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Construction Science

Investigation of Combined Indirect Evaporative Ducted Cooling Equipment Efficiency in Historical Building in Temperate Climate

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Abstract: **The present study is devoted to the evaluation of efficiency of the combined indirect evaporative – compressor water cooling system under various outdoor air humidity conditions of temperate climate. This is a building- based study, which represents the results of the analytical research, conducted in the recently restored 19th century historical building, The Art Museum Riga Bourse.**

Indirect adiabatic water chiller is equipped with a compressor, and supplies cooled fluid to the conventional air conditioning system, consisting of ventilation cooling coils and fan-coil units on the separated loop. Using the data acquired by the data logging system, we have analyzed the dependence of the cooling plant operation efficiency on outdoor air humidity for the cooling system operation period of the year 2012.

Keywords: **indirect evaporative, cooling system, historical building**

I. Introduction

Evaporative cooling is energy-efficient, environmentally friendly, and cost-effective in many applications. Innovative schemes combining evaporative cooling with other equipment have resulted in energy efficient designs [1].

The present engineering tendencies shows that due to the development of HVAC system and control equipment, this method of cooling becomes even more attractive for use not only in hot and arid regions, but also in the European countries with temperate climates. The main factors for choosing one or another type of cooling equipment are climate, cost efficiency, sizes, and availability of external recourses, such as spare heat energy, or the proximity of water sources. Outdoor air humidity is an element of climate, which affects heat transfer in air heat exchangers, and needs to be taken into account in calculations [1]. Recent studies showed that in case of water-air heat exchangers relative air humidity level increase from 50 % to 90 % results in heat transfer coefficient α growth by 1.68 times [2]. Facao and Oliveira obtained several correlations for mass and heat transfer coefficients by using different thermal simulation models for a closed wet cooling tower. They found that the influence of water inlet temperature on tower efficiency is negligible, it was also stated that simpler models with a global approach can give as good, or even better, results as the models based on finite difference techniques [13]. Costelloe and Finn found that thermal effectiveness is not affected by changes in load. Their study showed that the effectiveness is inherently greater when the external component of the cooling load is higher in summer, which significantly strengthens the case for water-side evaporative cooling in buildings. The results also suggest that the dry bulb temperature is a more important constituent of the adiabatic saturation temperature than the humidity ratio (particularly when the humidity ratio is above 6 g/kg) [4]. The dependence of evaporative cooling on the weather condition puts limitations on its applicability. One of the methods to use an adiabatic effect to increase energy efficiency of the cooling system is to use it in combination with the compressor-refrigerant system. For example, Casvendi et. al. reported about 14 % EER and 6 % air-cooled chillers' cooling capacity increase, when using mist-assisted condensers [5].

Latvia is located in the moderate climatic zone. Its temperature, moist climate is created by the Atlantic air masses and influenced by the Baltic Sea and the Gulf of Riga. Even in summer the outside air very rarely corresponds to the necessary supply air parameters. The rest of the time the air should be heated and dehumidified to achieve the necessary room air temperature and humidity [3]. The hourly outdoor air parameters in a year are shown in Fig. 1.

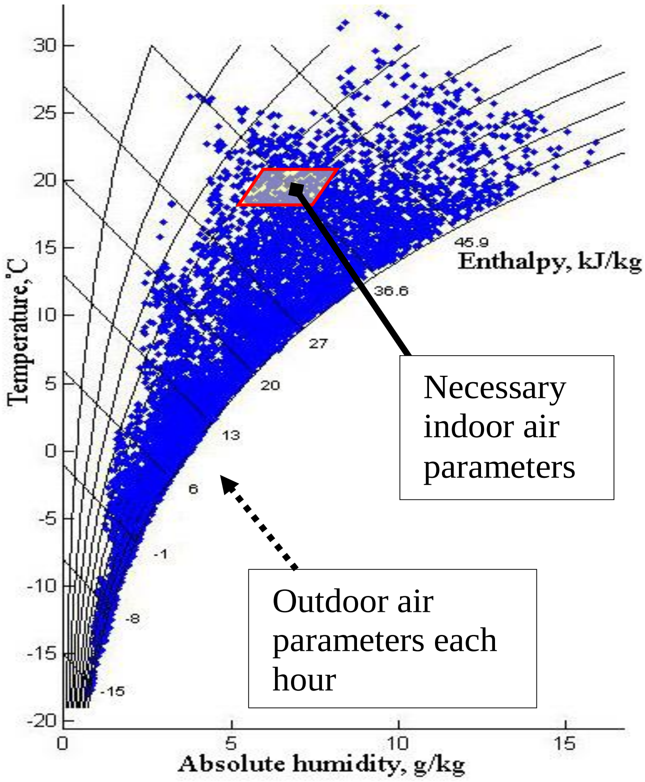


Fig. 1. Hourly average outdoor air parameters of the typical year [6].

Restoration of old buildings is a complex construction process, during which engineers and architects need to solve many atypical tasks concerning not only the structural stability of the building, but also the recovery of cultural-historical appearance of the building. The need to harmoniously integrate modern HVAC devices in the historical interior also enforces limits to the equipment selection. In these cases, the use of high temperature cooling, such as chilled ceilings, and chilled beams becomes complicated because of aesthetics requirements and the need for air dehumidification [6].

This paper focuses on the investigation of the impact of outdoor air humidity on the combined evaporative pre-cooled air to water chiller's energy efficiency ratio (EER). It is a building-based study, which represents the results of the analytical research conducted on the existing cooling system in the historical building (Fig. 2).

Fig. 2. The Riga Bourse building nowadays.

This is our third and the biggest study on the mentioned building. First and second studies cover smaller time periods depending on the accessible data. The investigated system is located in the recently restored building of the Art Museum Riga Bourse, which was initially built in the middle of the 19th century. It was recently reconstructed and opened in August 2011.

II. Materials and Methods

A. Air conditioning equipment concept

The cooling system consists of the ducted indirect evaporative air cooled water chiller with the integrated compressor, 5 air handling units with supply air cooling / dehumidifying coils and 98 recirculation fan-coil units on the separated loop.

The chiller, which we have used as an experimental unit, is located in the technical room in the attic, connected to the outside air duct system and equipped with an air-water plate heat exchanger (8), which cools the secondary loop with adiabatically pre-cooled outdoor air. Outdoor air is driven by the radial fan (5), pre-cooling is provided by water nozzles, located in adiabatic loop. Heat rejection from the compression cycle (4) is done by plate heat exchangers refrigerant/air (1), and refrigerant/water (10). Thereby, the primary loop, which supplies the building, on demand is cooled by the other one or two plate heat exchangers (free-cooling 3 or refrigerant 2). As shown in Fig. 1, the unit has outdoor air humidity (7) and temperature sensors at air intake, temperature sensor at the exhaust. Water temperature sensors (6) are installed at the primary and secondary loops, at the primary loop both for supply and return flow.

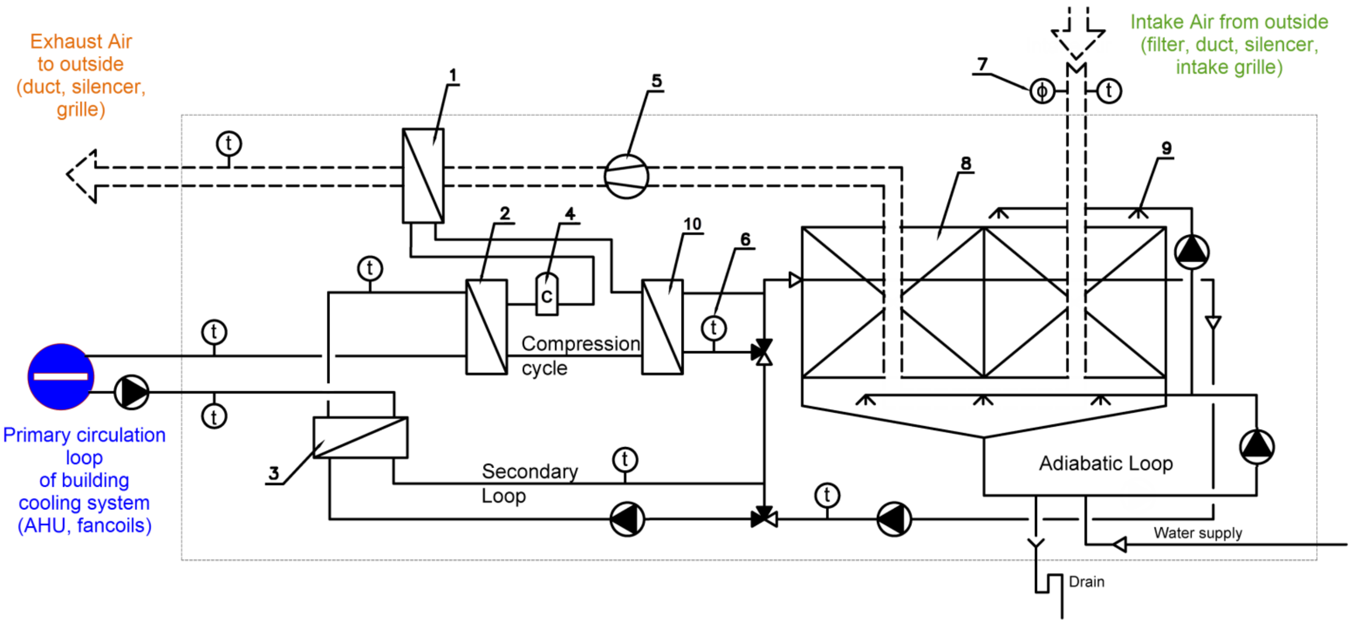


Fig. 3. Scheme of the experimental unit.

Electricity energy electronic meter has precision ± 1 kWh. The main technical data of the chiller is given in Table I.

TABLE I

TECHNICAL PARAMETERS OF THE CHILLER

Cooling output	318 kW
Nominal outside air volume flow	19000 m^3/h
Air static pressure for duct system	300 Pa
Fan motor, electrical power with FC	8.87 kW
Compressor, electrical power	2 x 59.8 kW
Type of refrigerant	R407C
Circulating pumps, electrical power	0.55 / 0.55 / 0.75 / 1.1, all kW
Evaporative water - air heat exchanger design data:	
Secondary loop water side temperature In / Out	41.5 °C / 31.5 °C
Air side temp, rel. humidity Out / In	37.3 °C, 78 % / 28.0 °C, 60 %
Cooling fluid flow, primary loop	54.5 m^3/h
Cooling fluid pressure drop	60,3 kPa
Cooling fluid temperatures	7 °C / 12 °C

B. Data Processing

Using fluid and air parameter sensors described above, electricity consumption, chiller operation stages, fluid temperatures, and outdoor air parameters data have been acquired for the period of eight month. Outdoor air parameters have been acquired also by the building management system data storing server and incidentally, in some cases, slightly (o.a. temperature about +–2 °C) differs from the chiller sensor readings. We have ignored those deviations and took into account only the data collected by sensors installed in the chiller, due to the fact that these data are determinative for the unit automatics. As a measure of the relation of the chiller operating efficiency to the outdoor air moisture content, we took the unit EER and intake air absolute humidity ratio at relatively constant temperatures. Operation data have been recorded every minute, including intake air temperature, relative humidity, In / Out cooling liquid flow in the primary loop, electrical energy meter readings. Saved each minute data have been exported as CSV files, recalculated to each hour average values and analyzed. After export to spreadsheet, the data amounted to more than 380 thousand rows (1 measurement per minute = 1 row). Hourly average values were calculated for analysis. The rows containing one or more major errors were ignored.

C. Calculations

Knowing the altitude and air temperature, saturation humidity ratio *Ws* can be found [7], using (1):

$$W_s = 0.62198 \frac{p_{ws}}{p-p_{ws}}. \tag{1}$$

Where:
W_s – saturation humidity ratio, kg_w/kg_{da}
p_{ws} – saturation pressure, kPa
p – barometric pressure, kPa

The barometric pressure is assumed as function of altitude *Z*, which is 6 m average for the Old Riga:

$$p = 101.325(1 - 2.25577 \cdot 10^{-5} Z)^{5.2559}. \tag{2}$$

The saturated vapour pressure in kPa is calculated using [8] Magnus formula (3):

$$p_{ws}(t) = \alpha \exp\left(\frac{\beta \cdot t}{\lambda + t}\right). \tag{3}$$

Where:
t – air temperature, °C
α = 0.6112, kPa
β = 17.62
λ = 243.12, °C

Using intake air relative humidity data acquired, air moisture content was obtained:

$$x = \varphi W_s. \tag{4}$$

Where:
φ = relative humidity, dimensionless

The formulas above concern intake air psychrometrics. To evaluate the efficiency of the equipment, cooling energy produced by the chiller per minute was calculated using equation (5), [17].

$$Q = g \rho c_{cw} (T_{in} - T_{out}). \tag{5}$$

Where:
Q – cooling output, kW
g – cooling fluid volumetric flow, m^3/s
ρ – cooling fluid density, kg/m^3
c_{cw} – cooling fluid specific heat, kJ/(kg·°C)

Cooling fluid in the system is 35 % ethylene-glycol and water mixture, ρ = 1045 kg/m^3, c_{cw} = 3.585 kJ/(kg·°C).

The chiller's overall energy efficiency ratio (EER), according to energy balance equation will be:

$$\text{EER} = \frac{\text{Cooling power}}{\text{Input power}}. \tag{6}$$

Input power of the chiller for electrical energy consumption is defined in this case by the following consumers:

1. Air side – direct driven fan motor, including frequency controller. Adiabatic loop circulating pumps for water spray nozzles;
2. Compressor cycle – compressor;
3. Control and automation – input/output modules, transformers, control display and controller;
4. Water side, the secondary loop – circulation pumps;
5. Water side, the primary loop (building circuit) – external circulation pump is ignored, because it depends more on the building pipe system, than on the chiller operation efficiency, and is connected to the other electricity consumption metering unit.

The input power was calculated for each hour of the analyzed period, using electricity meter data (every 60th minute value minus every 1st minute value of each hour).

III. Results and Discussion

Electricity and water consumption, chiller operation stages, cooling average temperatures, and outdoor air parameters data have been acquired for the period from March till October, 2012. The period length was chosen according to the cooling demand of the building.

The outdoor air relative humidity and cooling unit EER at OA temperature average hourly values for the studied period are shown in Fig. 4.

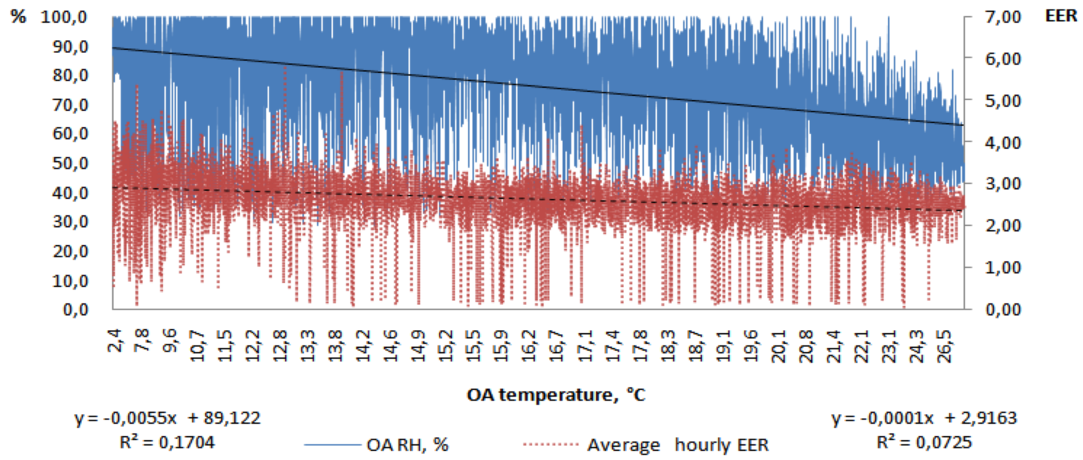


Fig. 4. Outdoor air RH and EER hourly values at OA temperatures for March–October, year 2012

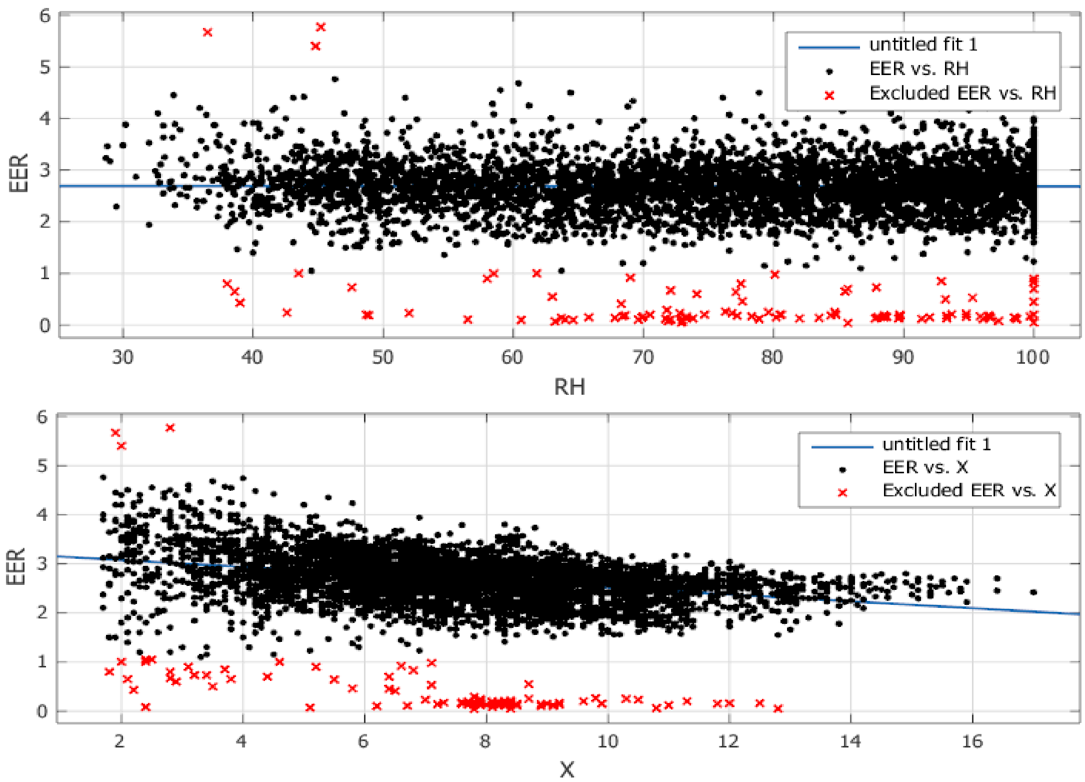


Fig. 5. EER at OA relative humidity, %, and EER at OA moisture content, g/kg, hourly values for March–October, year 2012.

We can see some data errors in the graph shown in Fig. 4, mainly extremely low EER values in the periods with zero cooling demand or during system maintenance. In further calculations extremely low, less than 1, efficiency values were ignored. Fig. 5. showed a stronger correlation between EER-X, rather than EER-RH shown in Fig. 5. To evaluate the correct dependence between chiller EER and outdoor air moisture content, it was decided to use the EER values within the constant air temperatures. To increase the amount of the usable statistical data, we assumed 2 °C temperature value ranges as a constant temperature. In the temperature range from 11 °C to 24 °C 7 graphs were plotted for constant temperature values with step 2 °C. The calculated EER and OA humidity graph at all registered OA temperatures showed that cooling unit EER dependence on outdoor air moisture content is clearly visible, and it is inverse. This dependence is more expressed in the lowest range of cooling temperatures – from 11 °C to 21 °C, which is shown in the group of figures below (Fig. 6).

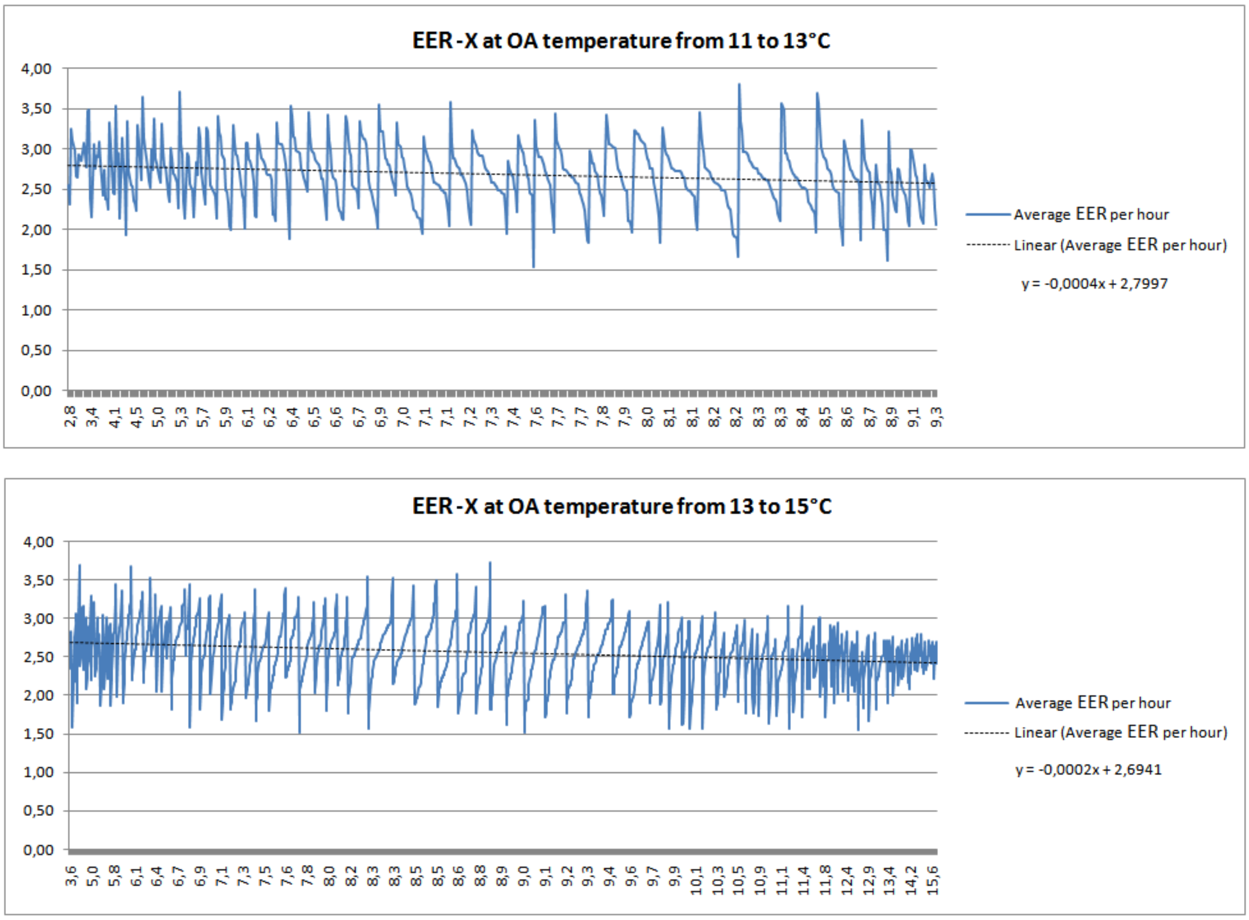
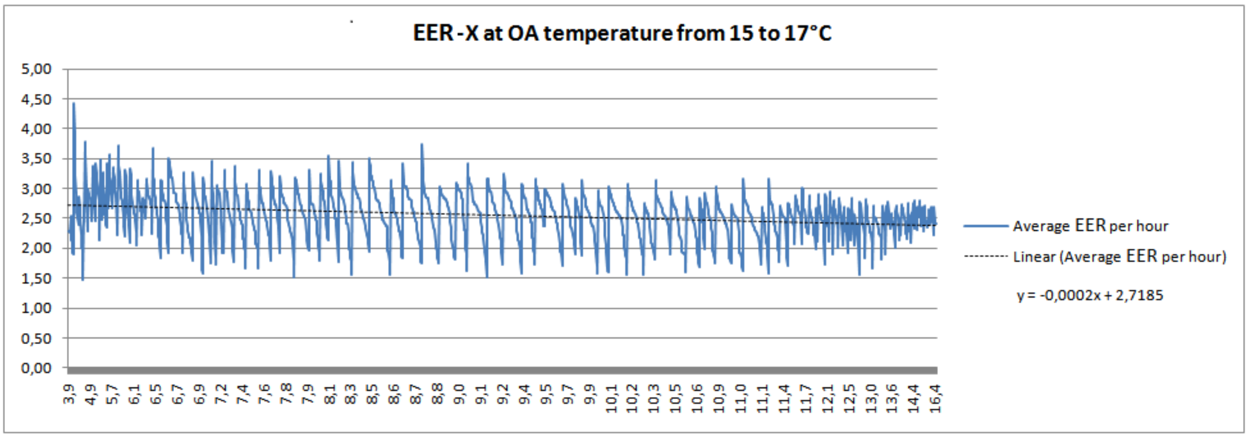


Fig. 6. EER (y) and OA moisture content (x, g/kg) at the temperature range from 11 °C to 13 °C and from 13 °C to 15 °C where ▪▪▪▪ – linear approximation of EER. Shows also linear approximation of EER – X dependence.

The outdoor temperatures from 11 °C to 13 °C were registered during 316 hours and while moisture content increases from 2.8 g/kg to 9.3 g/kg, the chiller EER decreases by 9.8 %. Temperatures in the range from 13 °C to 15 °C were registered during much longer period – 1,434 hours. In this case, moisture increase from 3.6 g/kg to 15.6 g/kg caused EER reduction by 10.0 %. EER X graph at air temperatures from 15 °C to 17 °C is shown in Fig. 7.



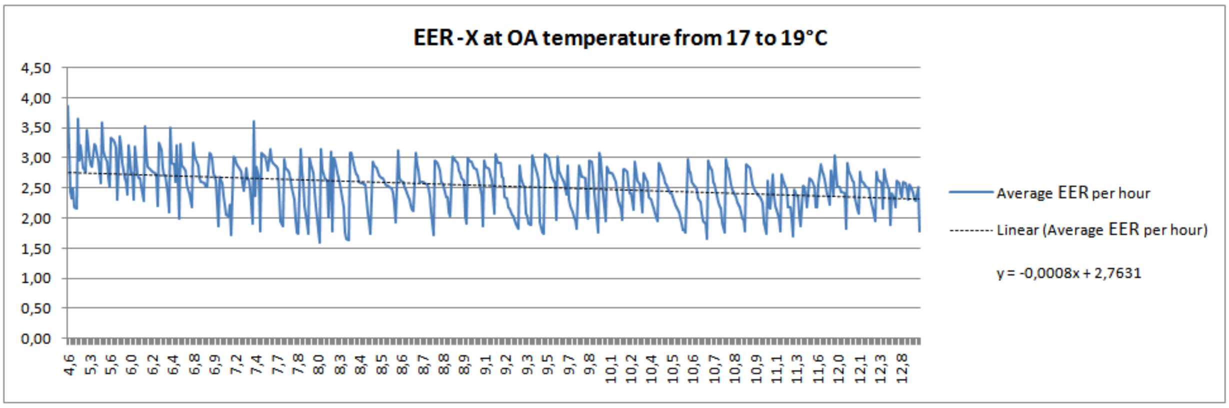


Fig. 7. EER (y) and OA moisture content (x, g/kg) at the temperature range from 15 °C to 17 °C and from 17 °C to 19° C.

The inverse correlation between these two variables is obvious, because absolute humidity changes sharply according to the OA temperature. The higher is OA temperature, the higher is OA moisture, which results in reducing the efficiency of evaporative intake air pre-cooling. In the temperature range 22 °C ± 1 °C, the EER – OA moisture dependence is still clearly visible. In this case, humidity rise from 5.7 g_w/kg_{da} to 14 g_w/kg_{da} causes chiller EER decrease from 2.74 to 2.42, which is equivalent to 11 %. However, chiller efficiency data for operation in outdoor air temperatures 21 °C and higher do not show strong dependence on outdoor air moisture. The graph and approximation of this dependence is shown in Fig. 8.

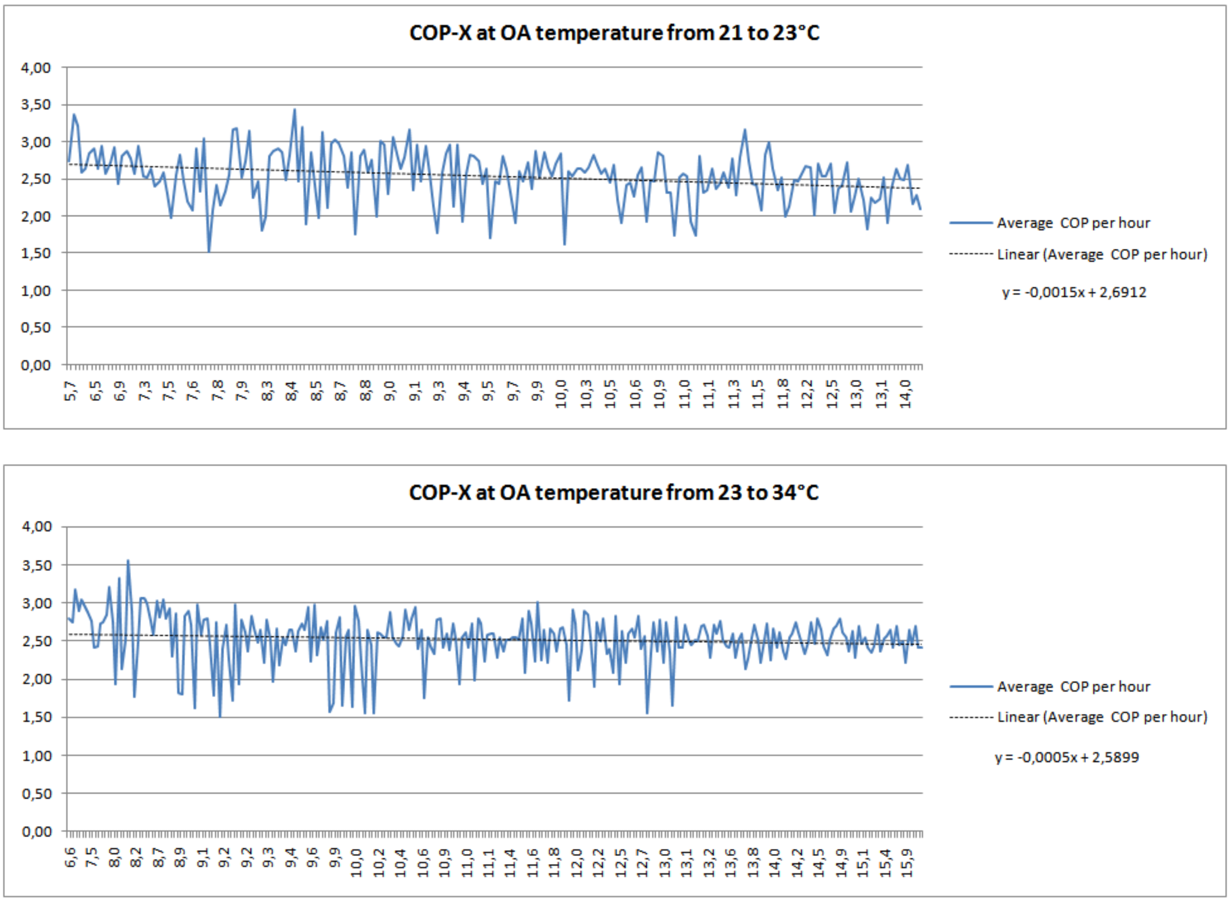


Fig. 8. EER (y) and OA moisture content (x, g/kg) at the temperature range from 21 °C to 23 °C and from 23 °C to 34 °C.

The graph shows that average chiller EER very slightly depends on OA moisture. At a higher range of OA temperatures, from 23 °C to 34 °C, moisture content increase from 7 g/kg to 15 g/kg causes 4 % decrease of EER. According to [2] heat transfer equations, the effectiveness of an air side of heat exchanger is dependent on the air specific heat c_p. The c_p and moisture content are in direct relation. We can conclude that in our case the decrease of refrigerant – air heat exchanger effectiveness caused by low humidity has been compensated by evaporative pre-cooling of intake air.

IV. Conclusions

Electricity consumption, chiller operation stages, cooling fluid temperatures, and outdoor air parameters data have been acquired for the period of eight month, during the year 2012.

The data, recorded for every minute, was processed and recalculated for hourly average values. Data analysis for 7 different constant air temperatures showed that for the studied period of time chiller's EER slightly depends on the outdoor air moisture and this dependence is inverse proportional.

This dependence is much less expressed at the highest registered outdoor air temperatures – from 23 °C to 34 °C. This fact can be explained by the adiabatic compensation effect. In our case the decrease of refrigerant – air heat exchanger effectiveness caused by low humidity is compensated by adiabatic intake air pre-cooling.

The results of the present and related studies are aimed to investigate the possibilities of increasing energy efficiency of air conditioning equipment by using indirect evaporative cooling systems in temperate climate.

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