooled chiller (in the city) and 7.4 kW for the air-cooled chiller (coastal area).

This research aims to study more than one method of pre-cooling of air intake to condensers of split air-conditioning units during the months July–September. During this period, in which ambient temperatures reach 60°C under direct sunlight, Kuwait power stations operate at peak electrical demand conditions. As the electrical demand increases, more power stations will be required. In an effort to reduce peak demand as well as to eliminate the need of more power stations, methods of energy conservation need to be researched and applied. No similar research has been found that conducts such a study in Kuwait on three different condenser intake air pre-cooling methods. Such a study in this part of the world, which is considered to be one of the hottest spots on Earth, is vital to cut down power plant costs, electrical bills charged to the consumers and the overall application of energy conservation principles.

2. THEORETICAL BACKGROUND

The cooling load of a split air-conditioning unit is calculated using measured parameters of the air side of the evaporator. If the dry and wet bulb temperatures of the air at the inlet and exit points of the evaporator are measured, then the cooling load is calculated as

$$Q = \dot{m}_{\text{air}}(h_{\text{air,in}} - h_{\text{air,exit}}) \tag{1}$$

where the enthalpies of air are obtained at the measured dry and wet bulb temperatures.

If the air flow from the evaporator is passed through a calibrated nozzle, and the pressure differential is measured across that nozzle, then the air mass flow rate is calculated as

$$\dot{m}_{\text{air}} = C\sqrt{\Delta P} \tag{2}$$

where C is a nozzle constant that depends on the nozzle calibration.

The cooling load in BTU is

$$Q_{\text{BTU}} = 3412.14Q \tag{3}$$

The coefficient of performance is calculated as

$$\text{COP} = \frac{Q}{\text{Power}_{\text{total}}} \tag{4}$$

where the total power consumption of the air-conditioning unit is measured.

3. EXPERIMENTAL WORK

The experimental work is carried out in the heat transfer laboratory of Kuwait University. The heat transfer lab is equipped with a psychrometric facility used to test air conditioners.

Table I. Specifications of the air-conditioning units.

Specifications	
Make	York
Description	Split unit
Indoor unit	Suspended ceiling type
Outdoor unit	4-sided condenser with top discharge
Cooling capacity at 46°C outdoor temperature	33300 Btu h^{-1} (2.8 tons)
Refrigerant	R-22
Compressor	Reciprocating
Expansion device	Restrictor
Power consumption	3 kW
Indoor airflow	1200 CFM

Three identical air-conditioning split units are used in the experimental study. The specifications of the units are shown in Table I. Each unit is used to study one of the proposed cooling systems (CM, CP and shade). The units are all brand new and are of the same make, model and cooling capacity. All three conditioning evaporators are placed in a psychrometric room. The outdoor units are placed right outside the lab to investigate the direct impact of outdoor summer conditions on the performance of the units. Two of the outdoor units are placed under direct sunlight to test the two different methods of evaporative cooling under simulated rooftop conditions (in Kuwait, outdoor units are usually placed on the rooftop of buildings without shades). A one-pass water circulation system is used, which makes the control of water flow a vital issue in this experiment in order to minimize the running costs of the evaporative cooling system and maximize savings. The third outdoor unit is placed under a removable shading device to test the shading effect on the performance of the unit.

Data is collected during the peak time of the summer, July–September, under outdoor conditions ranging from 39 to 45°C.

4. INSTRUMENTATION

The setup consists of three main systems; the indoor control system of the psychrometric facility, the outdoor system, consisting of the CM, CP and shade setups, and the data acquisition system.

The evaporators are placed in a psychrometric room that simulates a typical indoor environment. The outdoor units are placed outside the lab for a simulated typical rooftop environment. Both systems are hooked up to a central data acquisition (DAQ) system that collects the various measurements and stores them in a data file.

The completely insulated thermally controlled psychrometric room accommodates the air-conditioning evaporators. The evaporator under test is connected to an airflow sampling and measurement tunnel via a duct to measure its dry and wet bulb temperatures and air flow rate. Air flow rate is calculated according to Equation (2) by passing the air through a pre-determined calibrated set of nozzles, and measuring the pressure differential across them. Electrical wires are extended from the unit to a separate electrical box and power measurements are obtained with the aid of power transducers. The air inside the room passes through a series of heating and cooling coils and steam ejectors to control its temperature and humidity to the required indoor conditions. An air sampler measures the indoor air conditions. A schematic diagram for the indoor setup is presented in Figure 1. The indoor conditions are maintained at a temperature of 26°C and a relative humidity of 50% through the use of 40 kW, PID-controlled, trim electric duct heaters, duct humidifiers and duct dehumidifiers. This way, the AC unit is kept under continuous operation for the two-hour experimental period.

The three outdoor units are placed right outside the lab, each fitted to one of the proposed cooling setups.

The CM setup basically involves fixing a plastic mesh around each of the four air intake sides of the air-cooled condenser. More mesh material is fixed to cover the corners and any other exposed areas of the condenser. This way, any airflow is forced to pass through the wetted mesh first. A nozzle system is fixed to the fins of the condenser in such a way that it sprays water in the opposite direction to the condenser's air intake. The sprayed water is distributed on the mesh thereby cooling the air that passes through it. The main advantage of this system is that water is not directly sprayed onto the outdoor unit, causing tripping of the compressor and the destruction of the unit due to corrosion and water residues. Since a one-pass water circulation system is used, a water controller is used. The controller includes an ambient temperature sensor. The water circuit is switched on according to a set outdoor temperature range. In this experiment, the outdoor temperature range is set to 30–60°C. Furthermore, the controller allows

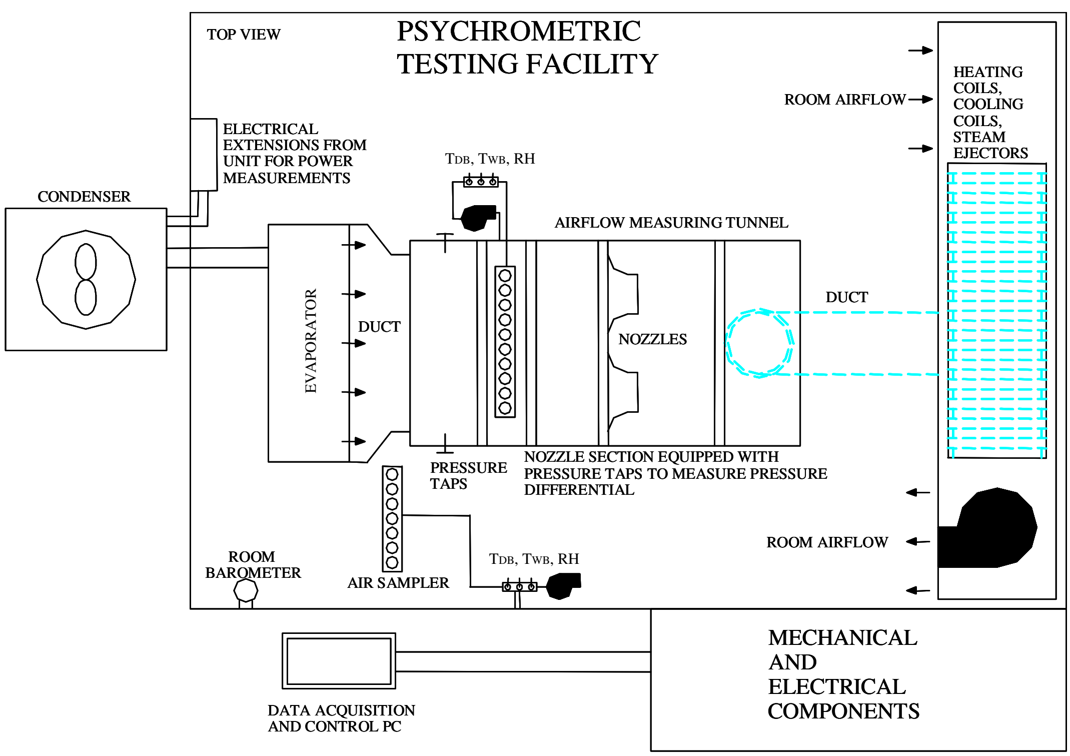

Figure 1. General testing schematic diagram.

the spraying of water at set time intervals. The controller is set to open the solenoid valve for 1 min and close it for 5 min. Excess water is not collected.

The CP setup involves constructing an aluminium frame around the four-sided condenser that holds four 20 mm cellulose bound cardboard CPs (one in front of each side of the condenser) and a perforated plastic tray on top of each cooling pad. A plastic water piping system, connected to the building's main water supply line fills the trays with water. The CPs get saturated with water thereby cooling the air that passes through them. A wooden cover is constructed to cover the opening gap between the condenser and the frame from the top so that it does not create a lower resistance area for the airflow to pass through. As with the CM setup, a one-pass water circulation system is used. The experiments are carried out using a similar water controller that controls the evaporative cooling system in the same way as explained for the CM setup.

The shade setup consists of a $2 \times 2\,\mathrm{m}^2$ steel frame that is fixed to the building's external wall, at 2 m above the condenser. The 2 m gap between the condenser and the frame is to allow the discharge of hot air without suffocating the condenser.

A Hewlett-Packard data logger is used to record all the measured parameters at 1 min intervals. A LabView monitoring and control program is used to monitor and control the psychrometric room conditions. After a steady state of 10 min is achieved, data is collected for 2 h and sent directly to MS-Excel. This data is, in turn, sent to Engineering Equation Solver (EES) through the Dynamic Data Exchange (DDE) available with EES and MS-Excel to perform cooling load and efficiency calculations.

The uncertainties in the experimentally determined values of the cooling load and the required power are estimated based on the ANSI/ASME Standard on Measurement Uncertainty (ASME, 1985) following the procedures of Coleman and Steele (1989).

The total uncertainty U in the measured values is expressed as follows:

$$U = (E_\mathrm{b}^2 + E_\mathrm{p}^2)^{1/2} \tag{5}$$

where E_b and E_p are the bias and the precision limits in the measured quantities.

The expression used in the determination of the cooling load is given in Equation (1). This expression can be expressed in the form

$$Q_m = Q_m(v_1, v_2, v_3, \ldots, v_m) \tag{6}$$

where m is the number of variables involved in the equation. The expressions for E_b and E_p are, respectively, given as follows:

$$E_\mathrm{p}^2 = \sum_{i=1}^{m} \left(\frac{\partial Q_m}{\partial r_i} e_\mathrm{pi} \right)^2 \tag{7}$$

$$E_\mathrm{b}^2 = \sum_{i=1}^{m} \left(\frac{\partial Q_m}{\partial r_i} e_\mathrm{bi} \right)^2 + 2 \left(\frac{\partial Q_m}{\partial r_1} \right) \left(\frac{\partial Q_m}{\partial r_2} \right) e'_\mathrm{b1} e_\mathrm{b2} + \cdots \tag{8}$$

where e_pi is the precision limit error in the variable r_i, e_bi is the bias limit error in the variable r_i and $e'_\mathrm{bi} e'_\mathrm{bj}$ are the correlated bias errors for the variables r_i and r_j.

The overall uncertainties in the cooling load and electric power range from 4 to 7% depending on the outside temperature and the size of the unit. However, the uncertainty analysis on the data reveals that the error is mostly bias and the data can be compared with each other with a good degree of accuracy. The uncertainty can reach 10% if the variation from unit to unit is included.

5. RESULTS AND DISCUSSION

The performance parameters that are of interest include the condenser inlet air dry bulb and wet bulb temperatures, the cycle's suction and discharge pressures, the power consumption of the unit, the cooling load of the unit and the coefficient of performance of the unit (COP).

Table II summarizes all the results for the three setups under consideration. The results are presented as the percentage increases (positive sign) or decreases (negative sign) relative to values obtained under normal operating conditions without any modification in the setup. Values in Table II will be referred to during the discussion of the figures.

The effect on the units' performance is plotted in Figures 2–5. Figure 2 shows the difference in air temperature measured across the condenser with and without the use of the CP. The figure presents the measurements under two different ambient conditions (45 and 39°C). It has been observed that by passing the condenser inlet air through wetted pads first, its dry bulb temperature drops by 30.6% at an ambient temperature of 45°C and by 26.3% at an ambient temperature of 39°C. As the air passes over wetted pads, it is adiabatically cooled and hence the dry bulb temperature drops. The change in wet bulb temperature depends mostly on the air dew point temperature and the water temperature. Under outdoor conditions of 45°C dry bulb and 39°C wet bulb, the dew point temperature is 38°C. Since the water used in the experiment was at a temperature of 30°C, the water vapour in the air condensed while passing through the wetted pads, and hence its wet bulb temperature decreased by 24%. This further decrease in the condenser inlet air humidity diminishes the possibilities of scaling problems usually associated with evaporative cooling.

The same trend of results obtained for the CP setup was obtained for the CM setup. Figure 3 shows the difference in air temperature measured across the condenser with and without the use of the CM. It has been observed that drop in condenser inlet air temperature is 23%, compared to a 30.6% drop in the CP setup. This difference in performance is expected due to the CPs ability to remain continuously wet, even during the intervals at which the water was off, whereas

Table II. Summary of the results of the different experimental setups for different ambient temperatures.

	Experimental setup					
	CP		CM		Shade	
Ambient temp. (°C)	45	39	45	39	45	39
Performance parameter	Percentage change					
Condenser inlet air temperature	−30.6	−26.3	−23	−33.9	12.9	9.6
Suction pressure	−37	−8.5	−20.9	−16.4	2.6	1.5
Discharge pressure	−35.3	−16.4	−23.8	−21.6	9	6.7
Power consumption	−19.7	−8.1	−20.5	−15.6	2.6	2.7
Cooling load	7.5	7.8	6.4	7.6	−17	−3
COP	36	18.4	32.9	59	−19.6	−4.9

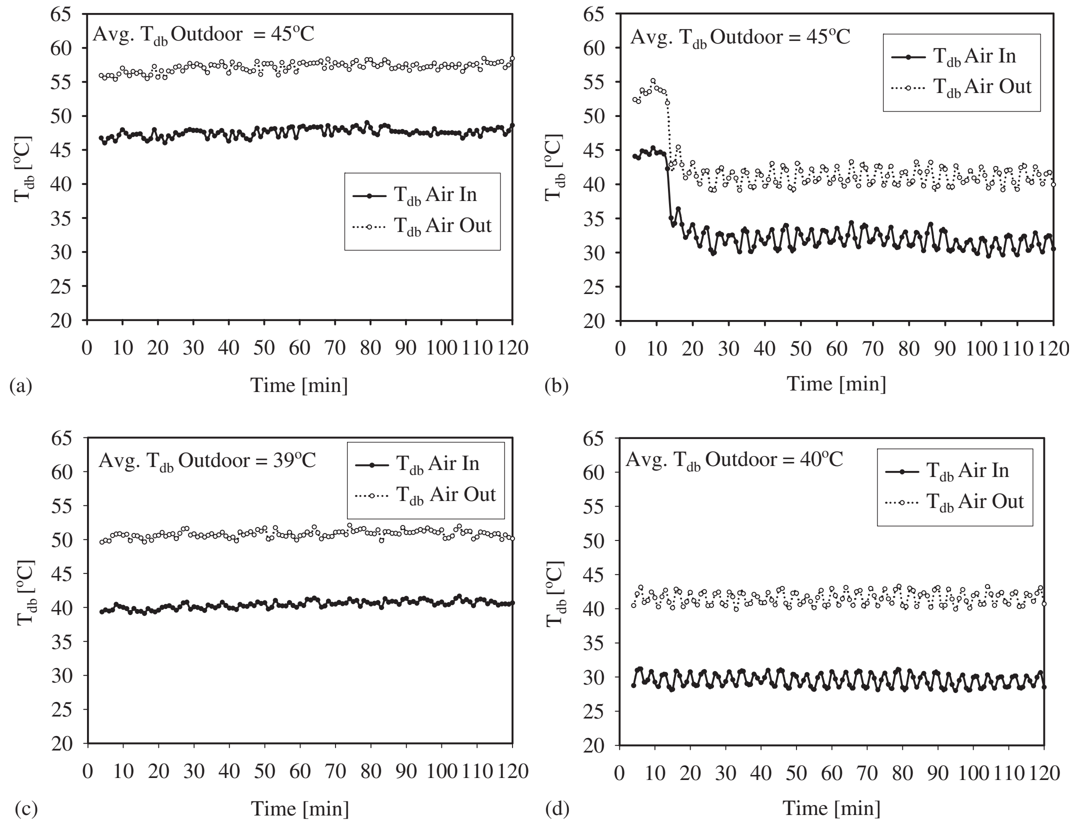

Figure 2. Cycle analysis for the air temperature difference across the condenser for the CP setup under different ambient conditions: (a) and (c) no CP; (b) and (d) with CP.

the mesh does not hold the water efficiently for a longer period of time, thereby allowing the air temperature to rise up to 45°C during the off intervals. This is clearly visible when comparing Figures 2 to 3. The fluctuation of the condenser air temperature is much higher in the CM setup than the CPs setup.

Figure 4 shows the results of the shade setup. An opposite trend of results is observed. Under ambient temperatures of 45 and 39°C, the condenser inlet air temperature increases by 12.9 and 9.6%, respectively. According to the model of the air-conditioning unit, the condenser's air outlet is positioned at the top of the outdoor unit. By placing the shading device 2 m on top of the unit, the hot air from the condenser outlet is trapped beneath the shade which results in heating up the region around the condenser. The heated air is partially re-circulated back into the condenser, thereby increasing the average inlet air temperature.

When choosing to apply a shading device to condensers, great care should be taken to avoid the recirculation of hot outlet air back into the condenser. The distance between the shading device and the condenser should be larger than 2 m for 2.8 ton units. The distance should increase with the increase in the unit's capacity. The units should also be placed sufficiently far from building walls to allow the dissipation of hot air away from the unit. Only under these

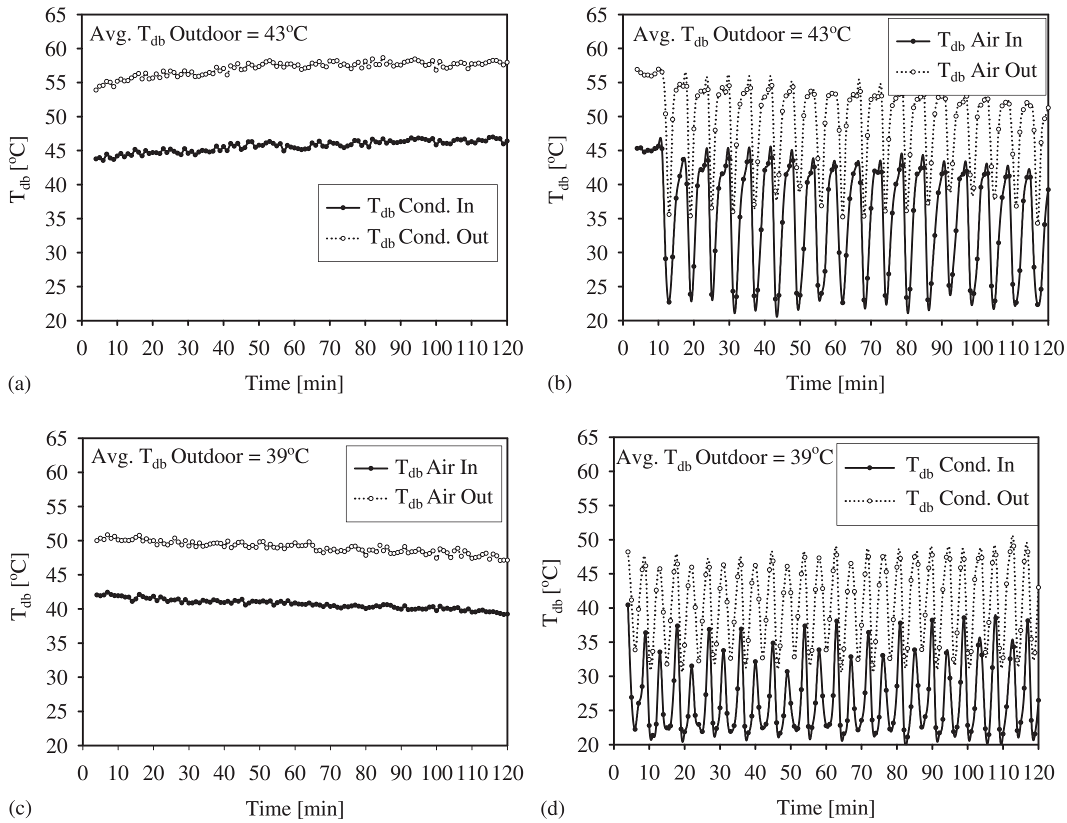

Figure 3. Cycle analysis for the temperature difference across the condenser for the CM setup under different ambient conditions: (a) and (c) no CM; (b) and (d) with CM.

conditions would the condenser air inlet temperature decrease and full advantage of the shading device be taken. If, however, the shading device was applied to a unit, which has its condenser air outlet positioned at the sides, then full advantage of the shade could be taken.

Figure 5(a)–(h) presents detailed results of the cycle pressures, the power consumption, the cooling load and the COP under ambient temperatures of 45 and 39°C as well as a comparison between the three different experimental setups.

The change in condenser inlet air temperature had a direct effect on the cycle's suction and discharge pressures, and hence on the power consumption of the units. For the CP setup, it has been observed that the suction and discharge pressures decrease by 37 and 35.3%, respectively, under an ambient temperature of 45°C. Under an ambient temperature of 39°C, the suction and discharge pressures decrease by 8.5 and 16.4%, respectively. Consequently, the power consumption decreases by 19.7% under ambient conditions of 45°C and by 8.1% under ambient conditions of 39°C. Usually, the higher the ambient temperature is, the higher the suction and discharge pressures will be and hence, higher power consumption of the unit. This explains the higher percentage drops in power consumption that is achieved at high ambient temperatures. The lower suction and discharge pressures add the advantage of increasing the

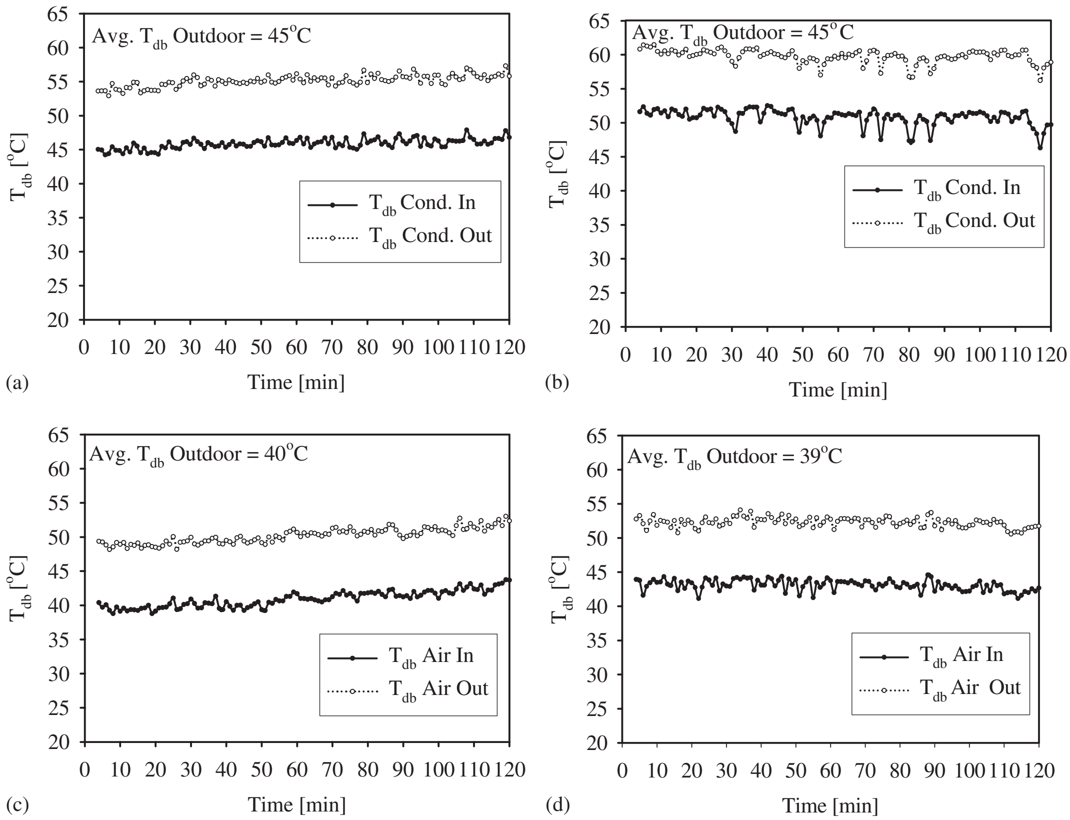

Figure 4. Cycle analysis for the temperature difference across the condenser for the shading setup under different ambient conditions: (a) and (c) under direct sunlight; (b) and (d) under shade.

compressor's life expectancy. A significant increase in the cooling load is observed. The lowered suction pressure results in further cooling of the air passing through the evaporator, hence, increasing the cooling load by 7.5%. Corresponding to the decrease in power consumption and increase in cooling load, the COP is found to increase by 36% under an ambient temperature of 45°C, and by 18.4% under an ambient temperature of 39°C.

Similarly, the results for the CM setup are presented in Figure 5. The reduction in power consumption is 20.5% under an ambient temperature of 45°C and 15.6% under an ambient temperature of 39°C. This is a higher reduction than is obtained from the CP setup. The thickness of the CP creates a resistance to the air flow, and hence creates a pressure drop in the air flow entering the condenser. This, in turn, increases the condenser fan power required which is included in the total power consumption. The increase in cooling load was observed to be higher using the CP setup than the CM setup (6.4% and 7.6% under ambient temperatures of 45 and 39°C, respectively, for the CM). This is again due to the ability of the CPs to hold water more efficiently. Corresponding to the decrease in power consumption and increase in cooling load, the COP of the CM setup are found to increase by 32.9% under an ambient temperature of 45°C, and by 59% under an ambient temperature of 39°C.

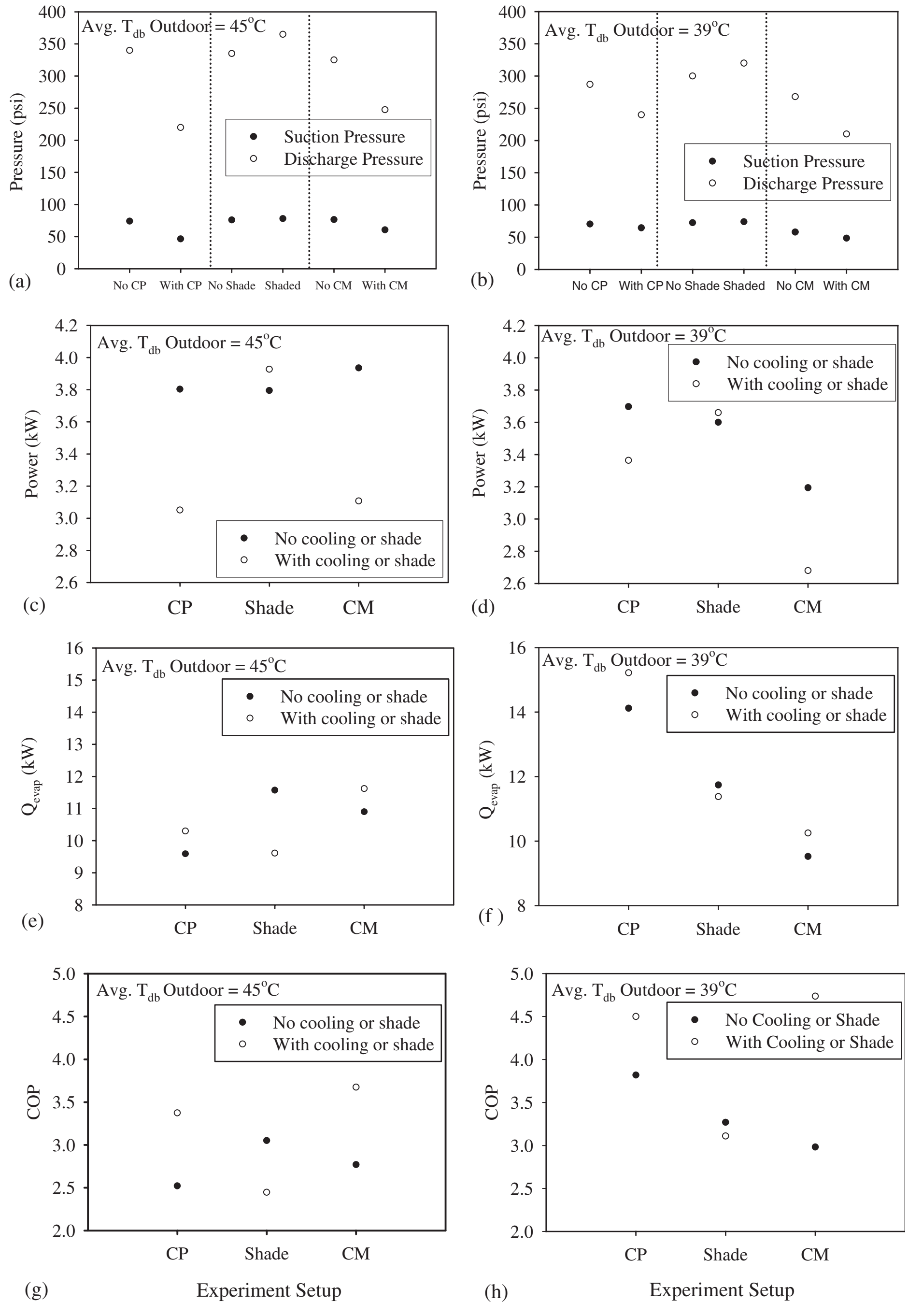

Figure 5. Effect of application of the different condenser air pre-cooling setups on the performance of the air-conditioning unit.

It is also noticed from Figure 5 that under 45°C, the CM method is advantageous over the CP method (higher cooling load and COP), but under 39°C, the CM method is disadvantageous over the CP method (lower cooling load and COP). The performance of an air-conditioning unit under a certain evaporative cooling method, is not directly proportional to ambient dry bulb temperature alone since ΔT across the condenser varies according to both ambient dry and wet bulb temperatures.

As per the shade arrangement, the increase in condenser inlet air temperature has a negative effect on the unit's performance. The suction pressure increases by 2.6 and 1.5% under ambient temperatures of 45 and 39°C, respectively, discharge pressure increases by 9 and 6.7%, respectively, power consumption increases by 2.6 and 2.7%, respectively, cooling load decreases by 17 and 3%, respectively, and the COP decreases by 19.6 and 4.9%, respectively.

The operating costs (including costs of electricity and water consumption) of the CP and CM setups are calculated according to the costs of electricity and water in Kuwait. The cost of $1\,\mathrm{kW\,h}^{-1}$ in Kuwait is \$0.0475. The cost of water varies from year to year. An average value of $\$2.112\,\mathrm{m}^{-3}$, obtained from the Ministry of Electricity and Water (MEW), is used in the calculations. The cost of water in Kuwait is exceptionally high since 75% of the fresh water is produced by multi stage flash desalination, and only 25% is groundwater, according to the MEW. Table III summarizes the average consumption of water and electricity used in the CP and CM setups, and the total operating costs for each case. An average of $0.56\,\mathrm{m}^3$ of water is used during the continuous water flow operation of the CP, while only an average of $0.16\,\mathrm{m}^3$ is used with the water controller. Since the CP remained moist during the 2 min off interval of the controller, results did not vary much. The amount of water consumed using the controller is used in the cost analysis presented in Table III. The same amount of water is used with the CM setup. The cost analysis presented in Table III shows that due to the relatively high cost of fresh water production in Kuwait, the application of the CP and CM methods is slightly more expensive, on a consumer level. However, the energy conservation and the decrease in peak load demand on the power stations are considerably high. On the other hand, other sources of water could alternatively be used, such as brackish water, since it is cheaper than fresh water. Furthermore, since the water does not come in direct contact with the condenser coils in both the CP and CM setups, mineral deposits of brackish water on the condenser coils would not be a problem. The mesh and pads, however, should be regularly replaced if brackish water is to be used. If, however, the CP and CM methods are used in other parts of the world where river water is available or the cost of water production is relatively low, higher cost savings will be achieved.

Table III. Summary of the consumption of electricity and water in the CP and CM setups.

	Normal operation	CP	CM
Average water consumption (m^3)	0	0.56	0.56
Average electrical consumption (kW h)	7.7	6.41	6.9
Cost of water (\$)	0	1.183	1.183
Cost of electricity (\$)	0.3663	0.3045	0.3278
Total operating cost (\$)	0.3663	1.4875	1.5108

6. CONCLUSIONS

By the application of the two evaporative cooling methods and the proper installation of shading devices, the overall energy conservation in Kuwait's power plant systems will be significant and could greatly reduce the peak demand of power plants, and hence reduce the capital costs invested in the construction of future power plants.

The application of both the CP and CM setups yielded high power savings ranging from 8.1 to 20.5%. The resulted increase in COP was considerably high, ranging from 36 to 59%. These savings, along with the elimination of water contact with the condenser coils, makes the use of the CP and CM methods a highly considerable option of energy conservation in Kuwait.

In choosing between the installation of the CP and CM setups, the CP setup is preferred for the fact that the pads remain moist during the off period of the water controller, whereas the mesh tends to dry up faster, resulting in a slightly less performance.

The results of the shading setup yielded a different trend of results. An increase of 2.6–2.7% in the power consumption has been observed, which resulted in a drop in the COP by 4.9–19.6%. This is a result due to the position and setup of the shading device relative to the design of the condenser. From here, it is concluded that the condenser should not be installed directly next to building walls or obstacles on the assumption that the shade created by these obstacles would increase the performance of the unit. It is also concluded that shading devices should be placed considerably away from the condensers so as not to suffocate the condenser. That way, full benefits would be gained from the shade.

NOMENCLATURE

A = area (m^2)
CM = cooling mesh
COP = coefficient of performance
CP = cooling pad
h = enthalpy (kJ kg^{-1})
$\dot{m}$ = mass flow rate (kg s^{-1})
P = pressure (kPa)
Q = cooling load (kW)
Q_{BTU} = cooling load (BTU)
R = gas constant of air
RH = relative humidity
T_{db} = dry bulb temperature
T_{wb} = wet bulb temperature
V = velocity

Greek letters

ρ = density
ΔP = pressure differential

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