BUILDING AND RENOVATION

INDIRECT EVAPORATIVE PRE-COOLED COMPRESSOR COOLING SYSTEM PERFORMANCE UNDER VARIOUS OUTDOOR AIR HUMIDITY CONDITIONS

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ABSTRACT

The present study is devoted to efficiency evaluation of a combined indirect evaporative – compressor cooling system under various outdoor air humidity conditions of temperate climate. The investigated system is located in the recently restored historical building, The Art Museum Riga Bourse, which was initially built in the middle of the 19th century. The indirect adiabatic chiller supplies cooled fluid to the conventional cooling system, consisting of ventilation cooling coils and fan-coil units on separated loop. Using the data, acquired by BACnet BMS controllers and experimental data logging system, we have analyzed the cooling plant operation efficiency dependence of outdoor air humidity for a period of four month. The saved each minute data have been exported as CSV files, recalculated to each hour average values and analyzed.

Key words: indirect evaporative, cooling system, historical building

INTRODUCTION

While water side evaporative cooling arrangements are occasionally used, with air–water systems, particularly in more arid climates, the use of the technique falls far short of its potential. This is particularly the case in west European temperate climates where many opportunities to benefit from evaporative cooling techniques are often overlooked (De Saulles, 1996). This situation is attributed by (Field, 1998) to a lack of in-depth knowledge of the energy performance of water side free cooling systems, in terms of the cooling generated per unit of primary energy expended (Costelloe et al., 2002). However, the present engineering tendencies show that due to the development of HVAC system and control equipment, this method of cooling becomes even more attractive for use also in European countries with temperate climate.

The main factors for choosing one or another type of the cooling equipment are climate, cost efficiency, sizes, and availability of external recourses, such as spare heat energy, or the proximity of water sources (Borodinec et. al, 2008).

The existing studies related to indirect evaporative cooling and combined systems efficiency in Northern European regions are focused mostly on high-temperature cooling systems, such as chilled beams and chilled ceilings (Duan et. al, 2012). It is obviously because of better cooling unit efficiency in higher cooling temperature conditions.

Restoration of old buildings is a complex construction process, in 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.

Necessity of harmonious integration of modern HVAC devices in the historical interior also enforces limits to the equipment selection. In those cases the use of high temperature cooling equipment becomes complicated because of aesthetic requirements and need for air dehumidification.

Figure 1. Riga Bourse building after restoration

Outdoor air humidity is a parameter of climate, which affects heat transfer in air heat exchangers, and needs to be taken into account in calculations (Kays et al., 1998). Recent studies have shown, that in case of water – air heat exchangers the relative air humidity level increase from 50 to 90% results in the heat transfer coefficient α growth in 1,68 times (Averkin et al., 2012). It is necessary to clarify, that in the mentioned paper the authors did not focus on the air temperature, which implies that such distinct α changes could be explained not only by changes in the relative humidity, but also outdoor air temperature decrease (adiabatic cooling).

Latvia is located in the moderate climatic zone. Its temperature, moist climate are created by the Atlantic air masses and influenced by the Baltic Sea and the gulf of Riga. 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 (Krumins et al., 2008).

This paper focuses on the investigation of outdoor air humidity impact on the chiller COP. The investigated system is located in the recently restored historical building, the Art Museum Riga Bourse, which was initially built in the middle of the 19th century (Fig.1). To preserve artefacts, the Museum is equipped with climate – control and building management (BMS) systems.

MATERIALS AND METHODS

Cooling system description

The cooling system consists of an indirect evaporative water chiller with integrated compressor, 5 air handling units with cooling / dehumidifying coils and 98 fan-coil units on separated loop. The museum premises, which are used for storing the most valuable exhibits, are equipped also with autonomous air humidifiers to keep a constant air moisture level in winter seasons.

The chiller, which we have used as an experimental unit, is equipped with an air-water heat exchanger (8), which cools secondary loop with adiabatically pre-cooled outdoor air. Outdoor air is driven by a radial fan (5), pre-cooling is provided by water nozzles located in adiabatic loop. Refrigerant – air (1), and refrigerant – water (2, right) heat exchangers utilize heat, produced by compression cycle (4). Thereby, primary loop, which supplies the building, on demand is cooled by one or two heat exchangers (3 or 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 primary and secondary loops, at primary loop both for the supply and return flow.

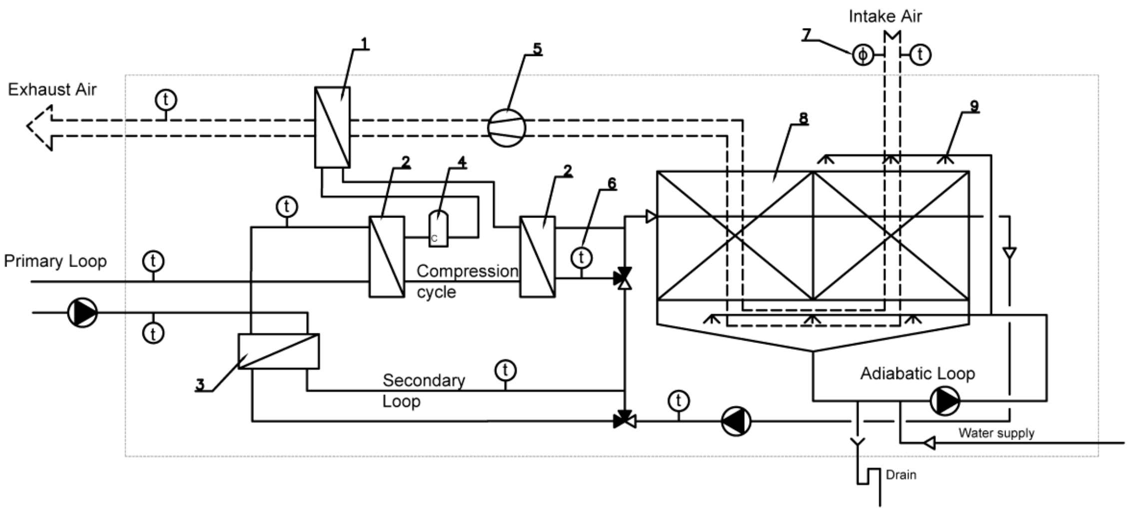

Figure 2. Scheme of the experimental unit

Water supply for "adiabatic" circuit is equipped with an impulse water meter, precision ±1 liter. The electricity electronic meter has precision ±1kW. Both meters are not shown in Fig.2.

Data processing

Using fluid and air parameter sensors, described above, electricity consumption, water consumption, chiller operation stages, fluid temperatures, and outdoor air parameter data have been acquired for the period from August 1 till November 29, 2012. The outdoor air parameters have been acquired also by the building air handling unit automatics and incidentally in some cases slightly (o.a. temperature about +-2°C) differ from the chiller sensor readings. We have ignored these deviations and taken into account only the data collected by the sensors, installed in the chiller, due to the fact that these data are determinative for the unit automatics. As a measure of the chiller operating efficiency relation to the outdoor air moisture content, we took the unit coefficient of performance (COP) and intake air absolute humidity ratio at relatively constant temperatures. The data storing server recorded the operation data every minute, including intake air temperature, relative humidity, In / Out cooling liquid flow in primary loop, energy and water meter readings. After export to spreadsheet, the data amount had more than 163k rows (1 measurement per minute = 1 row). Hour average values were calculated for analysis. Rows, containing one or more rough errors, were ignored.

Calculations

Knowing the altitude and air temperature, saturation humidity ratio W_s can be found (ASHRAE, 2001), using equation:

$$W_s = 0.62198 \frac{p_{ws}}{p - p_{ws}}. \quad (1)$$

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 the function of altitude Z, which is 6m average for Old Riga:

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

The saturated vapour pressure in kPa is calculated, using (Sensirion, 2009) Magnus formula:

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

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

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

$$x = \varphi W_s. \quad (4)$$

Where:
φ = relative humidity, dimensionless.

The formulas above concern the intake air psychrometrics. To evaluate the efficiency of the equipment, we have calculated the cooling energy produced by the chiller per minute using equation (5), (Krieder, 2001).

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

Where:
Q = cooling output, kW
g = cooling fluid volumetric flow, m^3/s
ρ = cooling fluid density, kg/m3
c_{cw} = cooling fluid specific heat, kJ/(kg*°C)

The 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 COP according to the energy balance equation will be:

$$COP = \frac{\text{Cooling power}}{\text{Input power}}. \quad (6)$$

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).

RESULTS AND DISCUSSION

The outdoor air (OA) temperature data analysis has been performed for 2720 hours, and it showed that the most common average hourly temperatures occurred in the range from +14 to +16°C, Fig. 3.

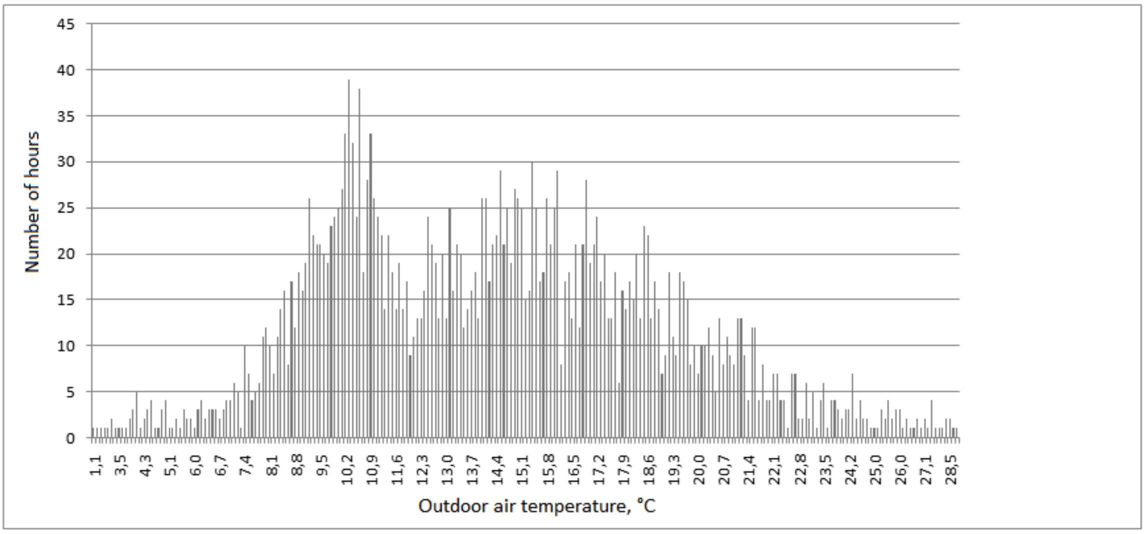

Figure 3. Average hourly temperatures – hours

Average hourly temperatures in the range +15±1°C registered in 477 hours during the analyzed period. We assumed this range as constant air temperature.

RESULTS AND DISCUSSION
The calculated COP and OA humidity graph at all registered OA temperatures showed that the cooling unit COP dependence of the outdoor air moisture content is clearly visible, and it is inverse (Fig. 4). It

is obvious, because absolute humidity changes sharply according to the OA temperature. The higher the OA temperature, the higher the OA moisture, which results in reducing the efficiency of evaporative intake air pre-cooling. Linear approximation for COP - humidity is also shown in Fig. 4. When defining the temperature diapason 15±1°C, the COP - OA moisture dependence still persists, not so expressed, but still. In this case, humidity rise from 4,2 to 14 g_w/kg_{da} causes the chiller COP decrease from 2.74 to 2.56, which is equivalent to 6.6%. The graph and approximation of this dependence are shown in Fig. 5.
The dependence shows that average chiller COP is less than 3, and it is very slightly dependent on OA moisture. According to (Kays et al., 1998) heat transfer equations, the effectiveness of the air side of the 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 refrigerant – air heat exchanger effectiveness decreasing, caused by low humidity, is compensating by adiabatic intake air pre-cooling.

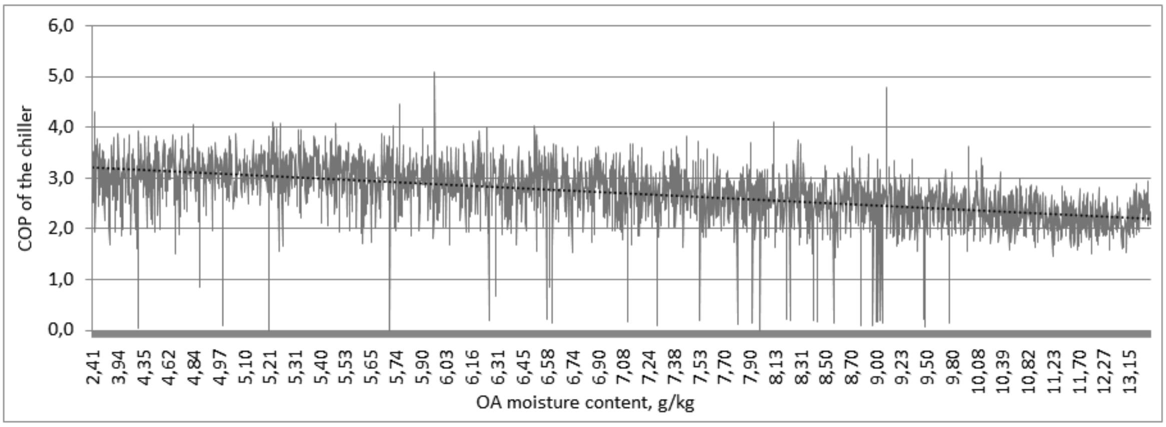

Figure 4. COP and OA moisture content at all registered temperatures, where ▪▪▪▪ - linear approximation of COP

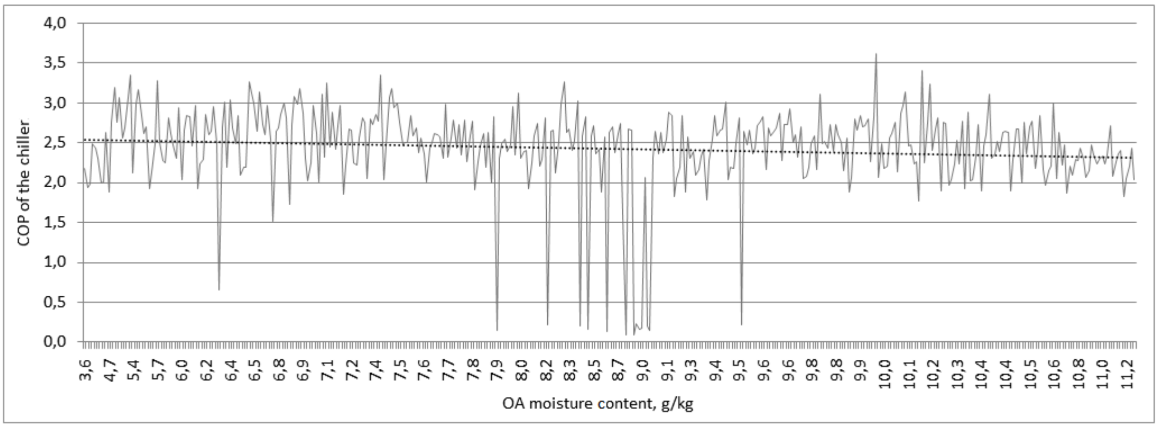

Figure 5. COP and OA moisture content at temperatures 15±1°C

CONCLUSIONS

Electricity consumption, water consumption, chiller operation stages, cooling average temperatures, and outdoor air parameter data have been acquired for the period of 4 months, during the year 2012 cooling.
The data, recorded for every minute, were processed, and recalculated for each hour average. The data analysis at constant temperature 15±1°C showed that for the studied period of time the chiller COP very slightly depends on the outdoor air moisture, and this dependence is inverse proportional.
The slight dependence can be explained by the adiabatic compensation effect. In our case refrigerant – air heat exchanger effectiveness decreasing, caused by low humidity, is compensated by adiabatic intake air pre-cooling.
The results of the present and related studies will clarify the efficiency of indirect adiabatic cooling systems in temperate climates.

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