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Influence of outdoor air conditions on the air source heat pumps performance

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Abstract

The purpose of the present work is the investigation of the effect of the outdoor air temperature and relative humidity on the performance of an air heat pump, when the reverse-cycle defrosting is considered. The frost formation process has been analyzed by developing a simplified model which relates the number of defrost cycles to the outdoor air conditions. Moreover the energy consumption due to the defrosting has been taken into account in the evaluation of the heat pump performance. The results, carried out for many Italian sites, point out that the outdoor air conditions play an important role in determining the amount of defrost cycles; however the frost formation is mainly affected by the relative humidity. The analysis highlights also that the defrosting contribution on the heat pump performance is not negligible when the heat pump that operates in wet weather, although cold; in these conditions the hourly COP may be reduced by up to 20%. However, this effect becomes less relevant, but not negligible, when the seasonal heat pump performance is evaluated; the maximum decrease of *SCOP*, observed for the all analyzed cases, is less than 13%.

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Keywords: Air source heat pumps; defrosting effect; defrosting cycles; *SCOP*.

1. Introduction

The reduction of buildings energy consumption and the use of the renewable energies are necessary measures for decreasing the European greenhouse gas emissions, as indicated in the last EPBD (i.e. Directive 2010/31/EU). In

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order to improve the efficiency of the building heating and cooling systems, different solutions can be adopted. The use of the heat pumps represent certainly an important measure to achieve energy saving.

Heat pump performance is significantly affected by the operating conditions, such as the features of the heat sources, the heating demand and the heat pump control system. Since the outdoor air temperature is extremely variable over the heating period the Coefficient Of Performance (*COP*) of the air heat pumps cannot be evaluated by considering the steady state monthly values of the outdoor air temperature but by using the so-called bin method, based on an evaluation of the cumulative frequency of the outdoor air temperature in a fixed locality.

However the energy performance of air heat pumps is affected not only by the outdoor air temperature, but by the air humidity as well. When the air moisture content is high and temperature is lower than 6°C, frost formation on the surface of the airside heat exchanger may occur and consequently the heat pump performance can be reduced or even the system may stop.

The frost formation is a complex process. In literature several works which investigate the mechanism of frost formation in heat exchangers characterized by simple geometry (i.e. cylinders, flat plates and parallel plates), have been presented [1–10]. Moreover the frost formation process in heat exchangers characterized by more complex geometries, has been investigated by many authors [11–21]. However they analyze specific geometries, therefore it becomes difficult to use these results for general purpose investigation.

The frost needs to be removed periodically to improve the efficiency of operation. This can be carried out in different ways: by resorting to the electric resistance or by inverting the cycle or by stopping the compressor or by using hot gas. When the heat pump can operate in reversible mode the defrost is generally obtained by inverting the cycle. The energy consumption due to the defrost cycles has to be considered in calculating the heat pump performance but the calculation methods proposed by the European standards ignore this effect.

The present paper intends to numerically investigate the effect of the outdoor air temperature and relative humidity on the frost production and the impact of defrosting on the performance of an air heat pump (*COP* and Seasonal *COP*), in the case of reverse-cycle defrosting.

For the purpose of this study the amount of defrost cycles per season has been evaluated by using a simplified model which enables to estimate the needed defrost cycles depending on the outdoor air conditions. The numerical simulations have been carried out by considering many Italian towns.

Nomenclature

c_p	Specific heat at constant pressure ($\text{kJ kg}^{-1} \text{K}^{-1}$)
<i>COP</i>	Coefficient of Performance
<i>E</i>	Energy (kJ)
<i>h</i>	Specific enthalpy (kJ kg_a^{-1})
\dot{m}	Dry air mass flow rate (kg s^{-1})
<i>p</i>	Pressure (Pa)
<i>P</i>	Building heating demand (kW)
\dot{Q}	Heat flow rate, thermal power (kW)
<i>RH</i>	Relative humidity
<i>SCOP</i>	Seasonal Coefficient of Performance
<i>t</i>	Temperature (°C)
<i>x</i>	Moisture content ($\text{kg}_v \text{kg}_a^{-1}$)
\dot{W}	Electric power (kW)
<i>z</i>	Number of bin hours
Greek symbols	
α	Performance reduction coefficient
λ	Heat of fusion (kJ kg^{-1})
$\Delta\tau$	Time interval (s)
Subscript	
<i>a</i>	Air
<i>adj</i>	Adjusted

<i>c</i>	Cooling
<i>elbu</i>	Electric backup heater
<i>ev</i>	Evaporation
<i>dp</i>	Dew point
<i>fr</i>	Frost
<i>in</i>	Inlet
<i>out</i>	Outlet
<i>sat</i>	Saturation
<i>sb</i>	Stand-by
<i>th</i>	Thermal
<i>w</i>	Wall

2. Problem statement

2.1. Frost production

In order to explain the frost formation process, let us start from the analysis of the heat transfer in the external coil. If one considers a specific *i*-th bin air temperature, the rate of heat captured from the cold heat source (outdoor air) by means of the external evaporator can be estimated by applying the energy conservation principle to the heat pump:

$$\dot{Q}_{ev,i} = \left(1 - \frac{1}{COP_i}\right) \dot{Q}_{th,i} \tag{1}$$

where $\dot{Q}_{th,i}$ is thermal output and COP_i denotes the Coefficient of Performance of the heat pump. For any bin air temperature, the value of the rate of energy transferred to the building and of COP is known by using the manufacturer data.

The specific enthalpy of the air at the outlet section of the evaporator can be estimated by the balance equation:

$$h_{out,i} = h_{in,i} - \frac{\dot{Q}_{ev,i}}{\dot{m}_a} \tag{2}$$

being \dot{m}_a the dry air mass flow rate through the external side of the evaporator (this value can be evaluated by the manufacturer data) and $h_{in,i}$ the specific enthalpy of the air at the inlet of the external coil which can be calculated as a function of the air temperature (*t*) and of the moisture content (*x*) by solving the classical equations describing the physical properties of moist air (ASHRAE [22]):

$$\begin{cases} h = 1,006t + x(2501 + 1.84t) \\ x = 0.622 \left(\frac{RH p_{sat}(t)}{p_{tot} - RH p_{sat}(t)} \right) \end{cases} \tag{3}$$

where p_{tot} is the total pressure of the moist air which depends on the altitude, p_{sat} is saturation pressure and RH denotes the air relative humidity.

For each *i*-th bin the value of the outdoor air enthalpy ($h_{in,i}$) is known if the values of the outdoor temperature and of the relative humidity are known.

In this way, by using equation (3) and equation (2) the value of the air enthalpy at the exit of the evaporator can be calculated as a function of the *i*-th bin temperature and of the outdoor air relative humidity. The value of the air

enthalpy obtained by equation (2) has to be compared with the dew point value ($h_{dp,i}$) which can be obtained by equation (3) using the value of the moisture content of the outdoor air ($x_{in,i}$) and the value $RH=1$:

$$\begin{cases} h_{out,i} > h_{dp,i} & x_{out,i} = x_{in,i} \quad (\text{no frost production}) \\ h_{out,i} < h_{dp,i} & x_{out,i} < x_{in,i} \quad (\text{frost production}) \end{cases} \quad (4)$$

If $h_{out,i} < h_{dp,i}$ the value of the moisture content of the outdoor air at the exit of the evaporator ($x_{out,i}$) is estimated by solving the system of equation (3), with $RH=1$ and $h=h_{out,i}$.

Once the decrease of the water vapor content in the air through the evaporator is known, the frost accumulation rate on the cold surface of the external coil can be calculated by:

$$\dot{m}_{fr,i} = \dot{m}_a (x_{in,i} - x_{out,i}) \quad (5)$$

However the choice of the RH value of the outside air for any bin is not a simple task because there is not a univocal link between the RH values and the air temperature.

2.2. Defrosting cycles

Experimental results showed that if the outdoor coil temperature is above -3°C there is practically no frost accumulation on the outdoor coil when the ambient temperature is in the range of 0 to 6°C , which is the range at which frost is most likely to accumulate [18]. When the ambient temperature is outside the range of 0 - 6°C the defrosting cycles are generally not necessary because above 6°C the coil temperature will be warm enough to prevent frost formation on the evaporator, and below 0°C the moisture content of the air will be too low for a heavy frost accumulation on the coil.

When the frost is built up on the external surface of the evaporator the refrigerant pressure during the evaporation process tends to decrease and this accelerates the frost formation because the surface temperature of the heat exchanger decreases. In order to avoid this effect, when the surface temperature of the external coil (or the refrigerant pressure) becomes lower than a setup value, the heat pump activates a defrost cycle. If the heat pump is reversible the most used defrosting method is the inversion of the thermodynamic cycle; this action temporarily warms up the outdoor coil and melts the frost from the coil. In this defrost cycle, the outdoor fan is prevented from turning on when the heat pump switches over, and the temperature rise of the outdoor coil is accelerated and increased. A heat pump unit will defrost regularly when frost conditions occur. The defrost cycle should be long enough to melt the frost, and short enough to be energy-efficient. The energy required to melt the frost produced during one hour is:

$$E_{fr,i} = \dot{m}_{fr,i} (\lambda_{fr} + c_{p,fr} (0 - t_{w,i})) \Delta\tau \quad (6)$$

being λ_{fr} and $c_{p,fr}$ the heat of fusion and the specific heat at constant pressure of the frost respectively. $\Delta\tau$ indicates the seconds in one bin (1 hour) and $t_{w,i}$ denotes the wall temperature which depends on the outdoor air temperature, external coil features, refrigerant temperature and heat transfer coefficients.

Once the energy required to remove the frost is known, the minimum defrosting time can be evaluated by considering the ratio of the energy required to melt the frost to the cooling capacity of the heat pump:

$$\Delta\tau_{fr,min,i} = \frac{E_{fr,i}}{|\dot{Q}_{c,i}|} \quad (7)$$

In general, $\Delta\tau_{fr,min}$ is lower than 1 hour and, depending on the ambient temperature and humidity conditions, it can vary from 10 s up to 2-5 minutes. Actually the defrosting cycle duration is longer than $\Delta\tau_{fr,min}$. Typical

defrosting cycles are represented in Figure 1; before activating the inversion, the heat pump is switched off for a period equal to $\Delta\tau_{sb}$ (i.e. 1 minute).

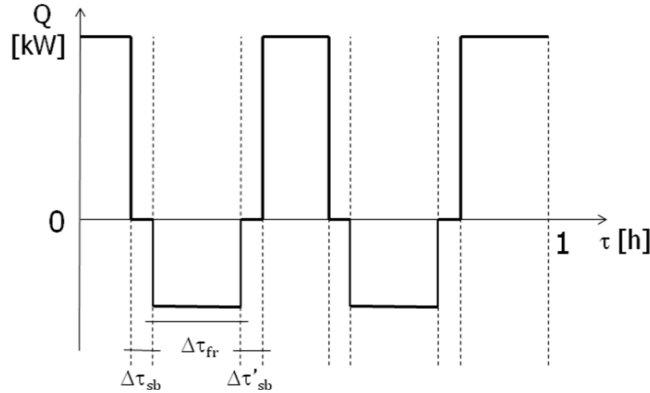


Fig. 1. Typical defrosting cycles and stand-by periods for an air heat pump ($N=2$).

During the stand-by period the defrost relays turn on the compressor, switch the reversing valve of the heat pump, turn on the interior electric heating element, and stop the fan at the outdoor coil from spinning and finally the heat pump operates following the cooling cycle. The unit remains in the defrost cycle (or cooling cycle) until the thermostat on the bottom of the outdoor coil senses that the outdoor coil temperature has reached the desired value (typically about 10-12°C). At this temperature, the outdoor coil should be free of frost. A typical defrost cycle might run from 30 seconds to a few minutes ($\Delta\tau_{fr}$) and it is generally fixed by the heat pump manufacturer. After that period, the frost thermostat opens the circuit then the defrost cycle stops, the internal heater turns off, the valve reverses, and the unit returns to the heating cycle after a stand-by period equal to $\Delta\tau'_{sb}$. (i.e. 30 seconds). The defrost cycles are repeated regularly at timed intervals during an hour.

Therefore the duration of the whole defrosting period for each hour can be evaluated by:

$$\Delta\tau_{fr,tot} = N(\Delta\tau_{sb} + \Delta\tau_{fr} + \Delta\tau'_{sb}) \tag{8}$$

where the needed number of defrosting cycles per hour N can be estimated as follows:

$$N = \frac{\max(\Delta\tau_{fr,min,i})}{\Delta\tau_{fr}} \tag{9}$$

Generally $N=1$ can be sufficient if the timer of the single defrosting cycle $\Delta\tau_{fr}$ is set as equal to the maximum value of $\Delta\tau_{fr,min}$ during the winter season; in this case, the defrosting cycle will be activated each 60 minutes.

2.3. Heat pump performance

When the frost is removed by inverting the cycle, the defrost cycles effect on heat pump performance is twofold: the energy supplied to the building decreases and the building heating demand increases due to the energy carried off by the heat pump in the cooling mode. At the same time, the electrical consumption increases due to the energy spent to melt the frost and to feed the electric-resistance heater equipped with the heat pump which allows to match the required heating demand.

The hourly overall electrical consumption can be computed with:

$$W_{adj,i} = \dot{W}_i(\Delta\tau - \Delta\tau_{fr,tot}) + \dot{W}_{c,i}\Delta\tau_{fr} + \dot{W}_{elbu}\Delta\tau_{fr} + (1 + \alpha)\dot{W}_i(\Delta\tau_{sb} + \Delta\tau'_{sb}) \tag{10}$$

where \dot{W}_h and \dot{W}_c denote the electric power supplied to the heat pump when it operates in heating and in cooling mode, respectively and \dot{W}_{elbu} is the required capacity of the internal electric backup heater which operates during the inversion of the cycle. The last term on the right side of equation (10) takes into account, by means of the α coefficient, the heat pump performance reduction due to the intermittent operation.

By considering the thermal energy transferred to the heat distribution fluid and the overall electrical consumption for each i -th bin temperature of the heating season (n is the number of bins having a number of hours larger than 0) the seasonal performance of the heat pump can be calculated:

$$SCOP = \frac{\sum_{i=1}^n z_i \dot{Q}_{th,i} \Delta \tau}{\sum_{i=1}^n z_i W_{adj,i}} \quad (11)$$

3. Results

The model defined above was applied to 6 northern Italian towns and 4 central and southern Italian sites, characterized by different weather conditions. To calculate the bin distribution for each locality during the whole heating season, the procedure described by the UNI TS 11300/4 Italian standard was followed. The frost accumulation rate was estimated by assuming a monthly average value of the outdoor air relative humidity (RH) derived from the UNI 10349 Italian standard.

The heat pump model chosen for the study (Galletti MCA 10H) is an air-to-water electrically-driven heat pump which can reach the thermal output of 11.13 kW and the water output temperature can range from 30 °C to 50 °C. The unit is equipped with two fans of axial type which move air at a volumetric flow rate of 5580 m³/h. The compressor is of variable speed operation: the thermal output of the heat pump can range from 25% to 100% of the maximum heat power, to match the building heating demand. The heat pump performance for each bin temperature was evaluated according to the UNI TS 11300/4 Italian standard; the trend of thermal output is shown in Figure 2.

The results here presented were carried out considering a building whose the overall heat transfer coefficient (transmission + ventilation) is equal to 697.14 W/K. For the considered building the heating demand (P) becomes zero when the outdoor air temperature is equal to 16°C, accounting for the effects due to the internal and solar heat gains, as shown in Figure 2. The bivalent temperature (i.e. the outdoor temperature point at which the heat pump is declared to have a heating capacity able to meet 100% of the building heating demand) was 2°C, as shown in Figure 2 and the cut-off temperature of the ASHP was fixed at -5°C.

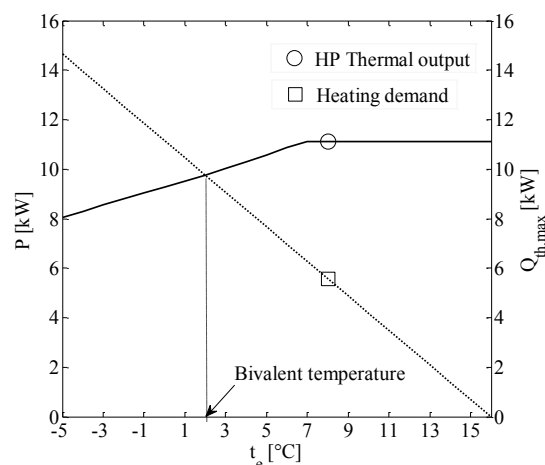


Fig. 2. Thermal output of the heat pump and building heating demand as a function of the external air temperature.

For the considered heat pump the stand-by period ($\Delta\tau_{sb}+\Delta\tau'_{sb}$) is set to 2 minutes and the performance reduction due to the intermittent operation (α) is equal to 2%.

For sake of brevity, only the most interesting cases are presented. In Figure 3 the monthly average values of *COP* for two Italian Northern sites, namely Bologna and Bolzano, is presented. The maximum and minimum reduction observed for Bologna was equal to 16.47% and 5.36%, respectively; the monthly *COP* reduction ranges between 3.41% and 0.19% if the heat pump operates in Bolzano (Figure 3b). The difference among the values obtained for Bologna and Bolzano highlights that the heat pump performance is more affected by the air relative humidity than the air temperature; the *RH* value ranges from 67% to 90% and from 54% to 74% for Bologna and Bolzano, respectively. On the contrary, Bologna is generally less cold than Bolzano (i.e. the minimum value of the outdoor air temperature is equal to -4°C and -14°C, respectively).

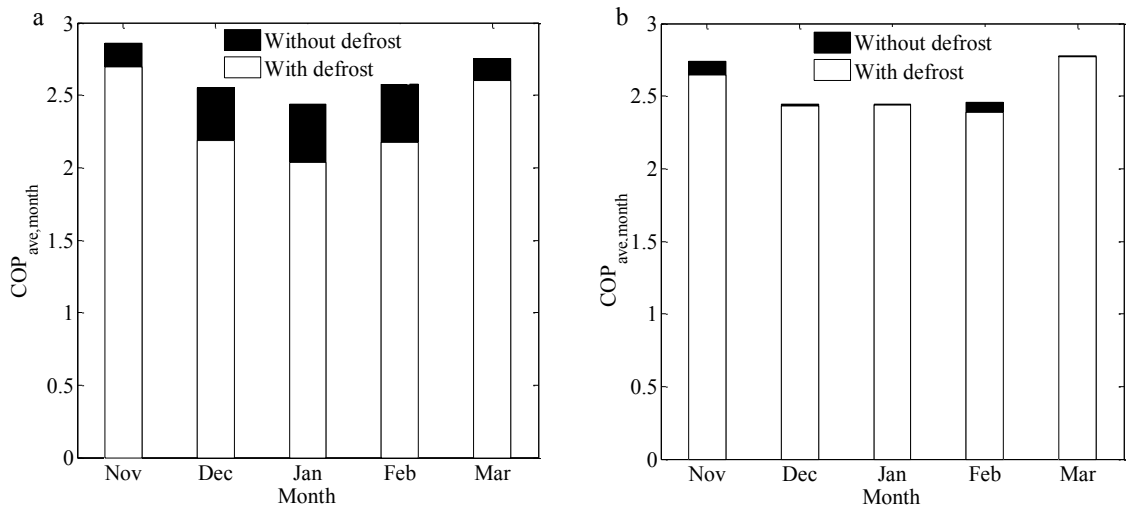


Fig. 3. Monthly average value of the COP: a) Bologna; b) Bolzano.

The needed number of defrosting cycles for the whole heating season are depicted in Figure 4, for the most interesting localities. The analysis points out that the number of defrosting cycles varies strongly by moving from Northern to Southern towns, ranging from 0 to 1400 cycles per season. This effect is due to the amount of hours during the winter season in which the outdoor temperature ranges between 0°C and 5°C ($z_{0.5}$) and to the *RH* value. Table 1 shows the values of $z_{0.5}$ and the minimum and maximum *RH* value for the localities considered in Figures 4 and 5.

Table 1. Weather data of the selected sites.

Town	$z_{0.5}$ (h)	RH (%)
Bologna	1516	67-90
Bolzano	935	54-74
Milan	1697	78-90
Turin	1605	54-86
Venice	1310	69-83
Verona	1453	63-86
Florence	996	72-88
Rome	448	67-82
Bari	406	67-78
Naples	161	65-68

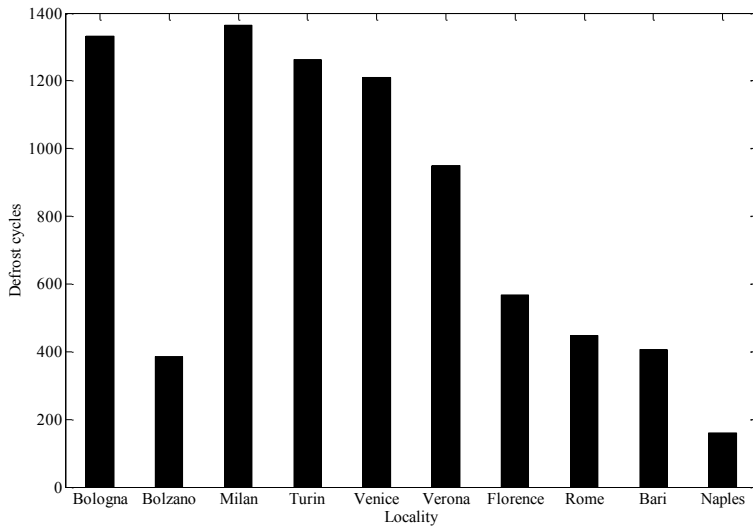


Fig. 4. Number of defrosting cycles per heating season.

The data shown in Figure 4 confirm that towns characterized by larger values of the relative humidity need more defrosting cycles. Figure 5 shows the comparison between the *SCOP* evaluated by taking into account the defrosting effect and the theoretical value of the *SCOP* obtained by ignoring the defrost of the external coil. The Seasonal *COP* reduction ranges between 1.51% (Bolzano) and 12.67% (Bologna).

By observing Figure 5 it can be noticed that in many cases in Italy, if the defrosting effect is ignored, as suggested by the calculation methods proposed by the European standard (EN 14825), the real energy consumption of the ASHP can be underestimated and hence the value of the *SCOP* overestimated. As discussed above, this effect becomes less relevant when the heat pumps operate in sites characterized by low values of the air relative humidity ($RH < 80\%$) and by outdoor air temperature that is out of the range 0-5°C for many hours. For the other localities, it can be recommended to account for the defrosting effect in order to properly evaluate the energy consumption of the air heat pump.

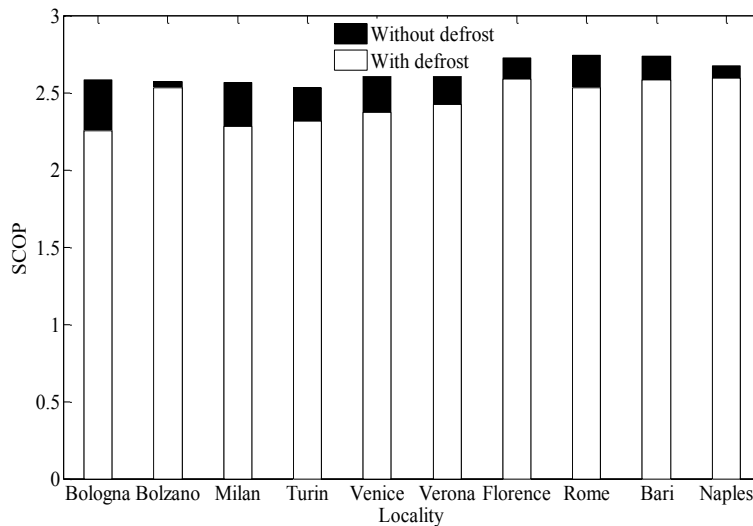


Fig. 5. Defrosting influence on the seasonal heat pump performance (*SCOP*).

Since the RH value plays a leading role in determining the frost formation, further numerical simulations, by using the hourly weather data provided by Meteonorm database [23], were performed. More specifically three different cases were analyzed: in the case 1, a monthly average value of the outdoor air RH was assumed for each air bin temperature of the month; in the case 2, a bin average value of the outdoor air RH was used for each bin of the month and in the case 3 the hourly data of RH were considered (i.e. hourly simulation).

The results of this further analysis point out that the effect of defrost can be overestimated if a monthly average value of RH is assumed; on the contrary by considering a bin average value of the RH the reduction of the heat pump performance may be underestimated. Obviously the hourly simulation (case 3) constitute the reference case, since hourly data for both temperature and RH of the outdoor air, were considered.

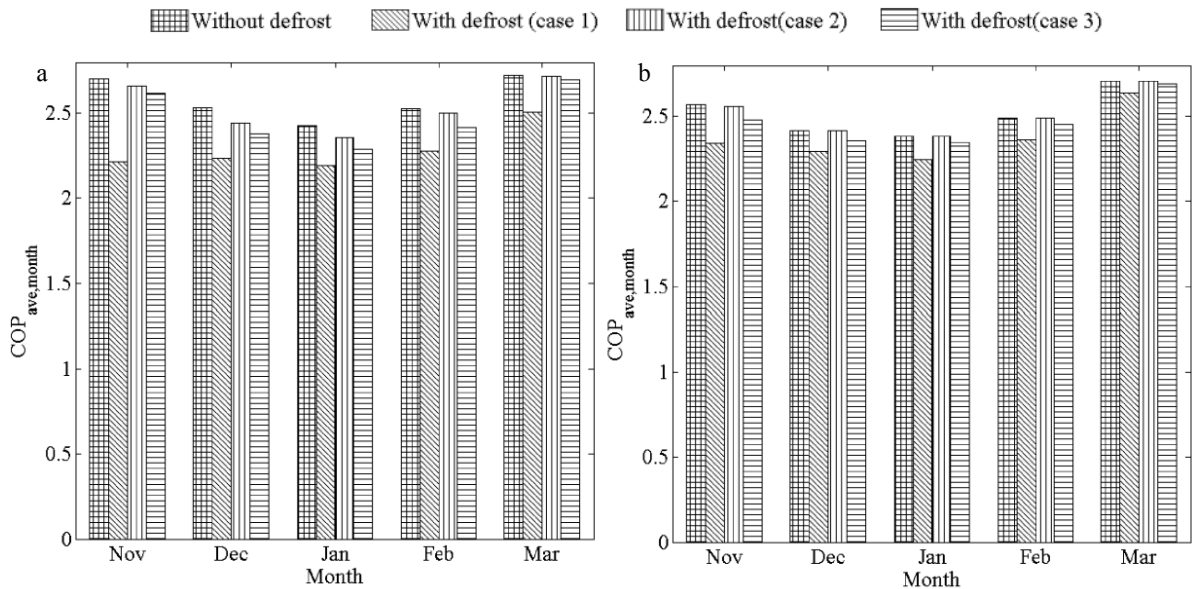


Fig. 6. Influence of RH value on the monthly average value of the COP: a) Bologna; b) Bolzano.

Figure 6 shows the comparison between the monthly average values of COP calculated by considering the above mentioned cases. The results depicted in Figure 6 were obtained by considering the weather data of Bologna and Bolzano, respectively. The same trend can be observed for other towns. The maximum $COP_{ave,month}$ reduction is about 17% (case 1), 4% (case 2) and 6% (case 3), for Bologna; if the heat pump operates in Bolzano, its performance can be reduced by about 9%, 1% and 4%.

However by considering the whole heating season the seasonal performance of the heat pump ($SCOP$) can be reduced by about 11% (case 1), 2% (case 2) and 5% (case 3), for Bologna; the $SCOP$ reduction can be less than 6%, 1% and 2%, respectively, if the heat pump operates in Bolzano. These results confirm that the choice of the RH value is very crucial in order to properly evaluate the frost accumulation rate on the cold surface of the external coil and consequently the effect of the defrost on the heat pump performance.

4. Conclusions

The influence of the outdoor air temperature and relative humidity on the energy performance of the air source heat pumps has been numerically investigated. Many Italian sites characterized by different weather data have been analyzed; temperature and relative humidity values within the range -5°C - 5°C and 50% - 90%, respectively, have been considered. For the purpose of the here presented study the frost formation process has been analyzed by developing a simplified model which enables to correlate the frost production and the external air conditions.

The numerical analysis points out that both temperature and relative humidity of the outdoor air affects the frost formation process; however the relative humidity plays a leading role in determining the amount of frost that can be accumulated. The effect of defrosting process has been also investigated, when the defrost is obtained by inverting the cycle of the refrigerant. The numerical results highlight that the monthly average of the *COP* can be reduced by up to 17% when the heat pump operates in sites in which the values of air relative humidity are very high ($RH > 80\%$) and the outdoor air temperature is within the range $0\text{--}6^\circ\text{C}$ during the winter for many hours. Therefore in these localities the role of the defrosting cannot be ignored for an accurate evaluation of the real energy consumption of an air source heat pump. By considering the whole heating season the *SCOP* reduction may range between 1.51% and 12.67%, for the considered localities. Finally, further investigations on the role of the air relative humidity in the frost formation process have been performed, by using hourly data derived from the Meteonorm database. This latter analysis highlights that depending on the chosen value of the *RH* for each air bin temperature, the heat pump performance can be overestimated or underestimated. Moreover the hourly simulation points out that the *COP* can be reduced by about 20% in wet sites.

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