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Heat-pump dehumidifier as an efficient device to prevent condensation in horticultural greenhouses

Etienne Chantoiseau*; Christophe Migeon; Gérard Chasseriaux; Pierre Emmanuel Bournet

Agrocampus Ouest, UP EPHor Environmental Physics and Horticulture Research Unit, F-49045 Angers, France.

* Corresponding Author (etienne.chantoiseau@agrocampus-ouest.fr)

Corresponding author contact information :

Etienne Chantoiseau

E-mail: etienne.chantoiseau@agrocampus-ouest.fr

Research Unit EPHor

Agrocampus Ouest, centre d'Angers,

2, rue Le Nôtre

49045 Angers Cedex 01

France

Highlight:

- A heat pump dehumidifier was designed to avoid condensation on ornamental crops
- A 4 W m^{-2} power consumption at the compressor sufficed to prevent condensation
- Heat pump significantly contributed to greenhouse heating, reducing energy costs
- Total operational cost was lower than estimated for ventilation with an exchanger
- Device profitability is discussed for different countries according to energy cost

26 **Abstract**

27 Abstract: In the context of increasing energy costs, alternative methods to the energy consuming venting-heating
28 method must be considered for greenhouse dehumidification. In this paper the performance of a heat pump used
29 as a dehumidifier is investigated. Contrary to the classical control aiming at maintaining the greenhouse air at a
30 relative humidity set point, the considered device is designed as a preventive tool to avoid condensation on the
31 crop and limit the energy consumption. The experimental set up was run during winter inside a 2350 m² plastic
32 greenhouse in the West of France for a set temperature of 16°C. During the experiment, no condensation
33 occurred on the plants with a mean condensation rate of 12 W m⁻² and a mean electrical power of 7.62 kW, for
34 an overall efficiency of 4.9. Moreover, the energy retrieved by vapor condensation was given back to the
35 greenhouse as sensible heat, contributing to the total heating of the greenhouse. While dehumidifying the
36 greenhouse air, the device reduces, or may even rule out the gas consumption. The total energy consumption of
37 the heat pump during the season was compared to simulated values for venting-heating dehumidification, with or
38 without an exchanger. The heat pump dehumidifier was shown to be 6 to 8.5 times less energy consuming than
39 the former and 3-8 than the latter, depending on the exterior climate. Using the energy cost of several significant
40 countries, a preliminary operative cost study was conducted and showed that the heat pump can be competitive
41 as a dehumidification alternative.

42

43 **Keywords**

44 Climate Control; Dehumidification; Energy Saving; Evapotranspiration; Heat exchanger.

45

46 Nomenclature

47 Latin

48	C_p	: Specific heat [$J \cdot kg^{-1} \cdot K^{-1}$]
49	e_o	: Global efficiency of the dehumidifying apparatus [-]
50	e_T	: Compressor efficiency of the dehumidifying heat pump [-]
51	ET	: Real evapotranspiration [$W \cdot m^{-2}$]
52	H	: Air enthalpy [$kJ \cdot kg^{-1}_{dry \ air}$]
53	m_{cond}	: Condensed water mass on a time step [kg]
54	m_{elim}	: Ventilated water mass on a time step [kg]
55	\dot{m}_w	: Flow rate in the low temperature heating system [$kg \cdot s^{-1}$]
56	P_{atm}	: Atmospheric pressure (101325 Pa)
57	P_{evap}	: Heat pump power exchange at the evaporator [W]
58	P_{LT}	: Power released by the low temperature heating system [W]
59	P_{cond}	: Heat pump power exchange at the condenser [W]
60	P_{comp}	: Electricity consumed at the heat pump compressor [W]
61	P_{elec}	: Total electric consumption of the dehumidifying apparatus [W]
62	P_{sens}	: Sensible heat exchange at the evaporator [W]
63	P_{sens}	: Heat transfer from the soil to the greenhouse [W]
64	$Q_{venting}$: Heat loss on a time step due to venting heating [J]
65	Q_{exch}	: Heat losses on a time step due to venting –heating with air exchanger [J]
66	q_{cond}	: Condensed water mass flow rate [$l \cdot h^{-1}$]
67	q_v	: Inlet air volumetric flow rate [$m^3 \cdot h^{-1}$]
68	r	: Humidity ratio [$kg_{vapour} \cdot kg^{-1}_{dry \ air}$]
69	S	: Greenhouse surface [m^2]
70	S_{tray}	: Surface of the balance tray [m^2]
71	T	: Temperature [$^{\circ}C$]
72	t	: Time [s]
73	v	: Air velocity [$m \cdot s^{-1}$]
74	V	: Greenhouse interior volume [m^3]
75	VPD	: Vapour pressure deficit [Pa]

76

77 Greek

78	Δt	: Recorded time step [s]
79	ΔH_v	: Water enthalpy of vaporization [$J \cdot kg^{-1}$]
80	φ	: Air relative humidity [%]
81	v	: Specific volume [$m^3 \cdot kg^{-1}_{dry \ air}$]
82	τ_r	: Air renewal rate [s^{-1}]

83

84 Subscript

85	1	: Air property at the dehumidifying apparatus
86	2	: Air property at the dehumidifying apparatus
87	3	: Air property at the dehumidifying apparatus
88	da	: Dry air
89	dew	: Property at dew point
90	e	: Exterior
91	i	: Inside the greenhouse
92	in	: Heating system input
93	out	: Heating system output
94	sat	: Saturation condition
95	v	: Vapour
96	w	: Water

97

98 Superscript

99	'	: Air exchanger output
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100 **1. Introduction**

101 **1.1. Dehumidification needs**

102 Humidity appears as a key factor in climate management of greenhouses. Excessive humidity is sometimes
103 combined with free water appearance (condensation), which creates favourable conditions for the development
104 of fungal diseases (*Botrytis cinerea* for instance) and enhances leaf necrosis as well as soft and thin leaves.
105 Droplets may also affect light transmission by modifying the optical properties of the cover material, and
106 consequently the photosynthesis activity. During the last two decades, the use of temperature integration with
107 lower temperature set point during the night to save energy led to an increase of relative humidity levels inside
108 greenhouses, and thus enhanced the dehumidification needs to preserve the crop.

109 A proper dehumidification method must be capable of avoiding water condensation on plant surfaces, which is
110 the first step towards disease prevention as fungi and other pests need water to develop. Condensation must also
111 be limited on the greenhouse cover in order to avoid dripping on the crop. Moreover, the operational cost of a
112 dehumidification method should be as low as possible to remain economically advantageous for the grower.

113 **1.2. Dehumidification methods based on venting**

114 **1.2.1. Venting-heating**

115 Venting-heating method is presently the most common method used by growers for dehumidification (Campen,
116 Kempkes & Bot, 2009). This method consists of opening the greenhouse to replace the moist greenhouse air by
117 relatively dry outside air. However, the outside air is often colder and must be heated to reach the set point
118 temperature. Even if this method remains acceptable for ornamental crops during the day when solar radiation
119 and outside air temperature are high, it is necessary to reconsider its economic and environmental acceptability
120 at night, dawn and dusk. At that time, venting-heating inevitably leads to an increase in energy consumption and
121 a decrease of the energetic efficiency of the greenhouse. Indeed, data retrieved by Grisey, Pommier, Chantry,
122 Piasentin, and Chassériaux (2007) in France and HDC in United Kingdom (Adams, Langton & Plackett, 2009)
123 state that venting-heating can account for 20% of the overall energy consumption of a greenhouse around the
124 year for a tomato crop and a relative humidity set point of 85% (respectively 12% for a humidity set point of
125 90%). For crops with lower transpiration rate as is the case for most of the ornamental species, the venting-
126 heating energy consumption should be lower. Nevertheless, due to the fact ornamental crops have lower heating
127 requirements, the venting-heating still accounts for a significant part of the energy consumption. Despite its
128 inefficiency, this method is still used as the installation requirements are only vents and a heating system that are
129 anyway mandatory for ornamental crops under temperate climates. Nevertheless, due to the increase of fossil
130 fuel cost in Europe and environmental impact of heating (CO₂ emission), the energy consumption has become a
131 growing concern and other methods have been investigated to limit the use of the venting-heating method.

132 **1.2.2. Venting heating with heat exchanger**

133 The simpler alternative to venting-heating is to retrieve energy from the outgoing warm and moist air and to
134 transfer it to the incoming colder dry air using an air-air exchanger. The aim is to reduce the heat losses due to
135 the ventilation and thus reduce the heating requirement. As all the heat cannot be retrieved from the outgoing air,

136 some energy is still lost. Moreover, a ventilation system is required in order for the air to pass through the
137 exchanger in both directions.

138 This system was evaluated by de Halleux and Gauthier (1998), using the properties of exchangers used for
139 livestock housing. For a soilless greenhouse tomato crop in Canada, they obtained a 40% saving with such a
140 device. Rouse, Martin, Thériault, Léveillé & Boily (2000) conducted a study using a low-cost exchanger
141 specially built for tomato and cucumber production in Canada. For a relative humidity set point of 75%, the
142 exchanger temperature efficiency was 78% and 84% for a ventilation rate of 0.9 and 0.5 air changes per hour.
143 The retrieved heat was significant: 1948 W for 0.5 air changes per hour and 2856 W for 0.9 air change per hour
144 (of which 40% was latent heat due to condensation of the outgoing vapour). In the latter case, the power needed
145 for the ventilation was measured to be 637 W for a 220 m² greenhouse (that is 2.9 W m⁻²) and the relative
146 humidity was not stable and often above 90%. The authors estimated that a ventilation rate between 2 and 5 air
147 changes per hour would be required to reach the humidity set point and a corresponding increase by 2 to 5 of the
148 power needed for the ventilation.

149 Nevertheless, the efficiency of such installation remains very high for a limited investment. In their study,
150 Campen, Bot, and de Zwart (2003) analysed a ventilation-heating method with an exchanger in order to compare
151 it with other dehumidification methods. The installation costs were between 8 and 13 € m⁻² for a heating gain
152 between 0.5 and 0.84 € m⁻² per year depending on the crop (and thus the transpiration). An air-air exchanger is
153 thus a very efficient solution to limit the supplementary energy consumption involved by venting-heating.

154 **1.3. Dehumidification heat pump**

155 An alternative to venting-heating is to use a heat pump dehumidifier that removes water vapour from moist air
156 using an electrically driven refrigeration cycle (Boulard, Baille, Lagier, Mermier & Vanderschmitt, 1989). The
157 goal is to limit the energy consumption by recycling the inside air instead of heating cold outside air and to
158 provide the energy retrieved from water vapour condensation back to the greenhouse. Consequently, the main
159 advantage of such a device is to minimise energy losses by re-using the energy extracted through condensation.

160 **1.3.1. Use in horticultural climate control**

161 To our knowledge, the first experimental works on using a heat pump for dehumidifying were conducted at the
162 end of the 1980's, for example, Boulard et al. (1989) and Chassériaux (1987).

163 Dehumidification uses mainly air-air heat pumps. The installed devices could consist of several small units
164 dispatched on the greenhouse surface (Campen & Bot, 2001) but most of the previous studies were based on one
165 larger unit for the entire greenhouse. For the latter, a system to homogenise the climatic conditions must be
166 installed, thus the initial investments in the homogenisation apparatus are expensive and deprive the economic
167 interest of the device by increasing the payback time to more than 30 years (Campen et al., 2003), although a
168 unique pump generally has a larger Coefficient Of Performance (COP) than several small ones. Moreover, the
169 use of a ventilation system implies a higher energy consumption although it guarantees a more homogeneous
170 greenhouse climate.

171 **1.3.2. Control strategies**

172 Experimental and simulated attempts with dehumidifying systems focused on maintaining a relative humidity set
173 point on crops with large leaf area index (LAI): Boulard et al. (1989) tried to maintain a relative humidity of

174 75% in a greenhouse with a tomato crop at 16°C and Campen et al. (2003) performed an analysis on simulated
175 data for a relative humidity set point of 85% for night temperatures between 18 and 20°C for rose, tomato, sweet
176 pepper and cucumber.

177 The former pointed out the ability of such apparatus to suppress the condensation on the cover by using a 2 kW
178 heat pump for a 400 m² greenhouse. However, the relative humidity set-point of 75% at 16°C was difficult to
179 maintain and a relative humidity around 90% was usual. Indeed, the water retrieval involved an increase of the
180 vapour pressure deficit (VPD), and thus an increase of the crop transpiration. For high LAI crop, the
181 transpiration largely compensated the condensation (Bakker, 2009), involving large dehumidification needs. In
182 the study of Campen et al. (2003), the dehumidifying heat pump made it possible to avoid condensation for a
183 maximum electric consumption at the compressor of 20 and 31 W m⁻² for rose and cucumber respectively. The
184 considered apparatus however was not economically interesting, mostly due to the high cost of the exchanger
185 (aluminium tubes) that accounted for more than two thirds of the installation cost. However, the operational cost
186 of the solution displayed an interesting saving for most of the crops, from 0.5 € m⁻² for sweet pepper to 1.15 € m⁻²
187 for rose.

188 As pointed out by Boulard et al. (1989), avoiding dew point on the roof and crop was relatively fast and easy in a
189 tomato greenhouse, but significantly reducing the inside air humidity was very energy consuming, particularly at
190 dawn and dusk. Hence, the investment cost of the dehumidification heat pump could not be balanced by
191 operational gains and the economic interest of such apparatus was null at that time.

192 The aim of this paper is to analyse the performance and to estimate the operational cost of a dehumidification
193 heat pump used in place of venting-heating to avoid condensation while maintaining a given temperature set
194 point at night. Contrary to previous published studies, the apparatus used in the present study acts as an
195 anticipative tool to avoid condensation rather than maintaining a relative humidity set point during the night.
196 Dehumidification is thus limited to what is strictly needed to avoid condensation at night. The characteristics of
197 the device used in the present work were established from the previous works of Chassériaux and Gaschet
198 (2009) who used a dynamic model of water vapour exchanges to determine changes in the temperature and
199 moisture concentration at the outlet of the evaporator. After a description of the experimental set up, the
200 dehumidification strategy is presented. Experimental results were obtained for ornamental crops in north western
201 France during the winter and the beginning of spring of 2010-2011. The energy consumption of the
202 dehumidification as well as the heating requirement were assessed and compared to the consumption of venting-
203 heating methods. This made it possible to compare the operating cost of different dehumidification methods.
204 Finally, a dehumidification strategy is proposed to choose the most efficient dehumidification method according
205 to outside climatic conditions and energy costs.

206 **2. Materials and methods**

207 **2.1. Greenhouse**

208 **2.1.1. Generality**

209 The experiment was conducted inside a 2353 m² (51.2 m width x 45.96 m length x 4 m height) double wall
210 inflatable greenhouse located in north western France (47.69216° N, 0.11259° W) from November 2010 to
211 March 2011. Different varieties of cyclamen (*Cyclamen persicum*) were grown from November 2010 to January

212 2011 with a density of 4 pots by square meter, and potted Hydrangeas (*Hydrangea macrophylla*, also several
213 different varieties) were placed in the greenhouse at the end of January till the end of our experiment (5 pots by
214 square meter). In both cases, the pots were watered by capillarity via a mat placed on the concrete ground. The
215 irrigation interval varied over the experiment ranging from every 4 days in November to every day in March.

216 **2.1.2. Heating system**

217 The main heating system of the greenhouse was a heating ground system: hot water plastic tubes were placed in
218 a concrete screed. Additional air heaters were installed and activated when the heating floor could not maintain
219 the temperature set point. The daily mean temperature in the greenhouse was set to 16°C. Both heating systems
220 were fed by hot water produced by a heating oil boiler.

221 To monitor the heating systems operation as well as the heat inputs in the greenhouse, the input and output water
222 temperatures were recorded for both the heating floor and the air heaters (Pt100, TConline, France). The energy
223 consumption for ground heating was calculated using the water temperature at the inlet and outlet, along with the
224 flow rate of the water circulation pump. The power released by the low temperature heating system (P_{LT}) may be
225 written as:

$$P_{LT} = \dot{m}_w C_{p_w} (T_{in} - T_{out}) \quad (1)$$

226 where $C_{p_w} = 4180 \text{ J kg}^{-1}$ is the water specific heat capacity, and T_{in} and T_{out} are the inlet and outlet water
227 temperature respectively. Due to the thermal dissipation in the system, the transferred heat was neglected when
228 $T_{in} - T_{out}$ was inferior to 2°C. The mass flow rate of water passing through the heating system, \dot{m}_w , was
229 calculated from the pump electric power alimentation using a linear relation between power consumption and
230 flow provided by the pump manufacturer, with a maximum flow rate of the heating system of $39.45 \text{ m}^3 \text{ h}^{-1}$.
231 Moreover, the heat flux between the soil and the greenhouse air was measured with two heat flux sensors
232 ($100 \times 100 \text{ mm}$ sensor, Captec, France) to determine the heat input from the floor heating system.

233 **2.2. Dehumidifying heat pump**

234 **2.2.1. Experimental setup**

235 Previous studies have shown that the elimination of water condensation can be achieved for an acceptable cost
236 using a dehumidifying heat pump (Boulard et al., 1989; Campen et al., 2003; Adams et al. 2009). A
237 dehumidifying heat pump operates by retrieving latent heat at the evaporator that is maintained colder than the
238 air dew temperature. After passing through the evaporators, the initial air (noted 1 on Figure 1) thus becomes
239 colder and dryer (2 on Figure 1). The latent heat of the condensed vapour is then given back on sensible form at
240 the condenser, resulting in the heating of the air. As the reheating of the air occurs only on sensible form, the
241 final air (3 on Figure 1) is both dryer and warmer than the initial one. As the crop continues to transpire
242 (especially if the relative air humidity decreases in the greenhouse), the machine must be powerful enough to
243 balance the transpiration and keep a safety margin on the water content to avoid condensation. Indeed, further air
244 dewatering was balanced by an increase of the transpiration and the dehumidification cost increased (Boulard et
245 al., 1989).

246 The heat pump dehumidifier retained here was a prototype unit manufactured by the ETT Company. The
247 machine was specifically designed to remove water vapour from moist air using two identical electrically driven
248 refrigeration cycles. The thermodynamic cycle of the air was the same as the one described on Figure 1. The
249 chosen installation was a dehumidification heat pump that could condense 32 litres by hour with a ventilation
250 flow of $10000 \text{ m}^3 \text{ h}^{-1}$, for a greenhouse of 2355 m^2 and a volume of 9280 m^3 .

251 To assess the operation of the dehumidifying heat pump, air temperature and relative humidity at the inlet and
252 the outlet of the machine were measured using shielded and ventilated sensors (Vaisala HMP45C, Vaisala,
253 Finland). The energy consumption of the heat pump compressors as well as the overall consumption of the
254 system including control and ventilation apparatus were measured by electricity meters (DIRIS A20, Socomec,
255 France). The condensed water flux was also measured by a tipping-bucket water fluxmeter (pluviometer 7852,
256 CIMA Technologie, France).

257 Air was sucked in at 3 m above the ground and driven towards the heat pump. The warm and dry air was then
258 injected using eight polyethylene (PE) ducts at 3 m above the ground (Figure 2). The dry and heated air was
259 forced through the tubes and distributed in the greenhouse by holes perforated in the PE. The mean output air
260 velocity through the holes was measured to be $v = 13.6 \text{ m s}^{-1}$ using a hot wire anemometer (type GGA-35,
261 ALNOR, Finland), with a total variation of 7% between the first and last hole along a duct. This involves that the
262 part of the greenhouse closer to the heat pump had a higher air renewal rate than the farther end of the
263 greenhouse. We considered the differences to be acceptable for the aim of this study based on the overall
264 efficiency of the prototype.

265 As the ducts were approximately 500 mm in diameter, the total horizontal surface of the ventilation system was
266 of the order of 190 m^2 , or 8 percent of the total greenhouse surface. However, to limit the shadowing effect, the
267 ducts were deflated during the day when the heat pump was off. Moreover, as the greenhouse cover film was
268 opalescent, the effects of the ducts on the radiation reaching the crop were limited.

269 **2.2.2. Control**

270 As the dehumidifying heat pump was designed to avoid condensation on plants in place of the classical venting-
271 heating procedure, it was turned on when the greenhouse openings were closed and when the Vapour Pressure
272 Deficit (VPD) of the air inside the greenhouse became lower than 3 hPa (Migeon, Pierart, Lemesle, Travers &
273 Chassériaux, 2012). The use of a VPD set point in place of a relative humidity set point was due to the better
274 robustness of this climatic parameter with respect to temperature changes. The adopted value was a security
275 margin to ensure that no condensation would occur at the end of the day. Indeed, the inside temperature then
276 remained high, with an elevated air humidity ratio, but due to the reduction of the exterior temperature at sunset,
277 an important amount of vapour should be eliminated from the inside air before it condenses. For that reason, and
278 in order to anticipate this problem, the VPD evolution rate (VPD slope) was also considered for an advanced
279 start-up of the dehumidifying heat pump in order to anticipate temperature decrease at sunset. The chosen values
280 were selected based on the results of trial and error testing during the seasons previous to this study.

281 Accordingly, at sunrise the crop transpiration started up as soon as the solar radiation was sufficient, but at that
282 time, the air was still cold, involving condensation risk. For that reason, the dehumidifying heat pump could
283 remain active after the VPD reached 3 hPa, depending on the vapour pressure deficit evolution rate.

284 **2.2.3. Operation measurements and parameters of interest**

285 The compressor efficiency of the heat pump was calculated by considering that the air, of which properties were
 286 measured at the inlet, was first cooled to its dew point and then followed the saturation curve until it left the
 287 exchanger (point 2 on Figure 1). The power exchange at the evaporator, denoted P_{evap} , was then defined as:

$$P_{evap} = -\frac{q_v}{v_3}(H_2 - H_1) \quad (2)$$

288 Where q_v is the air volumetric flow rate ($\text{m}^3 \text{h}^{-1}$) through the heat pump, H_1 and H_2 are the air enthalpies (kJ kg^{-1}
 289 $_{\text{dry air}}$) at points 1 and 2 on Figure 1, and v_3 is the specific volume ($\text{m}^3 \text{kg}^{-1}_{\text{dry air}}$) at point 3. The sensible heat
 290 retrieved from the air, denoted P_{sens} , was calculated using a classical heat balance on the dry air (subscript “ da ”)
 291 and the residual water vapour (subscript “ v ”) using their specific heat (C_p):

$$P_{sens} = -\frac{q_v}{v_3}(C_{p_{da}} + r_2 C_{p_v})(T_2 - T_1) \quad (3)$$

292 The air is then reheated at the heat pump condenser. The power exchange at the condenser, P_{cond} , may be written
 293 as:

$$P_{cond} = -\frac{q_v}{v_3}(H_3 - H_2) \quad (4)$$

294 The compressor efficiency e_T was then calculated using P_{comp} (power used by the compressor) by:

$$e_T = -\frac{P_{cond}}{P_{comp}} = \frac{P_{cond}}{P_{evap} + P_{cond}} \quad (5)$$

295 The overall efficiency of the installation (denoted e_o) including the electric consumption of the auxiliary devices
 296 of the heat pump and of the ventilation system is relevant to compare different dehumidifying installations. It is
 297 defined as:

$$e_o = -\frac{P_{cond}}{P_{elec}} \quad (6)$$

298 The condensed water mass (m_{cond}) was calculated from the difference between the humidity ratio at the inlet and
 299 at the outlet. Measurements were recorded every 10 minutes and the mass of water condensed on the time step
 300 was calculated according to:

$$m_{cond} = \frac{q_3}{v_3}(r_1 - r_3) \cdot \Delta t \quad (7)$$

301 with q_3 the mean air volumetric flow rate through the heat pump during a 10 minutes time step (noted Δt),
 302 evaluated at the dehumidifier outlet.

2.3. Climate measurements

2.3.1. Outdoor climate

To determine the influence of the outdoor climate on the greenhouse energy consumption, the outdoor conditions were monitored. Moreover, to estimate the energy consumption of venting-heating dehumidification, the outdoor air properties must be known.

A meteorological station was placed above the greenhouse roof to monitor the exterior climate. It recorded the temperature and relative humidity of the air with HMP45C sensors (Vaisala, Finland). The wind speed was measured using a cup anemometer (HA 430A, Geneq Inc., Canada), and its direction was recorded using a wind vane (W 200 P, Campbell Scientific Ltd, UK). The solar radiation was also measured with a pyranometer (CM-3, Kipp & Zonen, The Netherlands).

Given the air temperature the saturation vapour pressure $P_{v\,sat}$ can be determined following Nadeau and Puiggali (1995):

$$P_{v\,sat} = \exp \left\{ 23.3265 - \frac{3802.7}{T + 273.18} - \left(\frac{472.68}{T + 273.18} \right)^2 \right\} \quad (8)$$

Combined with the relative humidity, it was used to determine the dew temperature T_{dew} and the air humidity ratio r :

$$T_{dew} = -273 - \frac{5120}{\ln \left(\frac{\varphi}{100} \cdot \frac{P_{v\,sat}}{10^5} \right) - 13.73} \quad (9)$$

$$r = 0.622 P_{v\,sat} \frac{\varphi/100}{P_{atm} - P_{v\,sat} \frac{\varphi}{100}} \quad (10)$$

2.3.2. General climate in the greenhouse

To quantify the climate inside the greenhouse and thus the influence of the heat pump on the greenhouse air, temperature and relative humidity were recorded at four vertical positions at 0.47, 1.92, 3.37 and 4.82 meter high (Figure 3) in the middle of the greenhouse (shaded square on Figure 2) using shielded and ventilated sensors (Vaisala HMP45C, Vaisala, Finland). Moreover, a pyranometer recorded the solar radiation at the crop level (CE 180, CIMEL, France), and the total radiative thermal exchange was measured by a net radiometer (REBS, Campbell scientific, USA).

2.3.3. Microclimate and evapotranspiration

In order to assess the condensation on the crop, the microclimate inside the canopy was recorded in the middle of the greenhouse (shaded square on Figure 2) at two heights by measuring air temperature and relative humidity inside the crop (Vaisala HMP45C, Vaisala, Finland). The lower sensor was placed at 11.5 or 12 cm above the surface depending on the pot height. The upper one was placed at 25 m above the surface for cyclamens crop (top of the canopy from 26 to 36 cm above the surface) and for Hydrangeas crop (top of the canopy from 34.5 to 42 cm above the surface). The leaf temperatures were also measured at the upper, middle and lower parts of the

331 plants by two thermocouples for each level. These measurements made it possible to assess the occurrences of
332 condensation on the crop (Figure 3).

333 The evapotranspiration (ET) was measured using a specially designed scale. Ten plants were placed on the
334 balance tray ($S_{tray} = 2 \text{ m}^2$). A tension weighing load cell measured the water loss by evapotranspiration. The mat
335 plant watering system was the same on the balance and on the ground. As the mat was wetted for irrigation, the
336 measured weight was subjected to important variations, and the water evaporation remained important for
337 approximately an hour. Corresponding points were then dropped from the analysis.

338 The weight loss (Δm) between two time steps (t and $t + \Delta t$) was linked to ET using the enthalpy of vaporization
339 ΔH_v :

$$ET = -\frac{\Delta H_v \cdot \Delta m}{S_{tray} \cdot \Delta t} \quad (2)$$

340 All the measurements were recorded on one-minute intervals, but due to their large number and the duration of
341 the experiment, they were averaged (or summed over 10-min periods) by the data loggers before storing.

342 **3. Results**

343 **3.1. Climate and microclimate characteristics, evapotranspiration**

344 Climate homogeneity inside the greenhouse was assessed during a previous study (Migeon et al., 2012), and it
345 was shown that neither significant temperature nor humidity vertical gradients appeared (over the canopy) during
346 the heat pump operation. This seems due to the air mixing induced by the air jet coming out from the holes of the
347 PE ducts. Conversely, during the day, when the heat pump did not operate, a vertical temperature gradient was
348 established with a 1.5°C difference between the lowest and highest measurement points.

349 Concerning the microclimate inside the crop, results are displayed in Figure 4. As expected, the diurnal
350 conditions were safe from condensation risk, but at the end of the day, the leaf temperatures decreased toward
351 the dew temperature, that is the temperature at which the water vapour of the air could condense on the leaves.
352 By retrieving the water vapour from the greenhouse air, the heat pump operation limited the increase of the dew
353 temperature and no observed leaf was cold enough for the vapour to condense on it. During the night, the
354 dehumidifier stabilised the dew temperature while the heating of the greenhouse stabilised the leaf temperature,
355 ensuring that no condensation occurred. This was confirmed by visual inspections conducted by the producer
356 who did not observe any plant infected by *Botrytis Cinerea*.

357 **3.2. Dehumidifier operation**

358 As stated, the dehumidification heat pump was designed to balance the night evapotranspiration of the crop, as
359 the low temperature during the night was the most prone to induce condensation. Figure 5 displays the measured
360 evapotranspiration during one night in March along with the condensed water by the heat pump and its operating
361 period. It can be seen that during the night, evapotranspiration was balanced by condensation at the heat pump.
362 During operation, the condensation rate was around 12 W m^{-2} (or 45 l h^{-1}) for the considered period. Significant
363 variations of the evapotranspiration or condensation by heat pump occurred depending on the air temperature
364 and relative humidity. These variations involved changes of the heat pump operation and efficiency.

365 Measurements indicate that the average electrical power of the two compressors was $P_{\text{comp}} = 7.62$ kW and the
366 total electric power required by the machine was $P_{\text{elec}} = 10.23$ kW (4.4 W m⁻²). Consequently, it was deduced
367 that the power dissipated by the ventilation fan and accessories was approximately 2.61 kW. During heat pump
368 operation, the average power exchanged at the condenser was $P_{\text{cond}} = -48.7$ kW (deduced from Eq. 4), and P_{evap}
369 = 41.1 kW (deduced from Eq. 2) at the evaporator, 57% of which corresponded to the latent heat generated from
370 water vapour condensation. Along with the electric consumption, these data made it possible to calculate the
371 thermodynamic and the overall efficiencies of the heat pump during its operation, as displayed in Figure 6 for
372 March 20th and 21th. On this graph, the compressor efficiency (Eq. 5) was approximately $e_{\text{comp}} = 7.5$ and the
373 overall efficiency (Eq. 6) $e_o = 5$. As stated before, those values depend on the operating condition, mostly the
374 greenhouse air properties. For the entire experiment from November the 1st to April the 1st, the mean values for
375 thermodynamic and overall efficiency were $e_{\text{comp}} = 8$ and $e_o = 4.9$ respectively.

376 3.3. Heating operation

377 The energy consumption for ground heating was calculated using Eq. 1. However, the energy lost by the buried
378 hot water tube was not entirely transferred to the greenhouse air, as a significant part heated the soil downward
379 and between the tubes and the soil surface. In order to estimate the heat flux from the soil to the greenhouse air
380 (P_{soil}), the heat flux measurements from two flux meters were averaged to get an estimation of heat flux.

381 Moreover, the dehumidifier not only condensed water vapour but also transferred a significant amount of
382 sensible heat to the greenhouse air (Eq. 3). When the cold air passed through the evaporator it was first cooled,
383 leading to vapour condensation. Latent heat was then transferred from the water to the heat pump system. A
384 significant amount of heat was also retrieved from the dry air and the residual water vapour. At the condenser,
385 the heat transferred from the heat pump to the air was entirely sensible heat. Thus, apart from the condensation
386 involved by latent heat, the sensible heat balance should be considered in order to estimate the heating
387 contribution of the dehumidifier in the overall greenhouse heating system. Figure 7 displays the heat flux
388 transferred to the greenhouse by the ground heating system, the associated energy consumption for water heating
389 (named heat floor consumption) and the heat pump sensible contribution for seven consecutive days in March. It
390 can be seen that the heating system was only used at night, for an output power between 100 and 175 W m⁻².
391 Obviously, for winter conditions the ground heating period was much longer with a continuous operation at the
392 end of November. Part of this energy was lost for the greenhouse heating, which explains why the effective heat
393 flux input in the greenhouse (heat floor contribution in Figure 7) was much lower. The differences between the
394 two values was due to the heat flux downward from the hot tubes that heated the soil below, and the energy
395 needed to heat the soil between the tubes and the soil surface before heat could reach the greenhouse.
396 Consequently, when the ground heating system operated at full capacity, only a small part of the energy
397 consumed to produce hot water was used to heat the greenhouse. For the entire investigated period, the mean
398 utilisation ratio was of the order of 0.5, meaning that half the energy consumed (i.e., gas consumption) was lost.

399 Moreover, one can see that the heat flux between the soil and the greenhouse air was decoupled from the flux out
400 of the hot water tubes. Indeed, the soil was heated by solar radiation during the day (negative flux in Figure 7)
401 and part of the energy was given back to the greenhouse at the beginning of the night.

402 Also observable from Figure 7, the heat pump contribution to greenhouse heating is lower than that of the
403 ground heating system. However, for instance during the first part of the night between March 25th and 26th, the

404 heat pump contribution, with the heat destocking from the soil, was sufficient to delay the ground heating
405 operation up until midnight. Such a situation of a warm night following a warm day was usual during spring and
406 in this case a limited ground heating input was needed. It was also observed that the dehumidifying heat pump
407 sometimes transferred enough energy to the greenhouse to maintain the temperature above the set point of 16°C
408 without requiring the use of the greenhouse heating system (for instance from March 23rd to 25th).

409 When complementary ground heating input was necessary, its use was limited thanks to the heat pump. Figure 8
410 displays the part of the heat pump energy input to the total greenhouse heating input (soil flux and heat pump
411 sensible contribution) during the experiment together with the daily mean exterior temperature. It must be stated
412 that due to experimental difficulties, the ground flux was not always available. However, one can see that the
413 heat pump contribution was between 10 and 40% of the total heat input. If the total needs were important, as was
414 the case most of the time in November when temperatures were low, the heat pump contribution was reduced
415 due to the heavy use of the ground heating system. At the end of March the exterior mean temperatures were
416 higher, the total heating requirements were lower and the heat pump contribution was proportionally higher.

417 **4. Discussion**

418 The heat pump contribution to the greenhouse heating was crucial, as it significantly reduced the ground heating
419 system use, and therefore the natural gas consumption. Thus, when discussing the efficiency of the various
420 dehumidification methods, one needs to take the heating contribution of the dehumidifier into account.

421 **4.1. Total energy consumption**

422 In the condition described here, the total energy consumption was the sum of the natural gas consumption for the
423 base heating of the greenhouse (ground heating) and the electric consumption of the heat pump. These
424 consumptions depended on the outdoor climate (both the temperature and humidity) and on the temperature set
425 point, as depicted in Figure 9. However, if the consumption for greenhouse heating strongly depended on the
426 exterior climate, the power needed by the dehumidifier hardly varied with the exterior conditions. Indeed, it
427 operated only on the inside air, contrary to the dehumidifying procedures that involve air renewal with exterior
428 air. Hence, large variations of the power required for dehumidification may be expected in the case of the
429 venting heating and air heat exchanger procedure.

430 **4.2. Comparison with venting heating**

431 By using the simple venting-heating procedure, supplementary heating needs due to the vent opening, denoted
432 $Q_{venting}$, occurs. The method used to estimate the heat losses induced by the venting-heating process is described
433 in Appendix A, leading to the value on a time step:

$$Q_{venting} = m_{cond} \frac{H_i - H_e}{r_i - r_e} \quad (3)$$

434 Where $Q_{venting}$ represents the heat losses specific to the venting and thus the supplementary heating needs.
435 When added to the value of the dehumidifier sensible heat input, this gives the total heat input differences
436 between venting heating and thermodynamic dehumidifier.

437 As described before, not all the energy used for ground heating ultimately heated the greenhouse, and a large
438 part was lost. As other heating devices could be used with lower to no heat loss, $Q_{venting}$ must be seen as the

439 lower bound for the supplementary gas consumption to be provided. In the following, we therefore
440 underestimate the total energy consumption by a factor that depends on the heating system used.

441 The corresponding different energy consumptions are displayed on Figure 10 along with the energy consumption
442 ratio of the natural gas estimated consumption by venting-heating to the measured electricity consumption by the
443 heat pump. The electrical consumption of the heat pump remained stable but the base heating gas consumption
444 strongly depended on the daily climate. The calculated gas consumption for venting-heating (square symbols)
445 had an intermediate behaviour as the dehumidification needs were more stable, even if it also depended on the
446 exterior conditions. The dehumidification ratio (round symbols) is the ratio between the calculated gas
447 consumption for dehumidification by venting-heating and the measured electrical consumption for the same day.

448 The ratio is comprised between 6 and 8.5, meaning that in order to obtain the same dehumidification, the
449 venting-heating method would require 6 to 8.5 times more energy than the present heat pump. The ratio was all
450 the more important, as the exterior temperature was higher. Indeed, for high exterior temperatures, the sensible
451 heat pump heating contribution was proportionally higher. Due to the compressor efficiency being superior to 1,
452 this involved a more important decrease of the overall energy consumption.

453 As the thermodynamic dehumidifier required less energy than the venting-heating procedure, the operating cost
454 of the dehumidification was lower. However, the investment cost for a large heat pump could be significant,
455 contrary to the venting heating procedure, which only requires more natural gas consumption for an existing
456 installation.

457 **4.3. Comparison with venting heating using a heat exchanger**

458 The principle is here the same as in the previous paragraph, but a heat exchanger is placed in the path of the
459 input air in order to retrieve heat from the outgoing air. The calculation of the involved heat losses is provided in
460 Appendix B, leading to the supplementary energy needs Q_{exch} :

$$Q_{exch} = \frac{m_{cond}}{r_i - r_e} (H_i - \min(H_e + H_i - H'_i; (Cp_{da} + r_e Cp_v) T_i)) \quad (4)$$

461 As for classical venting-heating, this quantity represents the supplementary heating required for the
462 dehumidification of the greenhouse. In order to evaluate the entire energy needs to replace the heat pump, the
463 sensible heat contribution of the heat pump must be added to Q_{exch} .

464 An estimation of the operational energy cost of the venting heating with the air exchanger can then be made. It
465 should be noted that the energy consumption for the ventilation (mandatory to use a heat exchanger) was not
466 taken into account in this study. Moreover, like for classical venting-heating method, Q_{exch} was assumed equal to
467 the energy consumption, providing the lowest estimation of the total natural gas consumption.

468 Figure 11 displays the energy consumptions for heat pump dehumidification and venting-heating with an
469 exchanger along with the energy consumption ratio. As for the classical venting-heating method, the total
470 consumption was several times larger than for the dehumidifying heat pump. Thanks to the heat exchange, the
471 consumption was, however, smaller than with the classical venting-heating and the energy consumption ratio
472 was between 3 and 8.

4.4. Elements of dehumidification strategy

In order to discuss the opportunity to use a heat pump as dehumidifier, the operational cost must be assessed as well as the investment cost and depreciation. In this paper we only investigate the operational costs of the proposed apparatus in order to compare them to the operational costs of classical dehumidification methods. The comparisons serve two purposes: on the one hand, they define the operational conditions when dehumidification is more efficiently done with a heat pump and when it is not, providing some dehumidification strategy elements; on other hand, they give an operational cost to include in a proper economic analysis along with investment and depreciation costs. Indeed, as investment costs strongly depend on the effective horticultural system (plant needs and climate), such analysis goes beyond the scope of the present study.

4.4.1. Variations of the energy consumption ratio

Figure 12 displays the daily energy consumption ratio depending on the difference of mean air enthalpy inside and outside the greenhouse for both cases: venting-heating and venting-heating with heat exchanger. The energy consumption ratio for both venting heating method either remained stable or increased with the air enthalpy difference. Indeed, as the inside air was replaced by colder exterior air, the more energy differences there were, the more heat loss. As heat was retrieved from the inside air in the case of a heat exchanger, the heat loss was lower in this case, as depicted by the lower energy consumption ratio on Figure 12. However, for high differences of air enthalpy, the energy ratios tended to the same value as they corresponded to cold outside temperature, where the sensible heat input of the thermodynamic dehumidifier was more important. Thus, heat recovering accounted for only a small fraction of the overall needs, and the energy consumption ratio differences between the two methods decreased.

The influence of the air humidity ratio inside and outside the greenhouse on the energy consumption ratios is displayed on Figure 13. The energy consumption ratios increased with the difference of water contents. Indeed, for more or less constant interior air conditions (controlled temperature and relative humidity), large humidity ratio differences corresponded to situations for which exterior air had a low humidity ratio, and thus a low temperature. In such conditions, the heat losses were important in the case of ventilation, needing a high energy consumption to restore the greenhouse air temperature.

Overall, the heat pump energy consumption was at least 3 times lower than the energy consumption for the other dehumidification methods based on ventilation; however, using the proposed dehumidifier involved replacing the natural gas by electricity as an energy source. As the costs of these two sources are different, the operational cost should be assessed.

4.4.2. Impact of the energy prices

Table 1 displays the electricity and natural gas prices for selected countries, as well as the energy price ratio calculated by natural gas prices / electricity prices. A low ratio indicates that the natural gas is relatively cheap compared to electricity; a high ratio indicates that the natural gas price has a price close to the electricity one. Obviously, energy costs strongly depend on the country geographic location, energy production systems and applied energy charges. Canada has the lowest prices ratio because it produces natural gas. On the contrary, South Korea has the highest price ratio with the lowest electricity price of the list.

510 As electricity is generally more expensive than the natural gas, the energy consumption gains performed by
511 switching from ventilation (using natural gas) to heat pump (using electricity) is limited at the cost level by the
512 price ratio. In order for the heat pump dehumidifier to be more efficient, the product of the energy consumption
513 ratio by the price ratio must be superior to one. The last column of Table 1 displays the minimum energy
514 consumption ratio needed by the heat pump to be cost effective.

515 The energy consumption ratios depicted on Figure 12 and Figure 13 were above 6 for classical venting-heating
516 and 3 for venting-heating with heat exchanger. Thus, for the country where the minimum energy consumption
517 ratio displayed on Table 1 is inferior to 3, the heat pump dehumidifier is interesting from an operational cost
518 point of view. It is the case for Japan, France, Germany and South Korea in the examples depicted on Table 1.
519 For the country with high minimum energy consumption ratio such as Canada, the heat pump dehumidifier
520 would be favourable only under cold conditions with high air enthalpy difference between the greenhouse and
521 the exterior. Under such conditions, the interest of a heat pump dehumidifier installation therefore seems
522 unlikely.

523 For intermediate cases, where the minimum energy consumption ratio is comprised between 3 and 4 or 4.5, the
524 opportunities of a heat pump dehumidifier must be deeply analysed depending on the exterior climate and
525 growing conditions. Moreover, the installation of a heat pump dehumidifier does not involve its use
526 continuously, and the dehumidification method can be chosen according to the conditions (outside climatic
527 conditions + inside temperature and humidity set points).

528 Although the present study was based on ornamental crop with a low transpiration rate and low heating needs, its
529 conclusions could be applied to different species whose transpiration rates and heating needs are higher, such as
530 tomato or cucumber, with a more powerful heat pump to condense more water. However, for such crops, as the
531 heating needs would be larger, the sensible heat contribution of the dehumidifier would be less significant,
532 reducing the economic interest of the dehumidification heat pump. Moreover, the large greenhouses used for
533 such cultures would involve an important investment cost that would be difficult to justify.

534 **4.4.3. Recommendations**

535 As depicted on Figure 12 and Figure 13, the energy consumption ratio evolved when the outside climate
536 changed. It appears that, for a fixed greenhouse air, the energy ratio depended on the exterior air enthalpy – and
537 thus its temperature and humidity ratios. Thus, the measurement of air properties would give an indicator of the
538 most efficient dehumidification procedure for actual climate. As the classical venting-heating was the most
539 inefficient dehumidification procedure whatever the conditions, the choice was limited to venting-heating with
540 exchanger and heat pump.

541 In the former case, if ground heating is efficient for base heating of a crop on the greenhouse floor, using another
542 system to heat the air in case of venting-heating through an exchanger should be considered to limit energy
543 waste.

544 The natural gas heating system remains mandatory for a greenhouse to balance high heat loss during cold winter,
545 even if a heat pump is installed. It is then simple to switch from one dehumidification system to the other, and
546 hence to choose the most efficient one depending on the exterior conditions. Moreover, the inclusion of weather

547 forecast to greenhouse climate control automation would allow an easy use and an efficient dehumidification
548 solution.

549 Further researches remain however necessary to improve the energetic efficiency of the apparatus. Indeed, the
550 dry and hot air circulation in the greenhouse is still unknown, and some energy saving could be achieved by a
551 proper Computational Fluid Dynamics analysis of the air circulation. Thus, a better circulation scheme could be
552 determined to concentrate the dehumidification effects on the crop, while allowing for easy access for plants
553 handling. Moreover, the apparatus scaling and control remain crude, and the overall efficiency would gain from
554 a control based on accurate measurements of the VPD around the crop and of the leaf temperature: the heat
555 pump would then be more tightly controlled to minimize the energy consumption.

556 **5. Conclusions**

557 The proposed dehumidification heat pump was designed to avoid dew point during the night by balancing the
558 night evapotranspiration. Thanks to an anticipated operation before sunset, this strategy revealed its efficiency by
559 effectively preventing condensation on the crop. The apparatus thus made possible the growth of sensitive crops
560 that would otherwise potentially be infected by fungus such as *Botrytis Cinerea*.

561 The system analysis showed that the heat input by the heat pump as a by-product of the dehumidification, was
562 significant and could lead to a large reduction of the conventional heating used in favourable conditions. Indeed,
563 during autumn and spring when the heating needs were reduced, the heat pump sensible heat inputs were
564 sufficient to maintain the temperature set point, making it possible to avoid any natural gas consumption under
565 favourable conditions.

566 Model comparisons with classical venting heating showed that the described device was 6 to 8.5 less energy
567 consuming than the classical dehumidification method. Moreover, the comparison with venting-heating with an
568 air exchanger showed that despite the heat recovery at the exchanger, the heat pump dehumidifier presented here
569 remained 3 to 8 times less energy consuming than the greenhouse ventilation.

570 The operational cost gains of this dehumidification approach depend on the relative cost of electric and fossil
571 energy, suggesting that the presented dehumidification strategy could be relevant for a number of countries.
572 Furthermore, dehumidification could be obtained by using the most cost efficient method at a given time, either
573 the described system or by the venting-heating, depending on the operating conditions and the associated heating
574 requirement.

575 Further researches remain needed to optimize the apparatus, particularly the dry and hot air ventilation system
576 and the control of the heat pump. At last, a complete cost analysis of the system should also be conducted to
577 incorporate the cost of the heat pump in order to perform a thorough economic study

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615

616 8. Appendix A

617 By using venting heating combination, greenhouse vent are opened allowing warm and moist indoor air leaving
618 the greenhouse. This air is replaced by cold and dry air from the exterior. For a given time step, the water vapour
619 eliminated m_{elim} is equal to:

$$m_{elim} = \tau_r \frac{V}{v_i} (r_i - r_e) \quad \text{A 1}$$

620 Where τ_r is the ventilation rate, defined as the volume of the greenhouse renewed during the time step, V the
621 greenhouse volume and r is the air humidity ratio in kg vapour by kg of dry air at the interior (r_i) and the exterior
622 (r_e) of the greenhouse.

623 Replacing the warm air with colder and dryer one involves heat losses that must be balanced by more heating.
624 The supplementary heat loss, $Q_{venting}$, on the time step can be calculated by enthalpy balance:

$$Q_{venting} = \tau_r \frac{V}{v_i} (H_i - H_e) \quad \text{A 2}$$

625 In order to eliminate the same quantity of water as the dehumidifying heat pump, one must have:

$$m_{elim} = m_{cond} \Leftrightarrow \tau_r \frac{V}{v_i} (r_i - r_e) = m_{cond} \quad \text{A 3}$$

626 Thus, the ventilation rate must be equal to:

$$\tau_r = m_{cond} \frac{v_i}{V} \frac{1}{r_i - r_e} \quad \text{A 4}$$

627 Replacing this expression of τ_r in equation A.2 leads to:

$$Q_{venting} = m_{cond} \frac{v_i}{V} \cdot \frac{1}{r_i - r_e} \cdot \frac{V}{v_i} (H_i - H_e) = m_{cond} \frac{H_i - H_e}{r_i - r_e} \quad \text{A 5}$$

628 9. Appendix B

629 From Appendix A, the supplementary heat required to balance the venting-heating heat loss is given by:

$$Q_{vent} = m_{cond} \frac{H_i - H_e}{r_i - r_e} \quad \text{B 1}$$

630 This formulation can be used in the case of an exchanger:

$$Q_{exch} = m_{cond} \frac{H_i - H'_e}{r_i - r_e} \quad \text{B 2}$$

631 with H'_e the air enthalpy of the ingoing air after the exchanger (cf Figure 14).

632 If no heat loss occurs at the exchanger (ie. no heat flux from the ingoing or outgoing air toward the exterior), the

633 energy needs are the same whatever side of the exchanger is considered. Thus, Q_{exch} can be written at the exterior

634 side of the exchanger:

$$Q_{exch} = m_{cond} \frac{H'_i - H_e}{r_i - r_e} \quad \text{B 3}$$

635 And then:

$$Q_{exch} = m_{cond} \frac{H_i - H'_e}{r_i - r_e} = m_{cond} \frac{H'_i - H_e}{r_i - r_e} \quad \text{B 4}$$

636 Hence:

$$H_i - H'_e = H'_i - H_e \quad \text{B 5}$$

$$\Leftrightarrow H'_e = H_e + H_i - H'_i = H_e + Q_{trans} \quad \text{B 6}$$

637 The energy needs Q_{exch} can be determined by replacing H'_e in Eq. B 4 by its expression provided by Eq. B 6.
 638 Then, H'_e must be determined from H_e , H_i and H'_i . If H_e and H_i can be calculated using the properties of the airs
 639 using the temperatures and water, those properties are not known for the outgoing air after the exchanger (air i').
 640 Some assumptions must be made to determine H'_i .

641 If we assume that all the recoverable energy is effectively gained by the ingoing exterior air, then: $T'_i = T_e$.

642 9.1. $T_e > T_{dew, i}$

643 If T_e is above the dew temperature of the interior air ($T_{dew, i}$), the outgoing air is cooled without any vapour
 644 condensation. Thus, the exchanged energy on the 10 minute time step can be determined using:

$$Q_{trans} = H_i - H'_i = (Cp_{da} + r_i Cp_v)(T_i - T_e) \quad B 7$$

645 Injecting Eq. B 7 in Eq. B 6 yields:

$$H'_e = H_e + H_i - H'_i = H_e + Q_{trans} = H_e + (Cp_{da} + r_i Cp_v)(T_i - T_e) \quad B 8$$

646 9.2. $T_e \leq T_{dew, i}$

647 If the temperature of the exterior is inferior to the dew temperature of the interior air, both latent and sensible
 648 heat exchanges are involved. In this case, the final humidity ratio of the outgoing air is calculated using:

$$r'_i = 0.622 P_{v sat (T_e)} \frac{1}{P_{atm} - P_{v sat (T_e)}} \quad B 9$$

649 With $P_{v sat (T_e)}$ is the saturation vapour pressure at T_e . The total heat exchange is thus the sum of:

- 650 • The cooling of the initial greenhouse air from T_i to the dew temperature of the greenhouse air, denoted
 651 $T_{dew(i)}$:

$$Q_1 = (Cp_{da} + r_i Cp_v) \cdot (T_i - T_{dew, i}) \quad B 10$$

- 652 • The latent heat of condensation of $r_i - r'_i$ in kg by kilo of dry air:

$$Q_2 = \Delta H_v (r_i - r'_i) \quad B 11$$

- 653 • The cooling of the remaining water vapour and the dry air from $T_{dew(i)}$ to the final temperature T_e :

$$Q_3 = (T_{dew, i} - T_e) \cdot [(Cp_{da} + r'_i Cp_v) + (r_i - r'_i) \cdot Cp_{water}] \quad B 12$$

654 In this case, the total exchanged heat is:

$$\begin{aligned}
Q_{trans} &= Q_1 + Q_2 + Q_3 \\
&= (Cp_{da} + r_i Cp_v) \cdot (T_i - T_{dew,i}) + \Delta H_v(r_i - r'_i) \\
&\quad + (T_{dew,i} - T_e)[(Cp_{da} + r'_i Cp_v) + (r_i - r'_i) \cdot Cp_{water}]
\end{aligned}
\tag{B 13}$$

655 and

$$\begin{aligned}
H'_e &= H_e + Q_{trans} \\
&= H_e + (Cp_{da} + r_i Cp_v) \cdot (T_i - T_{dew,i}) + \Delta H_v(r_i - r'_i) \\
&\quad + (T_{dew,i} - T_e)[(Cp_{da} + r'_i Cp_v) + (r_i - r'_i) \cdot Cp_{water}]
\end{aligned}
\tag{B 14}$$

656 The final temperature of the ingoing air cannot rise above the inside temperature, otherwise heat would be
657 exchanged from the ingoing to the outgoing air. To depict this limit, and finally:

$$H'_e = \min(H_e + Q_{trans}; H_e + (Cp_{da} + r_e Cp_v) \cdot (T_i - T_e))
\tag{B 15}$$

658 where Q_{trans} is calculated either by Eq. B 7 or B 13 depending on the case.

659

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Figure 1

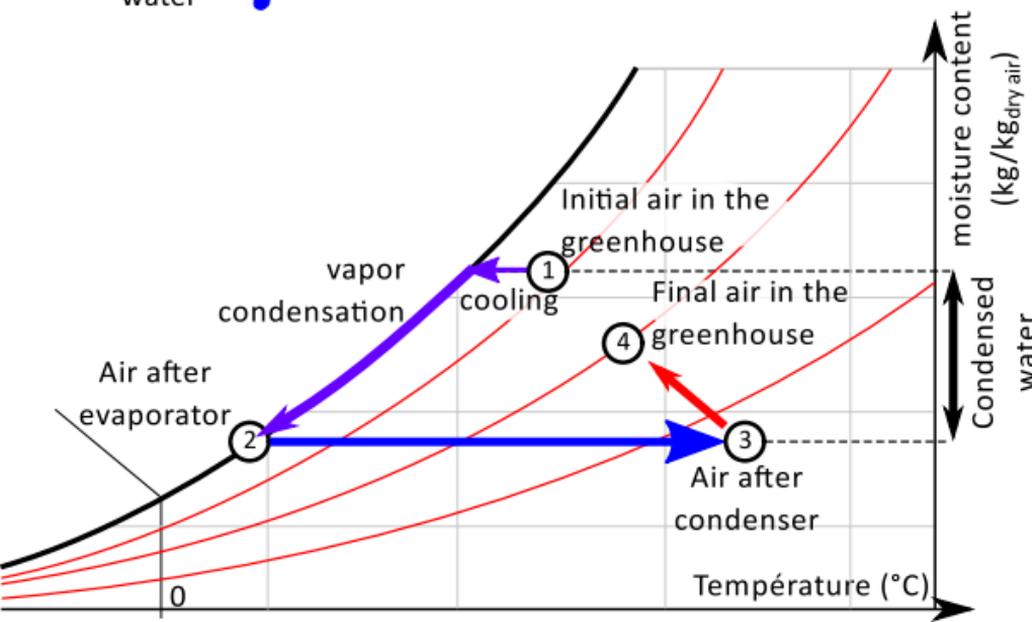
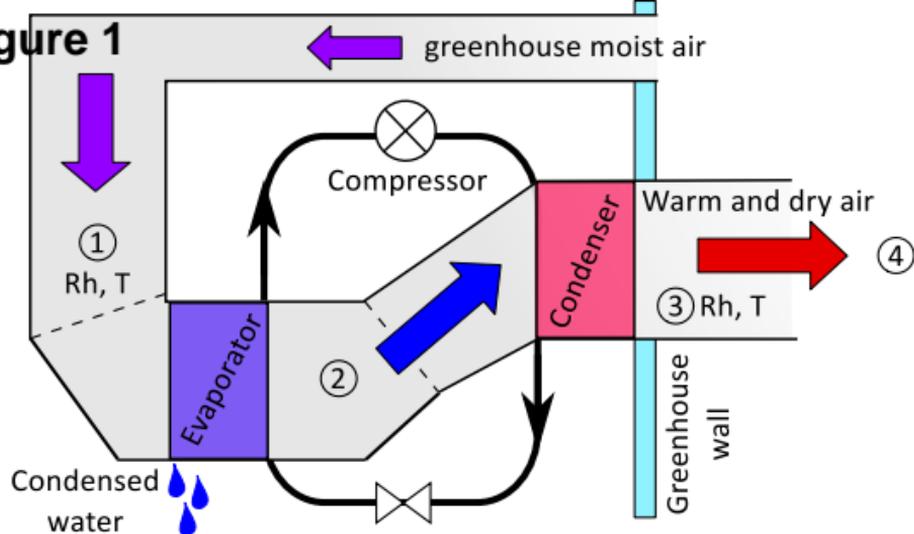


Figure 2

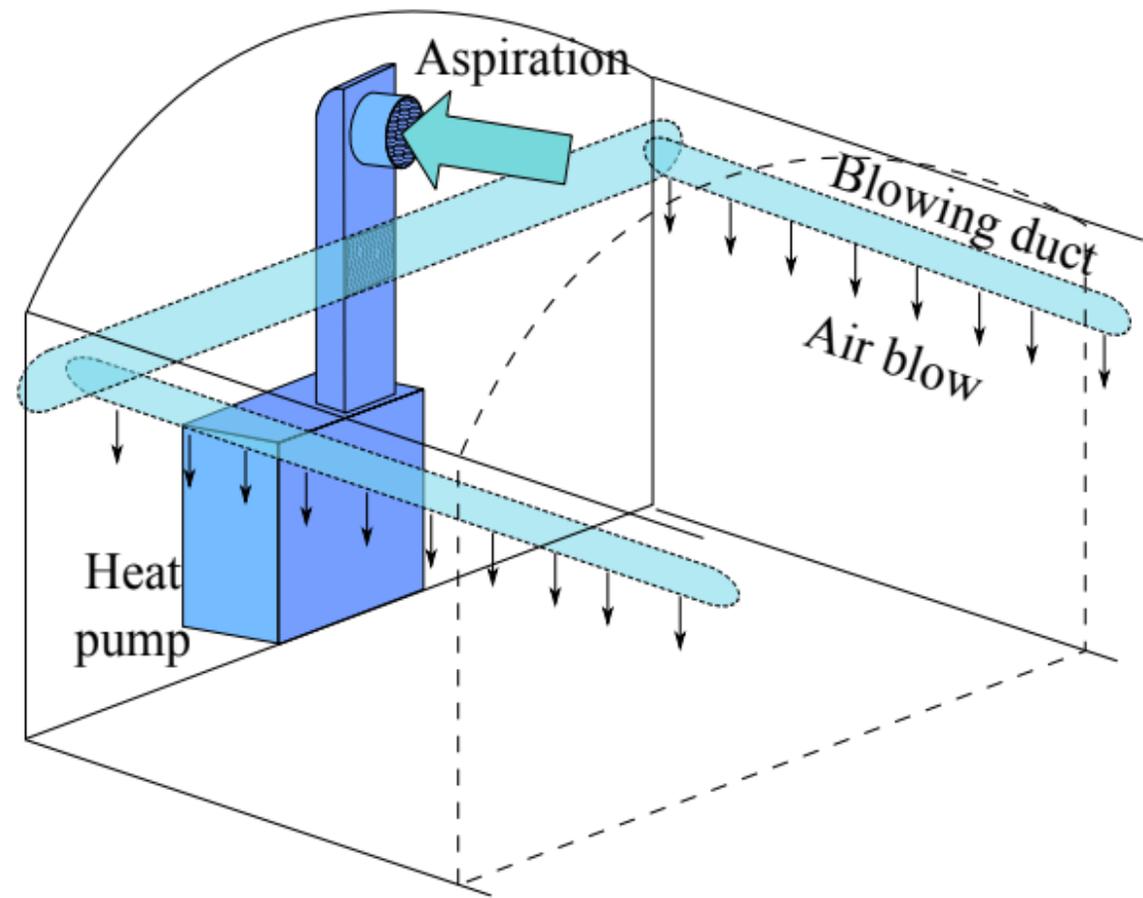
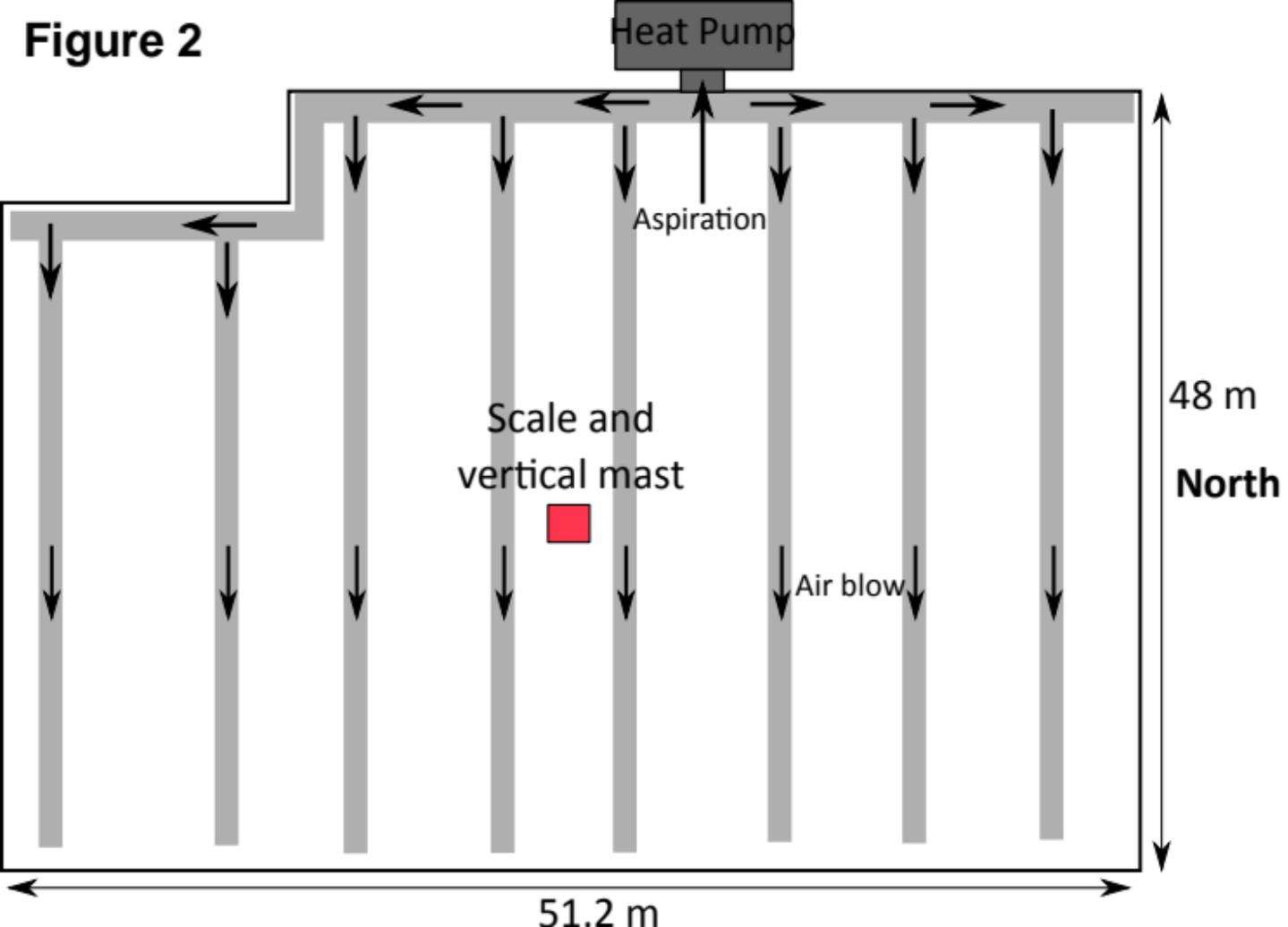


Figure 3

central mast for
vertical homogeneity

4.82 m T_a, Φ

3.37 m T_a, Φ

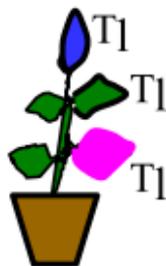
1.92 m T_a, Φ

0.47 m T_a, Φ



T_a Φ

T_a Φ

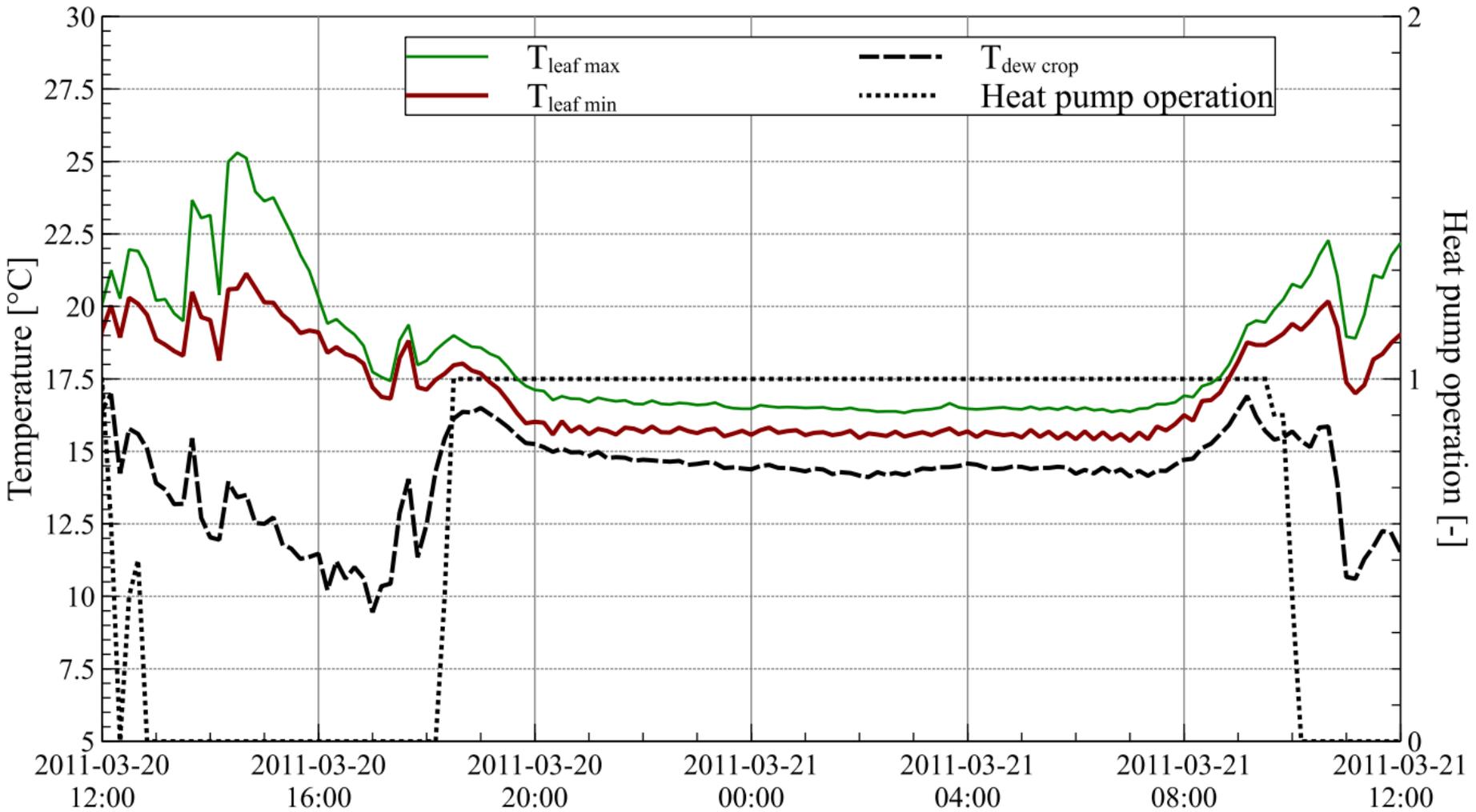


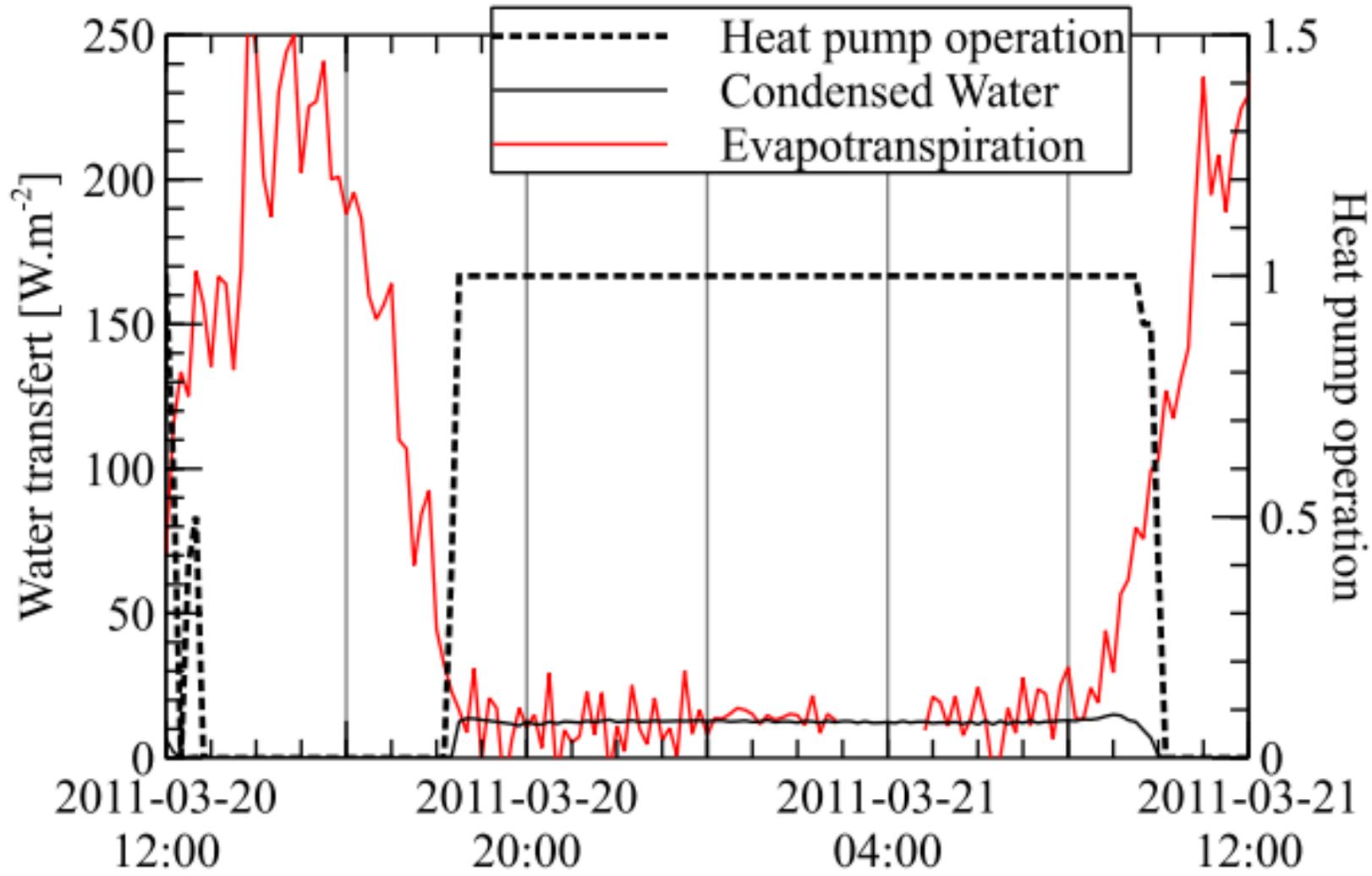
T_1

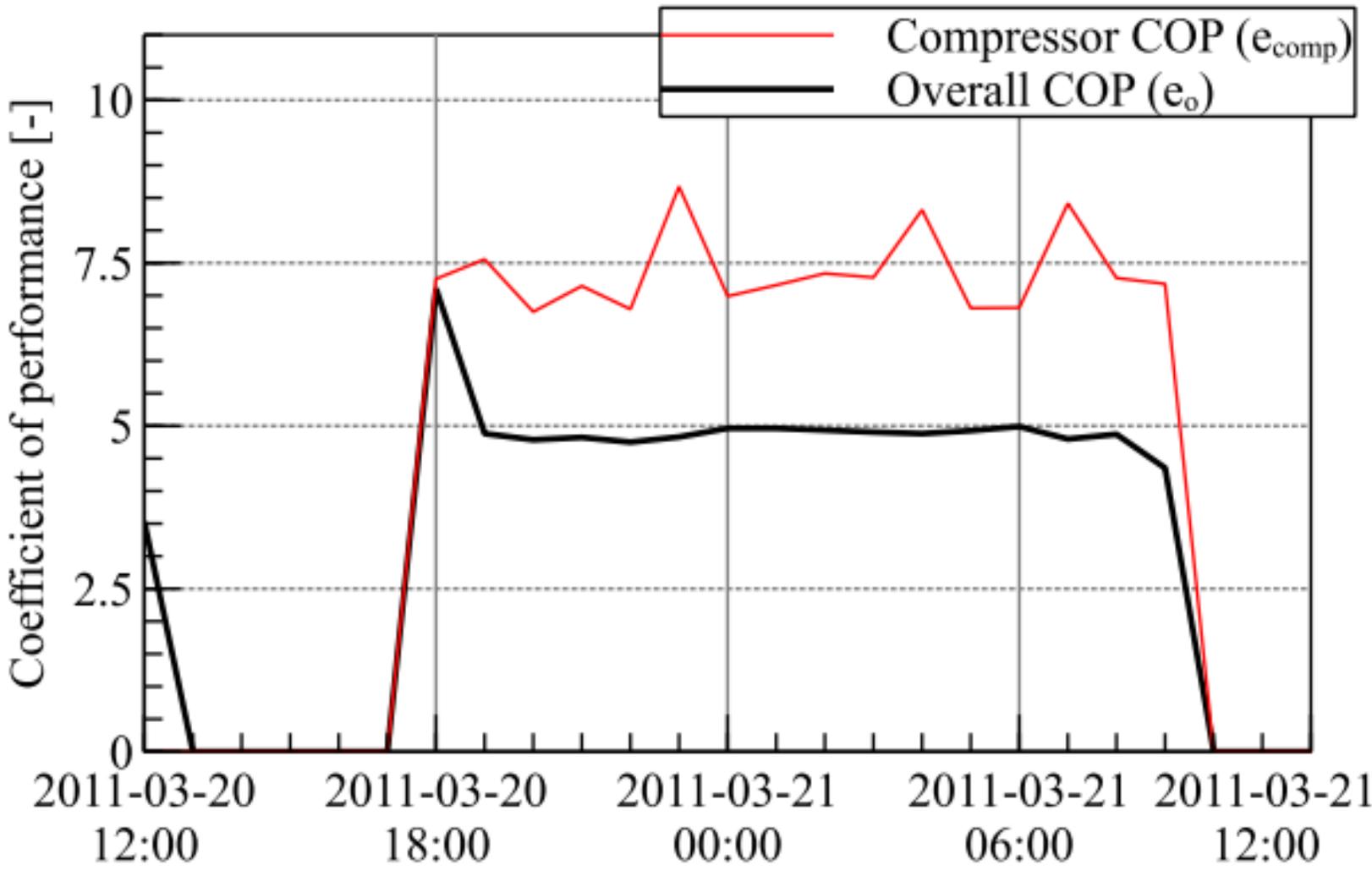
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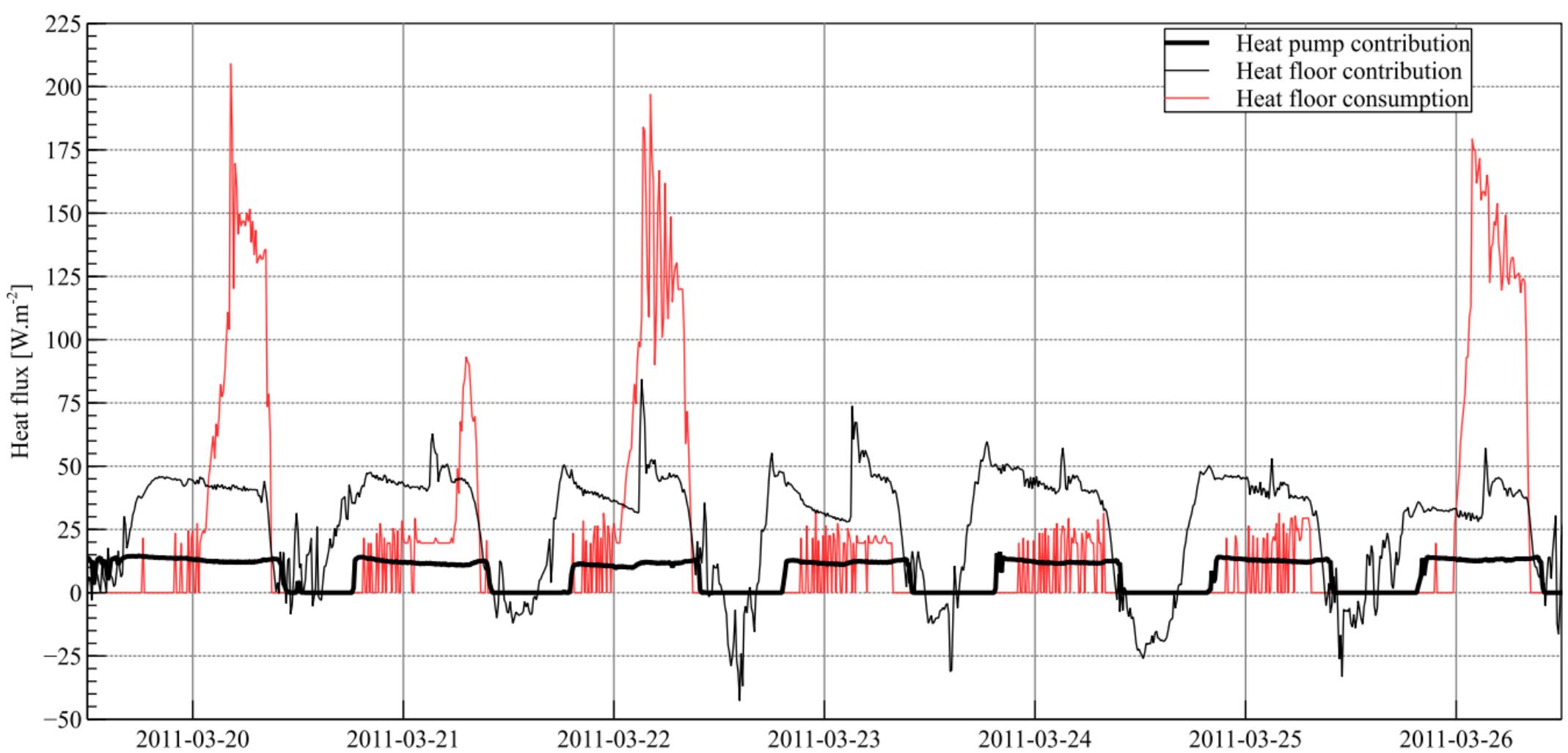
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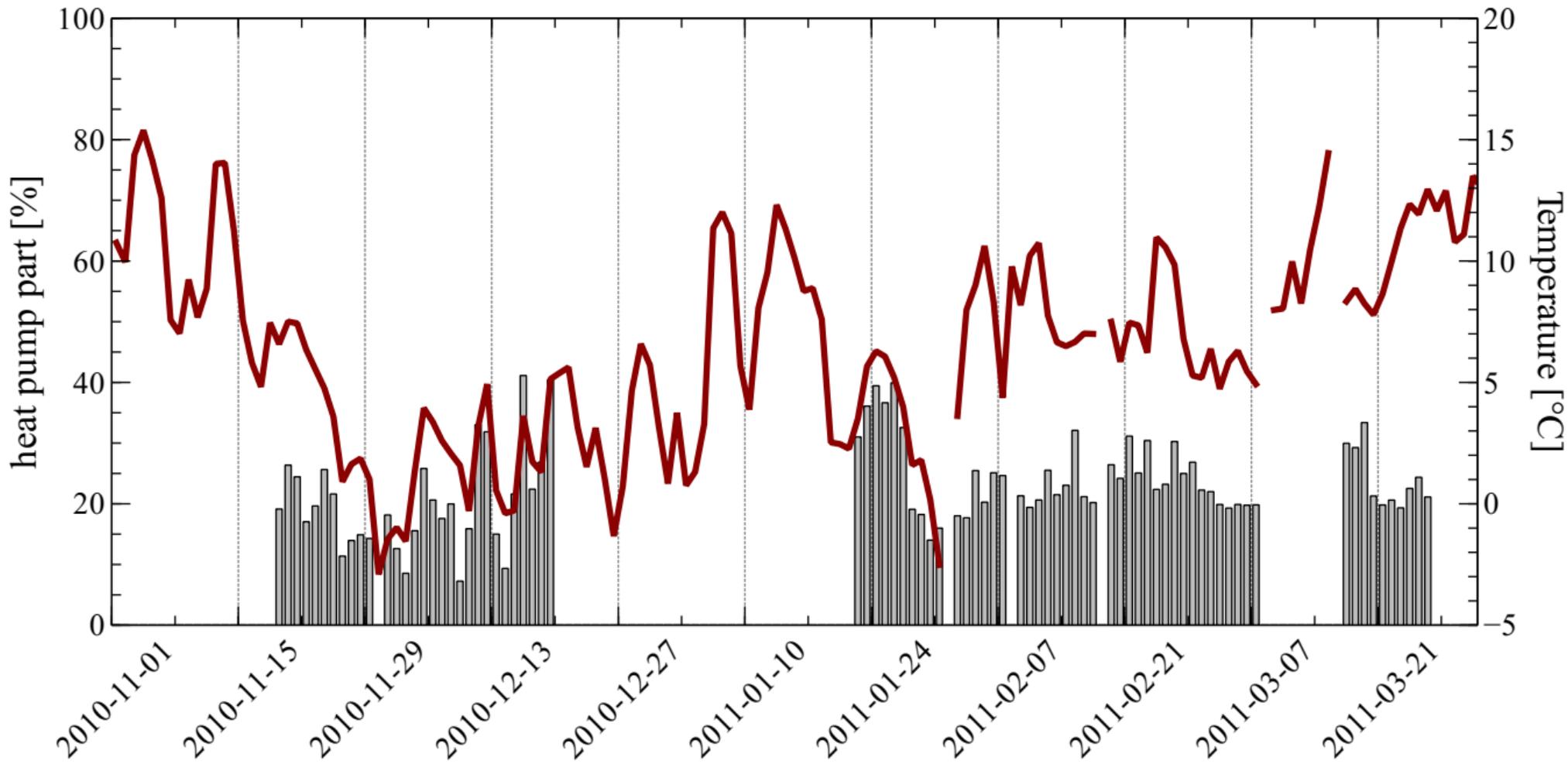
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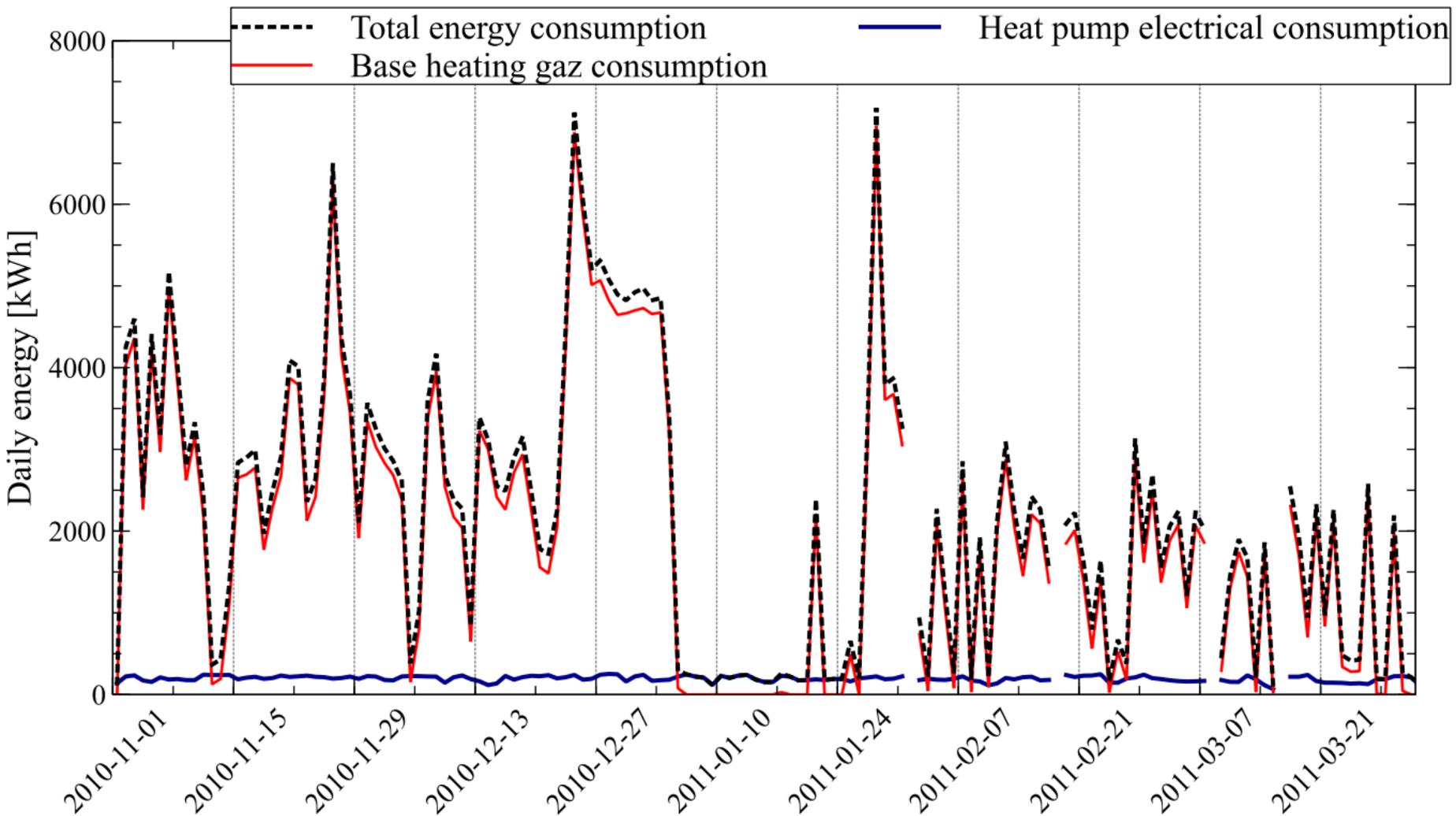


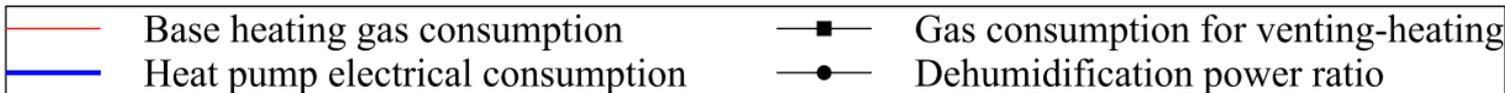
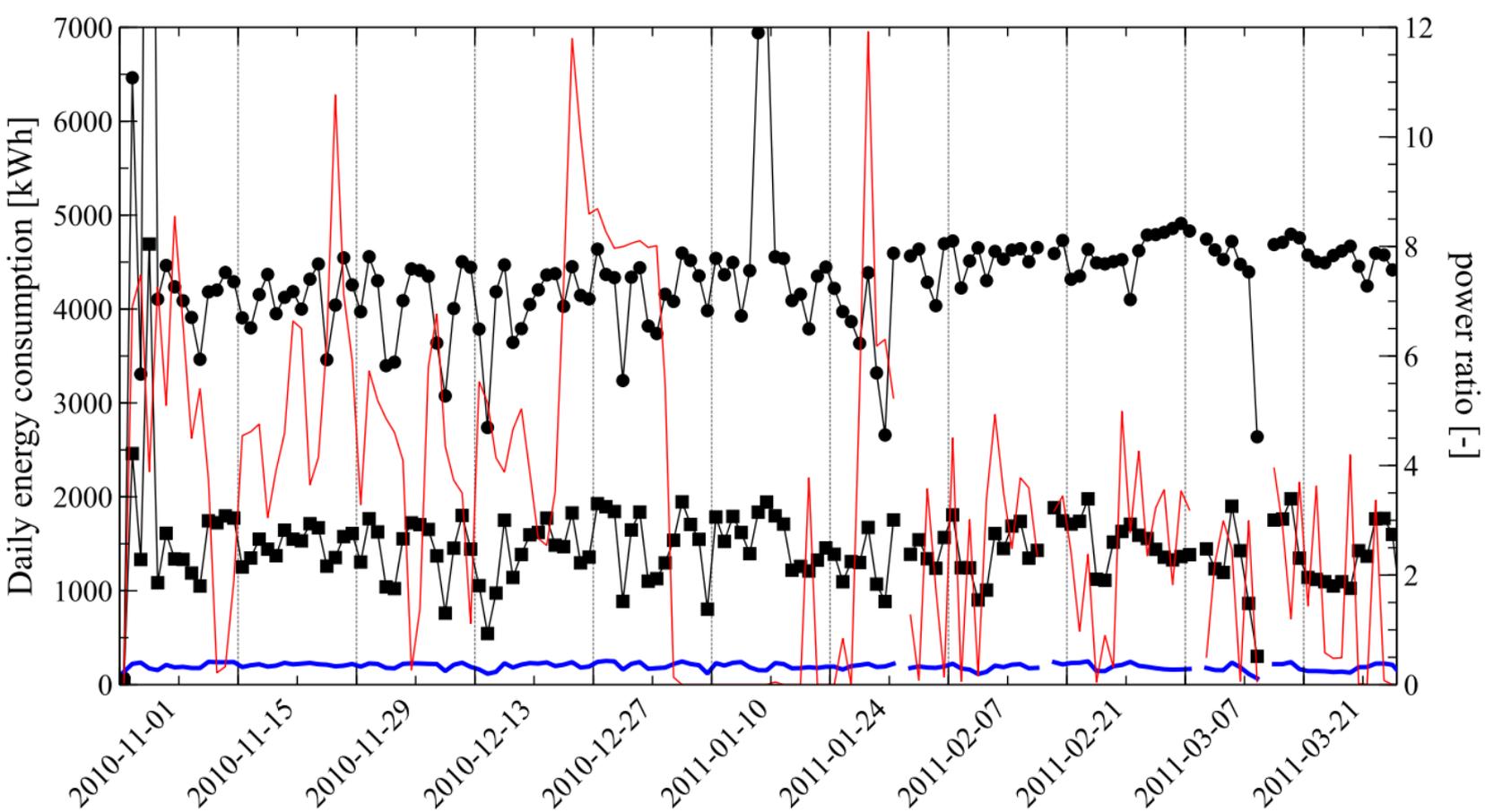


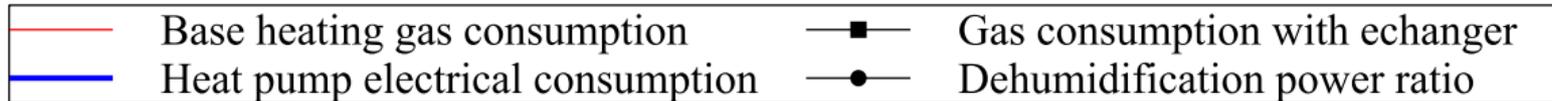
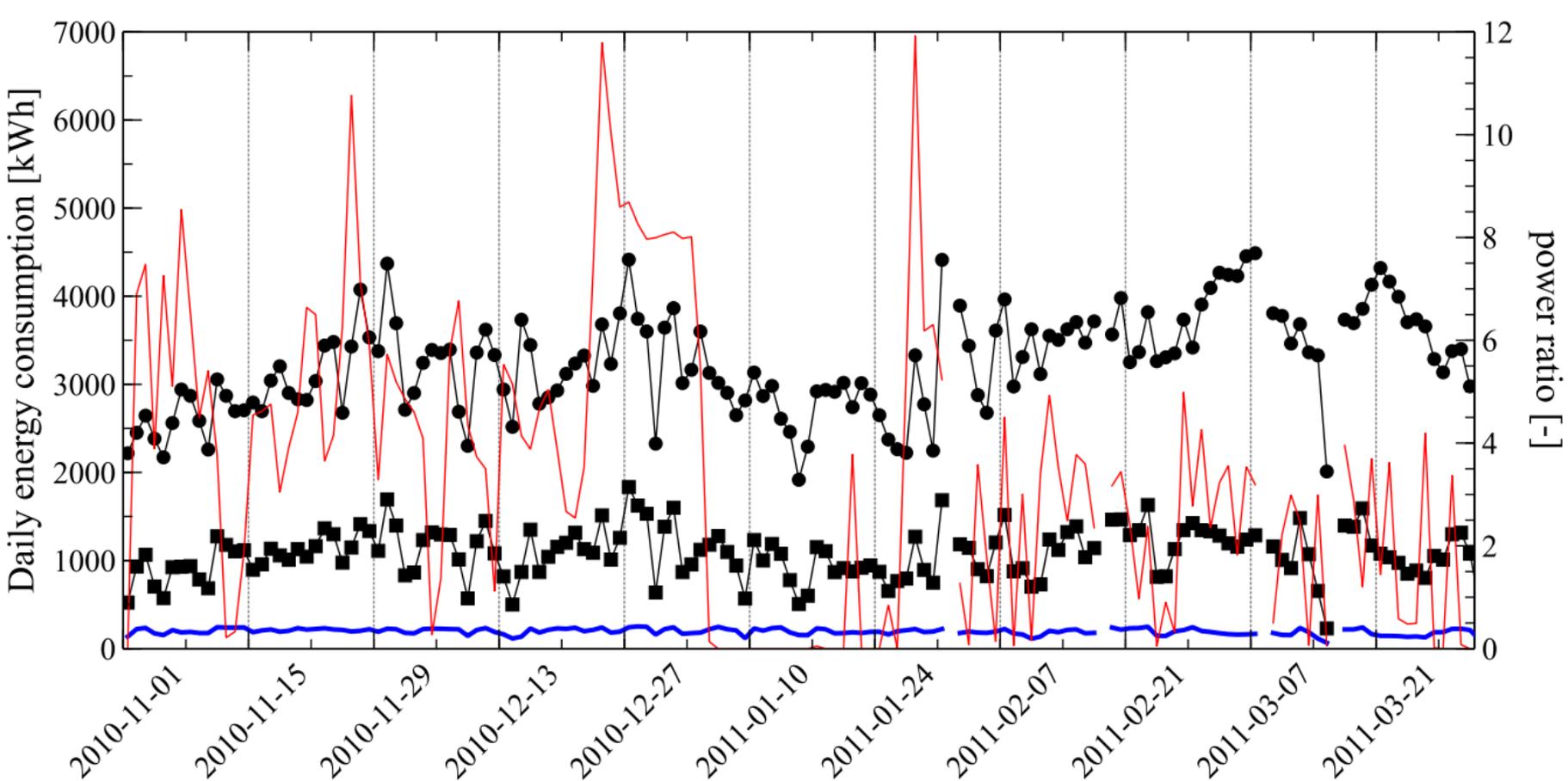


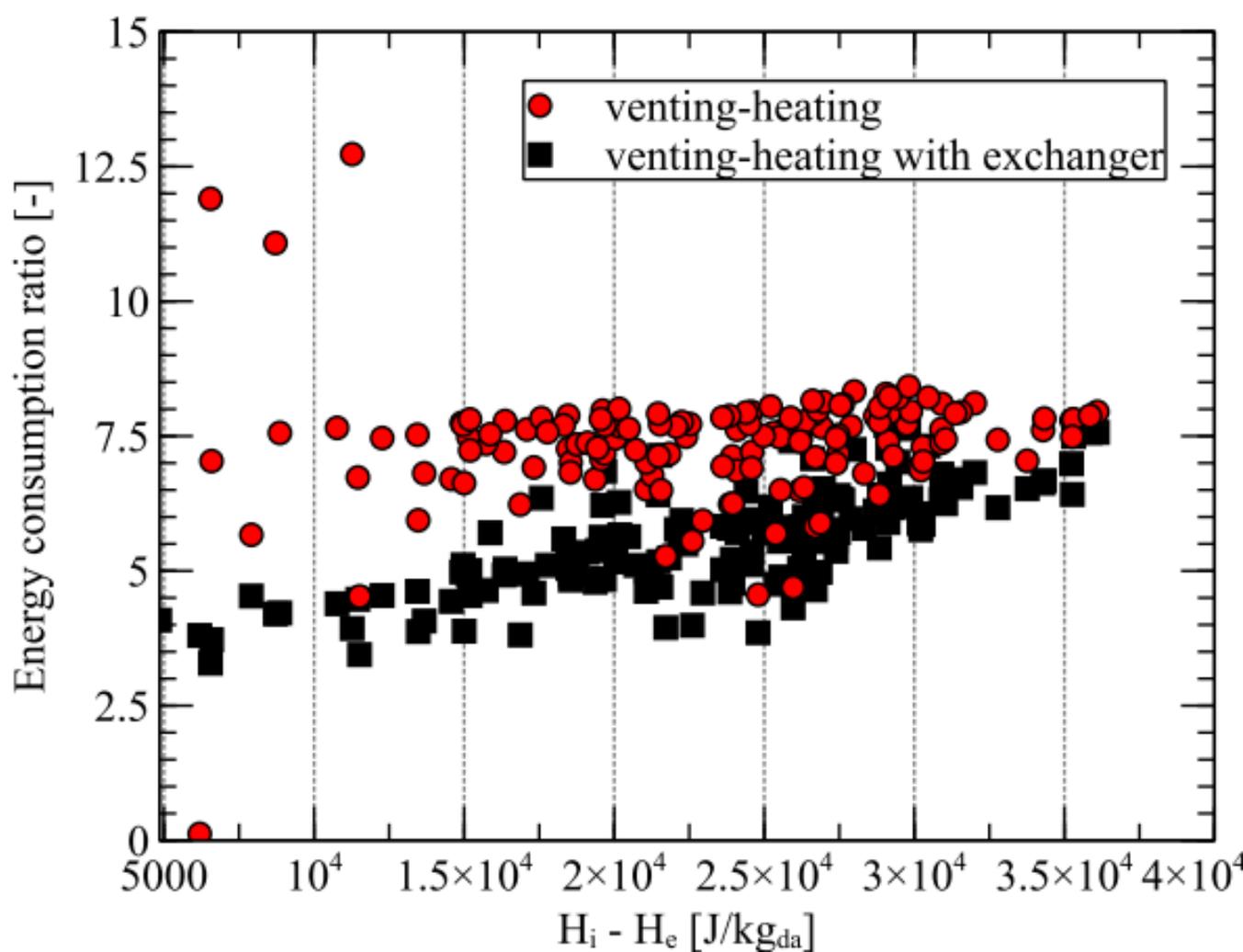












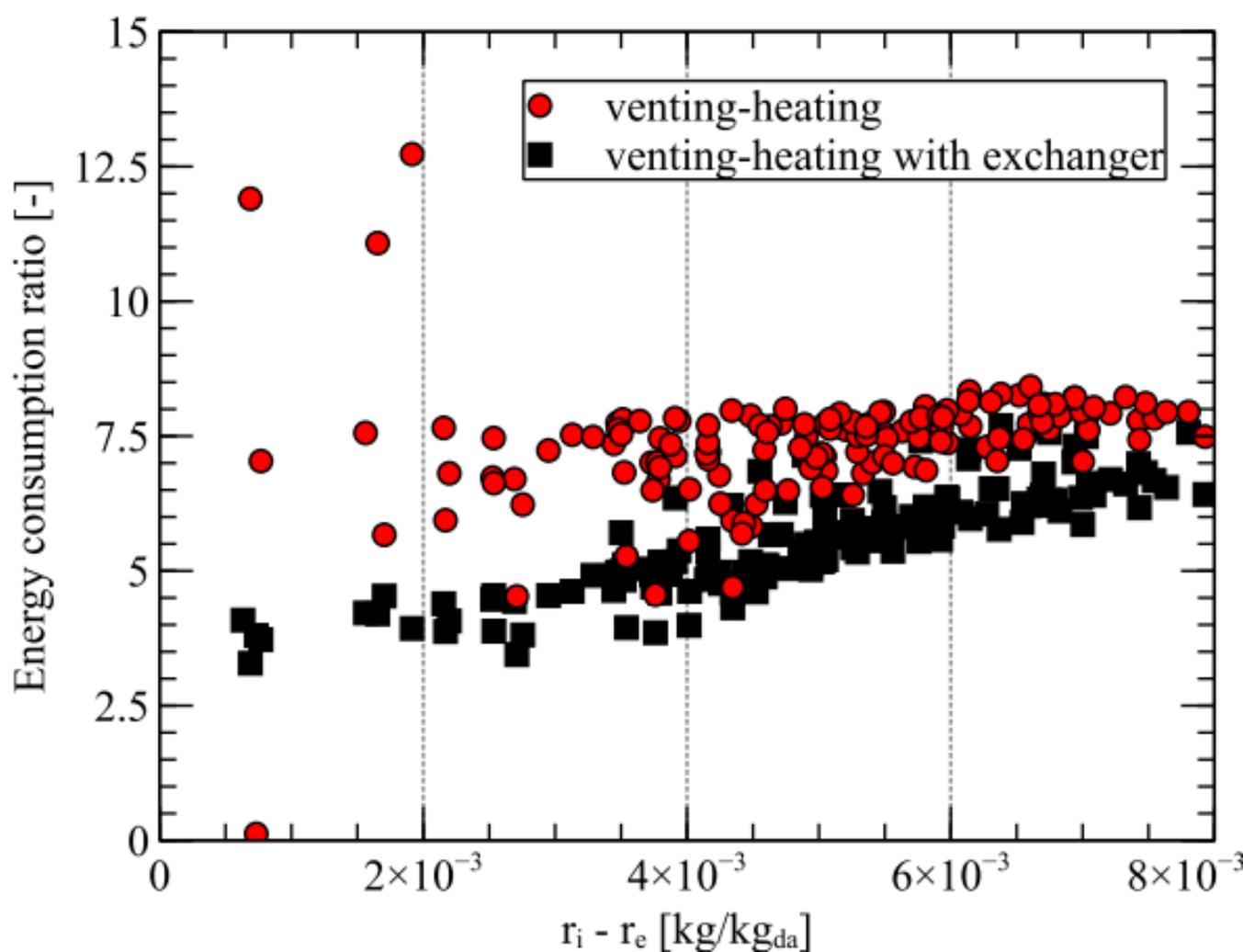


Figure 14
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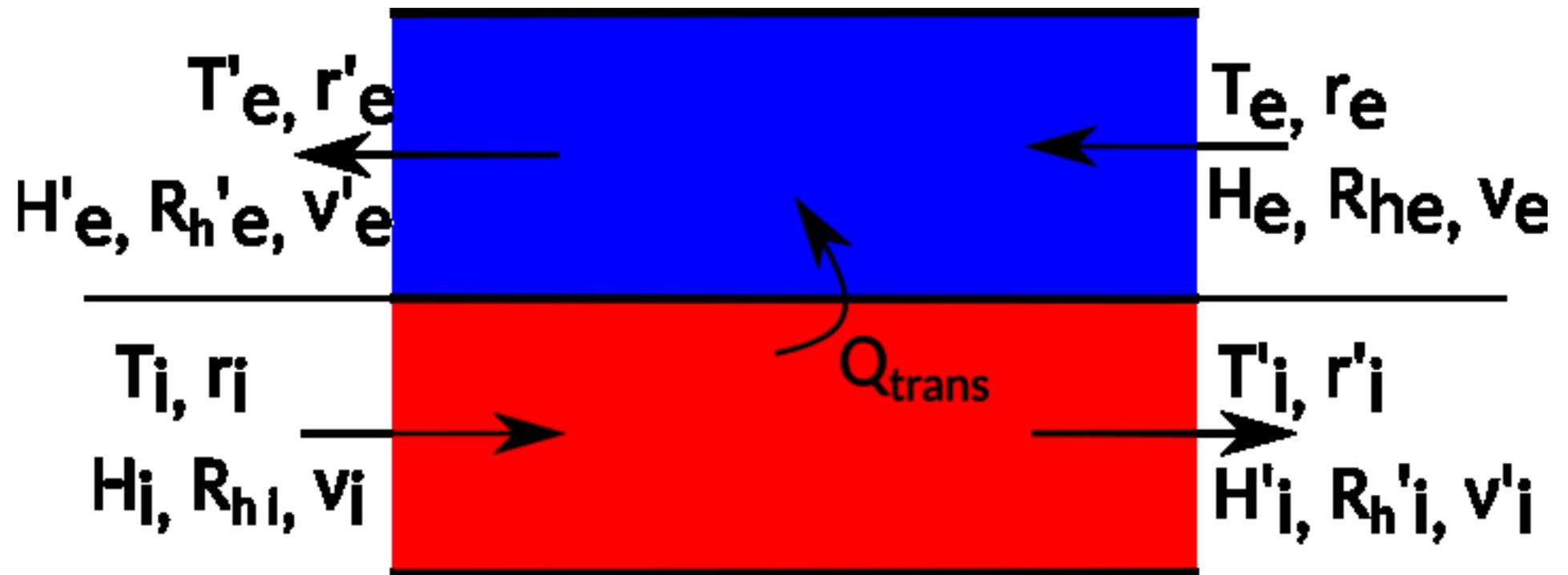


Table1

Country	Natural gas prices US\$/MWh	Electricity prices US\$/MWh	Price ratio gas/electricity	Minimum power consumption ratio for cost equivalence
Canada (2010 prices)	13.8	69.9	0.20	5.1
France	51.5	121.5	0.42	2.4
Germany	54.4	157.2	0.35	2.9
Japan	70.3	179	0.39	2.5
South Korea (2009 prices)	41.2	57.8	0.71	1.4
Mexico (2008 prices)	37.2	126	0.29	3.4
Netherland	38.5	118.5	0.32	3.1
Spain	37.7	148.8	0.25	3.9
United Kingdom	35.5	127.4	0.28	3.6