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Energy performance of air-conditioning systems using an indirect evaporative cooling combined with a cooling/reheating treatment

Claudio Cianfrini, Massimo Corcione*, Emanuele Habib, Alessandro Quintino

DIAEE Sezione Fisica Tecnica – Sapienza Università di Roma, via Eudossiana 18, 00184 Rome, Italy

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ABSTRACT

The energetic performance of an integrated energy-recovery system for air-conditioning applications consisting of an indirect evaporative cooling equipment combined with a cooling/reheating unit is analyzed numerically using an in-house-developed computer code. Simulations are performed for a wide variety of operating conditions and main features of the heat exchangers. The minor energy consumption consequent to the energy transfer from outdoor air to saturated indoor air to be exhausted and, subsequently, to supply air to be reheated is compared with that deriving from the adoption of traditional energy recovery strategies, and calculated by introducing a cooling effectiveness parameter. An empirical dimensionless correlation that expresses the cooling effectiveness parameter as a function of the several independent variables considered is also proposed.

1. Introduction

Air-conditioning accounts for a 60–70% of energy use in buildings, which, in turn, can be estimated to be of the order of 40–45% of the total energy consumption of developed countries, as reported in a survey recently collected by Pérez-Lombard et al. [1]. This is one of the reasons why the adoption of sustainable HVAC strategies has become a topic upon which nations are focusing a number of new dispositions in matter of rational use of energy, see, e.g., Directive 2002/91/EC [2]. In this perspective, evaporative cooling, both direct and indirect, may help to reduce the energy demand of air-conditioning systems.

Direct evaporative cooling is performed keeping the airstream to be cooled in direct contact with a liquid water film. The resulting water evaporation gives rise to an air temperature decrease, but, at same time, to a moisture content increase, which limits the applicability of such a cooling technique to hot arid climates.

Indirect evaporative cooling is the alternative option for hot humid climates. In fact, its main feature is that air cooling is achieved without any increase in moisture content, which extends its effective applicability to a wider range of climatic conditions. Indirect evaporative coolers are heat exchangers, mainly plate-type units or finned-tube coils, in which hot outdoor air to be cooled flows through dry-surface passages, while colder indoor air to be exhausted travels along wet-surface passages where water evaporation occurs. Since the temperature of outdoor air at exit cannot be lower than the wet-bulb temperature of the incoming indoor air, in most cases indirect evaporative coolers are operated in combination with a conventional cooling equipment, typically a finned-tube coil supplied with either chilled water or refrigerant, in order to ensure both temperature and humidity control inside the conditioned spaces.

Indirect evaporative cooling has been abundantly studied in the past decades, as reflected by the significant research effort dedicated to this subject, whose main results are well summarized in a couple of recent review articles compiled by Duan et al. [3], and Santamouris and Kolokotsa [4]. However, no correlation is readily available for the prediction of the minor energy consumption that derives from the installation of an indirect evaporative cooler, expressed as a function of the heat exchanger features and operating conditions.

A related issue worth being cited and discussed is the fact that quite often the ventilation rates employed to ensure acceptable levels of indoor air quality are rather high. As a consequence, the outdoor air that leaves the cooling coil at the low moisture content required for indoor humidity control usually needs to be reheated before its introduction into the conditioned space. Accordingly, the indirect evaporative cooler could be conveniently followed by a cooling/reheating unit for energy transfer between the outdoor air that needs to be further cooled and the same outdoor air that leaves the cooling coil and must be reheated at an acceptable supply temperature level.

Framed in this background, a numerical study on the performance of an integrated energy-recovery system consisting of an

* Corresponding author. Tel.: +39 06 44 58 54 43; fax: +39 06 48 80 120.
E-mail address: massimo.corcione@uniroma1.it (M. Corcione).

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Nomenclature

A	heat transfer surface area (m^2)
c	constant pressure specific heat (J/kg K)
C_i	i-th coefficient of the Hyland–Wexler equation (i = 1 to 6)
G	mass airflow rate (kg/s)
h_C	average coefficient of convection heat transfer (W/m^2 K)
h_D	average coefficient of water vapour transfer (kg/s m^2)
i	specific enthalpy of moist air (J/kg_a)
NTU	number of transfer units
p	pressure (Pa)
T	temperature (K)
t	temperature in Celsius degrees (°C)
U	overall heat transfer coefficient (W/m^2 K)
w	humidity ratio (kg_v/kg_a)

Subscripts

a	dry air
dew	dew point
in	inlet
j	referred to the j-th airstream (j = 1, 2)
O	outdoor air
out	outlet
sat	saturation
t	total
v	water vapour
W	separation wall
wet	wet bulb
0	at 0 °C
1	referred to the primary hot airstream
2	referred to the secondary cold airstream

Greek Symbols

δ	Boolean operator
ε	effectiveness in sensible heat transfer of the heat exchanger
ε_{sat}	saturation effectiveness of the adiabatic air washer
γ	heat capacity rate ratio
λ	latent heat of vaporization of water (J/kg)

indirect evaporative cooling equipment combined with a cooling/reheating unit is executed. Simulations are performed for a wide variety of operating conditions and different characteristics of the heat exchangers. Primary scope of the paper is to investigate in what measure the energy transfer from outdoor air to saturated indoor air to be exhausted and, subsequently, to supply air to be reheated affects the energy performance of the HVAC system in comparison with traditional energy recovery strategies, as well as to develop accurate correlations for predicting the consequent energy saving.

2. Theoretical model

The integrated energy-saving system (IES), whose performance is analyzed here, basically consists of an air-washer humidifier (AW) coupled with an indirect evaporative cooler (IEC), and a cooling/reheating unit (CRU). Supplementary to a supply fan (SF), an exhaust fan (EF), and a filter (F), other components of the air-conditioning system for summer operation, which our attention is focused on, are a conventional cooling coil (CC) equipped with a droplet eliminator (DE) and a drain pan (DP), two coupled dampers (CD), and an additional conventional reheating coil (RC). A sketch of the whole arrangement is depicted in Fig. 1.

The operational mode is now described. Indoor air extracted from within the building at state I is subjected to a direct evaporative cooling in the adiabatic humidifier AW up to reaching state Y and then delivered to the indirect evaporative cooler IEC, where outdoor air enters at state O. The states of exhaust air and outdoor air at the exit of IEC are denoted as X and F, respectively. Outdoor air at state F is then further cooled in the cooling/reheating unit CRU, in which the same outdoor air that leaves the conventional cooling coil CC, here called supply air, enters at state R, whose humidity ratio is that required for the indoor ambient humidity control. The states of outdoor air and supply air at the exit of CRU are denoted as C and H. The final cooling treatment of outdoor air from state C to state R is executed in the conventional cooling coil CC. As far as supply air is concerned, an indoor air temperature controller modulates the mutual position of the coupled dampers CD thus permitting cool air at state R to bypass CRU. This cool air is then mixed with the hot airstream leaving CRU in the proper proportions so as to satisfy the required indoor ambient temperature control. When the bypass damper is closed, i.e., the whole airstream flows through CRU, the indoor air temperature controller operates on the conventional reheating coil RC, so that supply air is heated from state H to state S. The thermodynamic states of moist air at both inlet and outlet of any component, and the corresponding psychrometric paths, are displayed in Fig. 2.

The energetic performance of the system described above is calculated by way of a numerical code based on a set of specifically developed simulation models, one relative to the behavior of the adiabatic air washer AW, another relative to the behavior of the indirect evaporative cooler IEC, and a further one relative to the behavior of the cooling/reheating unit CRU, whose basic outlines are described below.

2.1. Model of the adiabatic air washer

The temperature and humidity ratio of moist air at the exit of the adiabatic air washer are calculated through the saturation effectiveness ε_{sat} defined as [5]:

$$\varepsilon_{sat} = \frac{T_{in} - T_{out}}{T_{in} - T_{wet}}, \tag{1}$$

where T_{in} and T_{out} are the inlet and outlet dry-bulb temperatures, respectively, and T_{wet} is the wet-bulb temperature of air at inlet, whose value is calculated under the assumption of constant specific enthalpy of moist air by a trial-and-error procedure based on the following equations:

$$i_{in} = i_{sat} \tag{2}$$

$$i_{in} = c_a(T_{in} - 273.15) + w_{in}[\lambda_0 + c_v(T_{in} - 273.15)] \tag{3}$$

$$i_{sat} = c_a(T_{wet} - 273.15) + w_{sat}[\lambda_0 + c_v(T_{wet} - 273.15)] \tag{4}$$

$$w_{sat} = 0.62198\frac{p_{vsat}(T_{wet})}{p_t - p_{vsat}(T_{wet})} \tag{5}$$

$$\ln[p_{vsat}(T_{wet})] = \frac{C_1}{T_{wet}} + C_2 + C_3T_{wet} + C_4T_{wet}^2 + C_5T_{wet}^3 + C_6\ln(T_{wet}). \tag{6}$$

In the above equations i_{in} and w_{in} are the specific enthalpy and humidity ratio of moist air at inlet; i_{sat} and w_{sat} are the specific enthalpy and humidity ratio of moist air at saturation; c_a and c_v are the constant-pressure specific heats of dry air and water vapour; λ_0 is the latent heat of vaporization of water at 0 °C; p_{vsat} is the water vapour saturation pressure; p_t is the total barometric pressure; and