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SENSITIVITY ANALYSIS OF A NOVEL HUMIDIFIER RETROFITTED INDIRECT EVAPORATIVE COOLER

Muhammad Ahmad Jamil[1*], Kim Choon Ng[2], Ben Bin Xu[1], Muhammad Wakil Shahzad[1*]

[1]Mechanical & Construction Engineering Department, Northumbria University, Newcastle Upon Tyne, NE1 8ST, UK
[2]Water Desalination and Reuse Centre, King Abdullah University of Science and Technology, Thuwal 23955, Saudi Arabia

ABSTRACT

The current work is focused on investigating a novel humidifier retrofitted indirect evaporative cooler. The cooler is designed to overcome the major limitations associated with water management, surface wettability, fouling, and unreliability of hydrophilic surfaces. The proposed system achieved high heat transfer capability, low manufacturing, and maintenance cost, and high operational reliability because of a major design change of water-air interaction outside the cooler body. The experimental analysis revealed that the proposed system can achieve an efficiency of as high as 83% which is comparable with the existing systems in addition to offering the aforementioned benefits. A rigorous calculus-based sensitivity analysis showed that the cooler coefficient of performance and efficiency becomes less sensitive to the primary air inlet temperature at higher primary air inlet temperatures. While their sensitivity on the primary air outlet temperature increases with an increasing outlet temperature of the primary air. The presented results are a base for further development of the proposed system for expansion at commercial-scale testing.

Keywords: Indirect evaporative cooler, humidifier assisted, sensitivity analysis, experimental and numerical analysis.

1. INTRODUCTION

Indirect evaporative cooling (IEC) technology has emerged as a sustainable alternative to chemical-based cooling systems. This is because the conventional cooling systems are not only energy-intensive (with global energy consumption of ~7 EJ in 2015) but also responsible for significant carbon emissions (~170 gigatons) [1]. The prime reason for the high energy consumption of these systems is the inherent inefficiency caused by a simple cooling coil-based operation which involves simultaneous sensible and latent load handling. Therefore, the specific electricity consumption and coefficient of performance (COP) for these systems have not seen any significant improvement in the last few decades and are stagnant at 0.85±0.05 kW/Rton and COP 3-3.5, respectively [2]. On the other hand, the IEC systems employ fresh water and evaporative potential of air for cooling, thus eliminating the need for low-efficiency compressor as well as chemical-based refrigerant from the system [3]. The generic cells for these systems developed and tested on the laboratory scale have shown excellent potential for the future cooling market as they can achieve COP as high as 10 (~3 times the conventional systems). However, their commercial-scale development is still in progress because of several major design limitations. These include water management in the wet channel, surface wettability, hydrophilic membrane fouling, and peeling of wicking material with which the surface is enhanced for better water retention. These issues reduce system operational availability, reliability and increase maintenance requirements as well as high operational expenses. Therefore, the current focus in the cooling industry is to resolve these issues for commercial-scale development. One important advancement in this regard is the retrofitting of a humidifier with the conventional IEC system to efficiently handle the water-air interaction. The detailed design and working of the proposed system are presented in the following sections.

*Corresponding Authors: muhammad2.ahmad@anorthumbria.ac.uk, muhammad.w.shahzad@northumbria.ac.uk

2. HUMIDIFIER RETROFITTED IEC SYSTEM

The proposed system consists of an IEC cooler with dry and wet channels separated by a thin (0.025 mm) aluminum foil, a humidifier, and a blower. The process flow diagram for the proposed system is presented in **Figure 1.** The blower is attached at the outlet of the dry channel to create induce draft for outside air in the dry channel and a forced draft for wet air in the wet channel for minimum energy consumption. The dry air at the dry channel outlet is purged (depending upon purge ratio i.e., 35-55 in this case) to the humidifier where it is cooled and humidified (RH = 100%) through water showering using fine mist nozzles. This cold humid air is then directed to the wet channel to cool down the outside air in the dry channel through orthogonal heat transfer. It is important to mention that the proposed configuration addresses all the major limitations identified in the conventional IEC systems. This is because, in the current system, all the accessory items with maintenance requirements like a humidifier, pump, blower, etc. are placed outside the system with easy access without opening the main cooler. Similarly, water air interaction occurs in the humidifier where it can be optimized without changing the cooler configuration. Moreover, the wet channel does not require any hydrophilic membrane or felt material, rather a high thermal conductivity aluminum foil separates the dry and wet channel. This increases the orthogonal heat transfer rate eliminating the additional resistances due to the felt material and water layer. Furthermore, the complexities and cost associated with heat transfer surface development (e.g., sticking of wicking material, etc.) evade because of simple aluminum foil use. Therefore, the proposed system is underpinning the commercial-scale IEC development. The current study presents, an experimentally verified [4] numerical simulation-based sensitivity analysis of a generic cell of the proposed system. The design specifications of the cooler are presented in **Table 1**. The sensitivity of inlet and outlet primary temperature is investigated. For this purpose, an Engineering Equation Solver-based numerical code is developed to simulate the sensitivity analysis based on the partial derivative. The results are presented in terms of Normalized Sensitivity Coefficients (NSC).

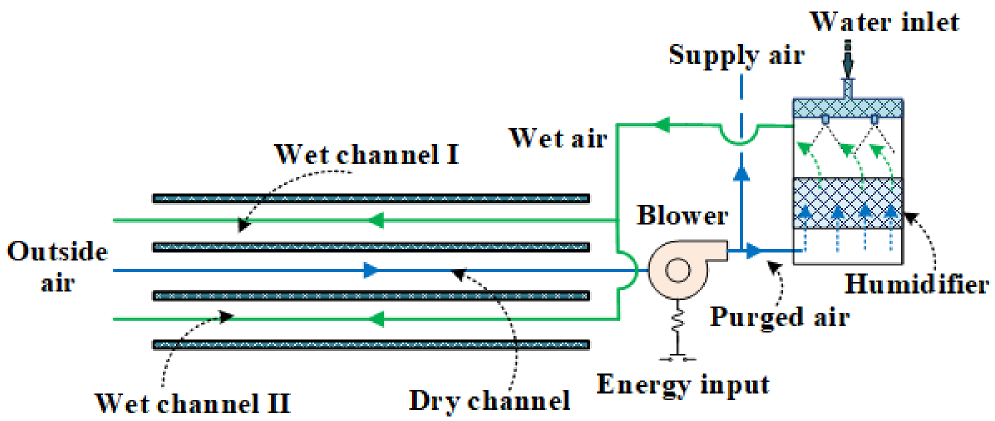

Figure 1: The proposed IEC systems, (a) schematic diagram, and (b) psychrometric chart.

Table 1. Design and operational parameters of the proposed generic cell.

Parameter	Value
Generic cell effective length, *L*, mm	1800
Generic cell effective width, *W*, mm	280
Separator Aluminium foil thickness, *δ*, mm	0.025
Dry and wet channels height (spacing), *H*, mm	5
Purge ratio, *r*, %	35-55

3. MATHEMATICAL FORMULAE

The coefficient of performance (COP) is calculated on the cooling load and the power input i.e., the energy consumption of the blower.

$$COP = \frac{\dot{Q}_{pa}}{P_{blower}} \tag{1}$$

The cooler effectiveness is defined based on the enthalpies which are calculated as a function of temperature, pressure, and humidity ratio.

$$\eta = \frac{h_{pa,i} - h_{pa,o}}{h_{pa,o} - h_{sa,o}} \ \& \ \eta_{wb} = \frac{T_{pa,i} - T_{pa,o}}{T_{pa,i} - T_{sa,wb,i}} \tag{2}$$

Normalized sensitivity coefficients allow a one-to-one comparison of parameters whose magnitude can vary significantly [5,6] and are calculated by normalizing the uncertainties in output parameters (Y) and input parameters (X) by their nominal values. The general form of the equation is given in Eq. 3 where, U_Y is the total uncertainty in $\bar{Y}$ is the nominal value of Y [7].

$$\frac{U_Y}{\bar{Y}} = \left[\sum_{i=1}^{N} \overbrace{\left(\frac{\partial Y}{\partial X_i} \frac{\bar{X}_i}{\bar{Y}} \right)^2}^{NSC} \overbrace{\left(\frac{U_{X_i}}{\bar{X}_i} \right)^2}^{NU_{Xi}} \right]^{1/2} \tag{3}$$

For each response parameter, the above general equation can be modified in terms of independent parameters. Such as, for wet bulb effectiveness η_{wb}, the above equation can be given as:

$$\frac{U_{\eta_{wb}}}{\bar{\eta}_{wb}} = \left[\begin{array}{l} \left(\frac{\partial \eta_{wb}}{\partial T_{pi}} \frac{\bar{T}_{pi}}{\bar{\eta}_{wb}} \right)^2 \left(\frac{U_{T_{pi}}}{\bar{T}_{pi}} \right)^2 + \left(\frac{\partial \eta_{wb}}{\partial T_{po}} \frac{\bar{T}_{po}}{\bar{\eta}_{wb}} \right)^2 \left(\frac{U_{T_{po}}}{\bar{T}_{po}} \right)^2 \\ + \left(\frac{\partial \eta_{wb}}{\partial T_{si}} \frac{\bar{T}_{si}}{\bar{\eta}_{wb}} \right)^2 \left(\frac{U_{T_{si}}}{\bar{T}_{si}} \right)^2 + \left(\frac{\partial \eta_{wb}}{\partial T_{so}} \frac{\bar{T}_{so}}{\bar{\eta}_{wb}} \right)^2 \left(\frac{U_{T_{so}}}{\bar{T}_{so}} \right)^2 \end{array} \right]^{1/2} \tag{4}$$

4. RESULTS AND DISCUSSION

The experimentation results of the proposed cooler [4] showed that (refer to **Figure 2**) the maximum cooler effectiveness is calculated as 83.82% at the highest outdoor air temperature (43 °C) and purge air ratio (55%). It asserts that the highest performance of the cooler is achieved at the maximum outdoor air temperature and the purge air ratio and the performance is comparable with the existing systems. The findings of sensitivity analysis are presented in the form of sensitivity coefficient (SC), Normalized Sensitivity Coefficients (NSC), and finally sorting the parameters in descending importance ($\Lambda \downarrow$). In this regard, **Table 2** presents the sensitivity of COP on different input parameters i.e., primary air inlet/outlet temperatures ($T_{pa,i}/T_{pa,o}$), secondary air inlet/outlet temperatures ($T_{sa,i}/T_{sa,o}$), primary/secondary air velocity (V_{pa}/V_{sa}), and humidity ratio of primary air (ω_{pa}). It is observed that the COP is the most sensitive to $T_{pa,i}$ with NSC = 4.285 followed by $T_{pa,o}$, and V_{pa} with NSCs of 1.396, and 0.999, respectively. The other parameters are almost insensitive to COP, however, their NSCs follow the order as ω_{pa}> $T_{sa,i}$,> V_{sa},> $T_{sa,o}$. **Figure 3** shows the variation in NSC of the COP and η against inlet temperature of primary air keeping other parameters constant. It is observed that the values COP and ηare increased by increasing the inlet temperature of primary air because the difference between the inlet and outlet temperature increases. However, the combined effect of all other constant parameters and perturbation causes the NSC of COP and η to decrease by increasing the inlet temperature of primary air. Therefore, it indicates that the sensitivity of COP and η over $T_{pa,i}$ decreases at higher values. Similarly, **Figure 4** shows the variation of NSC of the COP and η against the outlet temperature of primary air keeping the other operating parameters constant. It is seen that the values of COP and η decreased due to a decrease in the difference between the inlet and outlet temperature of primary air. However, the overall impact of constant quantities and perturbation causes to increase in the NSC of COP and η. It means that the COP and η will be more sensitive to the $T_{pa,o}$ at its higher values.

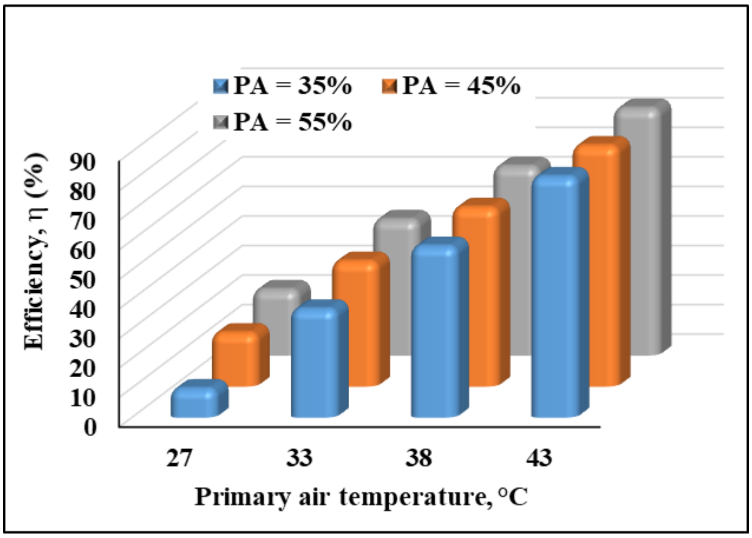

Figure 2. Cooler efficiency at different primary air temperatures and purge air ratios.

Table 2. The sensitivity of the coefficient of performance (COP) to different parameters.

Variable	$U_{X\,i}$	$\bar{X}$	SC	NSC	$\Lambda \downarrow$
$T_{pa,i}$	1 °C	45	4.145	4.285	$T_{pa,i}$
$T_{pa,o}$	1 °C	24	4.748	1.396	$T_{pa,o}$
$T_{sa,i}$	1 °C	22.2	3×10^{-32}	7×10^{-33}	V_{pa}
$T_{sa,o}$	1 °C	25.7	0	0	ω_{pa}
V_{pa}	1 %	5.2	0.195	0.999	$T_{sa,i}$
V_{sa}	1 %	2.86	1.2×10^{-35}	6×10^{-35}	V_{sa}
ω_{pa}	1 %	0.01	1.12×10^{-06}	5×10^{-6}	$T_{sa,o}$

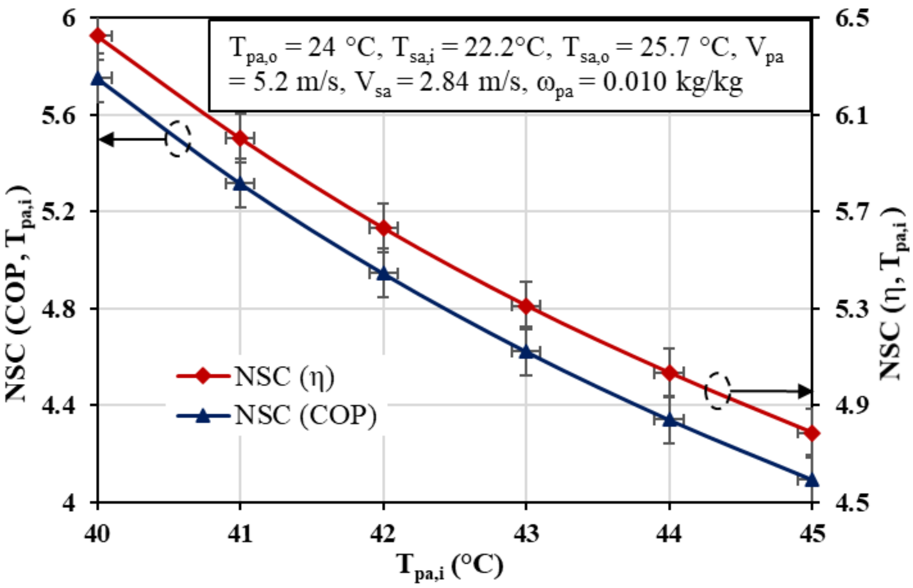

Figure 3. Variation in NSC of the COP and η against $T_{pa,i}$.

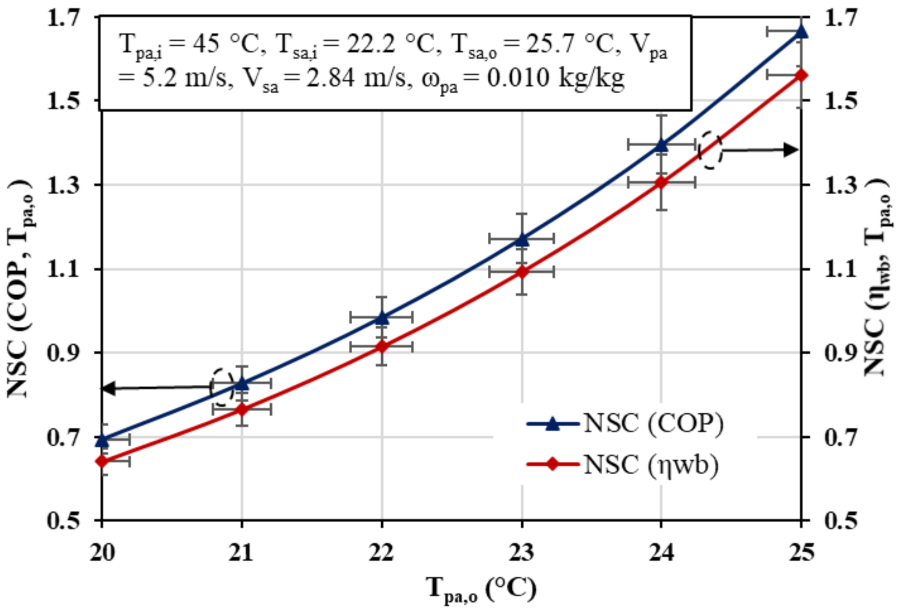

Figure 4. Variation in NSC of the COP and η_{wb} against $T_{pa,o}$.

5. CONCLUSIONS

A humidifier retrofitted indirect evaporative cooler design resolving the major limitations in the existing IEC systems is proposed and analyzed. The analysis reveals that the proposed design can achieve a cooling efficiency ≥ 80% which is comparable to existing systems as well as has a low maintenance cost, high system reliability, and easy manufacturing. Moreover, the sensitivity analysis showed that the sensitivity of COP and efficiency on inlet primary air temperature decreases at higher values. However, the sensitivity of COP and efficiency on outlet primary air temperature increases at higher values. These findings can be used for further development of the proposed system.

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