A Computer Model for Assessing Dew/Frost Surface Deposition

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Abstract: Using simple meteorological data, we were able to model the amount of nocturnal dew or frost deposited on a given surface thermally isolated from the ground. The surface temperature can be obtained either. The data needed are: ambient temperature and humidity, wind velocity at a given height (e.g. at 10 m from the ground or at 0.1 m from the deposition surface) and cloud cover. The condenser parameters that are needed for the simulation include the dimensions, density, emissivity and specific heat. We show that the dew amount at Ajaccio, Corsica island (France) can be predicted within three adjustable parameters that do not vary with time and which are determined by calibration tests. The developed interactive computer applications will be available from `http://www.opur.u-bordeaux.fr`.

1 BASIC EQUATIONS

The mathematical model is based on the models by Pedro and Gillespie [1982] and Nikolayev *et al.* [1996]. The heat balance equation for the condenser is

$$\frac{dT_c}{dt}(Mc_c + mc_w) = R_i + R_{he} + R_{cond}, \qquad (1)$$

where T_c is the condenser's temperature, M and m are the masses of the condenser and condensed water respectively, c_c and c_w are the specific heats of the material of the condenser and water; t is time. Hereafter, SI units are supposed for all the values except the temperature which is expressed in Celsius degrees. The variables in the right-hand side represent the different physical processes involved in the heat energy coming to or leaving the condenser surface: R_i is the irradiation, R_{he} is the heat exchange with the surrounding air, R_{cond} is the energy gain due to the latent heat of the condensation (L per unit mass). Thus

$$R_{cond} = L\frac{dm}{dt}. \qquad (2)$$

The convective heat exchange term can be expressed in the usual form as

$$R_{he} = S_c a(T_a - T_c), \qquad (3)$$

where a is a heat transfer coefficient, T_a is the ambient temperature. S_c is the condenser's surface area. a relates to the width of the aerodynamical boundary air layer and thus depends on the wind speed u:

$$a = kf\sqrt{u/D}, \qquad (4)$$

in which the numerical factor $f = 4\,\mathrm{WK^{-1}m^{-2}s^{1/2}}$ is empirical (Pedro and Gillespie [1982]) for the flow parallel to a plane sheet of size $D = \sqrt{S_c}$. We introduced here a correction coefficient k that depends on the relative position of the condenser with respect to the device measuring the wind velocity, and on the particular air flow conditions.

The total irradiation term from Eq.1 can be divided into several parts:

$$R_i = R_s + R_l - R_c. \qquad (5)$$

R_s is the sun-induced (direct+scattered) irradiation that we do not consider in the following because the dew forms mainly during the night. R_i is the incoming long-wave irradiation and R_c is the outgoing irradiation of the condenser. It can be represented

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$$R_c = S_c \varepsilon_c \sigma (T_c + 273)^4, \tag{6}$$

where σ is the Stephan-Boltzmann constant and ε_c is the emissivity of the condenser. The long-wave radiation term is given by Pedro and Gillespie [1982] and Campbell [1977]:

$$R_l = S_c \varepsilon_c \varepsilon_s \sigma (T_a + 273)^4, \tag{7}$$

where ε_s is the emissivity of the sky, which depends on the ambient temperature T_a and on the fraction N of the sky covered by clouds:

$$\varepsilon_s = \varepsilon_{s0} + N \left(1 - \varepsilon_{s0} - \frac{8}{T_a + 273}\right), \tag{8}$$

where $\varepsilon_{s0} = 0.72 + 0.005 T_a$.

The equation for m represents the condensation rate:

$$\frac{dm}{dt} = \begin{cases} S_c b(p_{sat}(T_d) - p_c(T_c)), & \text{if positive,} \\ 0, & \text{otherwise.} \end{cases} \tag{9}$$

Here $p_{sat}(T)$ is the saturation pressure at given temperature T, T_d being the dew temperature that can be determined from the equation $H\,p_{sat}(T_a) = p_{sat}(T_d)$, where H is the relative humidity. $p_c(T_c)$ is the vapor pressure over the condenser at the temperature T_c, at which condensation on its surface begins. Generally speaking, $p_c(T_c)$ does not coincide with $p_{sat}(T_c)$ and depends on the degree of wetting of the surface by the water, see Beysens *et al.* [1991]. When the surface is wetted, the condensation on it can begin even when $T > T_d$. We assume that it begins when $T = T_d - T_0$, i.e. $p_c(T_c) = p_{sat}(T_c + T_0)$, where $T_0 < 0$ is another fitting parameter that depends on the wetting conditions of the condenser surface and that does not vary with time. We take $T_0 = -0.35$°C in the following. Eq. 9 assumes the absence of evaporation of the already condensed water as if it were removed from the condenser as soon as condensation has stopped.

The value of the mass transfer coefficient b is proportional to a from (4):

$$b = 0.656 g a/(p c_a), \tag{10}$$

where p is the atmospheric pressure (assumed constant) and c_a is the specific heat of air. This expression as well as the numerical factor comes from the calculations by Pedro and Gillespie [1982]. We added here another adjustable time-independent parameter g to account for the particular air flow conditions around the condenser.

Eqs. (1) and (9) form a set of ordinary differential equations that is integrated for each night of observations separately. The initial time for the calculations is chosen somewhere after sunset, before the condensation starts, so that $m = 0$ initially. The ending time for the calculation should be chosen before sunrise.

2 DATA FITTING

The data used were those obtained by Muselli and Beysens [2001] (in this book). The data acquisition was entirely computer controlled, the resulting data files having a uniform format. This feature allowed the automatic data processing to be performed. Several Windows applications were written. First of them extracts the data to be input to the simulation to another data file. This latter file contains 15 min periodicity data for u, N, T_a, H, used as input for the simulation. The measured condenser temperature $T_{c,exp}$ and mass of condensed water m_{exp}, are also a part of this file in order to be fitted. The N data were not acquired but taken from the nearby (10 km) meteorological station and inserted into this data file by using another program. The fitting application receives the data from this file and another auxiliary file that contains the parameters of the condenser shown in Table 1. The purpose of the fitting

Parameter	*Notation*	*Value*
Emissivity	ε_c	0.94
Specific heat	c_c	1674 J/kgK
Thickness	–	5 mm
Density	–	1190 kg/m^3
Surface	S_c	0.16 m^2

Table 1: Condenser parameters which were used for the calculations. The condenser plate was made of Plexiglas (PMMA).

procedure is to obtain the values for two parameters k and g. The least squares method was used for the fitting. The fitting is performed in two stages. First, a value for g is guessed. This value influences directly the mass of the condensed water m. Since $m \ll M$, the influence of the m evolution (and g) on T_c is very small. Therefore, the error in g has very little effect on the T_c calculation. We note that it is for the same reason that we can neglect the difference between the condensation and sublimation latent heat in (2). Therefore the frost mass can be assessed by the same procedure as the dew amount.

By minimizing the difference between T_c and $T_{c,exp}$ we obtain a value for k. This value is used on the second stage where we minimize the difference between m and m_{exp} by adjusting g. The value of $T_{c,exp}$ at the initial moment of time is used as the initial condition for (1). The interactive Windows 98/NT applications will be available from the internet site of the International Organization for Dew Utilization (OPUR, `http://www.opur.u-bordeaux.fr`) together with the examples of the data files.

3 RESULTS

We analyzed the period from September 1, 1999 to January 1, 2001. The results of this analysis are

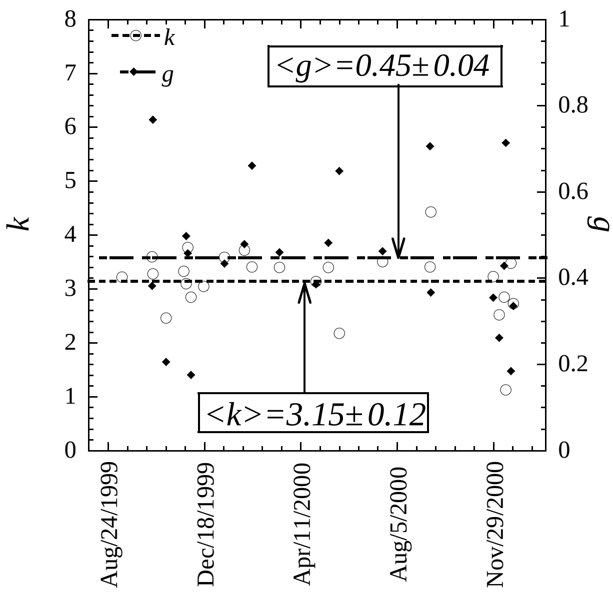

Figure 1: The results of calculations of the values for k and g. Their average values over the 16 months are shown.

presented in Fig. 1. One can see that the deviations from the average values $\langle k\rangle = 3.15$ and $\langle g\rangle = 0.45$ are not very large and do not vary with time. This proves that the choice of these parameters as adjustment parameters is good. Both k and g deviate from unity, which means that it is necessary to perform an adjustment.

A medium quality data fit is presented in Fig. 2 for the sake of discussion. The T_c fit is drawn by the solid line. It shows a sharp drop around 23h when the wind was absent (cf. the wind velocity curve, the dashed line). At the same time, the experimental curve shows only a slight decrease. The calculated temperature drops because the outgoing irradiation R_c is large while the heat exchange R_{he} that heats the condenser becomes zero. Indeed, this fit failure is due to both the poor sensitivity of the wind velocity data (velocities < 0.7 m/s result in $u = 0$) and an error in the correlation (4) for the convective heat transfer coefficient. According to (4), there is no convective heat exchange when $u = 0$, which is apparently wrong. This inconsistency appears be-

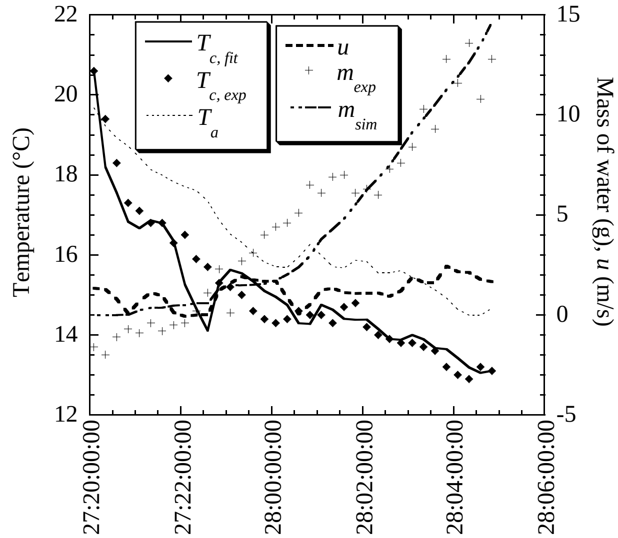

Figure 2: The data fit for the night of May, 27–28, 2000. The time format is dd:hh:mm:ss.

cause Eq. 4 is valid only for the forced convection (large u). When u is small, the natural convection becomes the dominant mechanism of the convective heat transfer. This shows the necessity to correct (4) in order to account for the natural convection and to have more sensitive u measurement.

Although the calculated condenser temperature rebounds quickly when the wind restarts, it cannot reproduce the experimental data immediately after this artificial temperature drop. As a result, the mass cannot be fitted correctly. For this reason, we could not fit properly the data during the days where wind was weak for long periods of time.

The influence of the wind on the mass measurement can be seen in Fig. 2. In spite of the correct calibration of the balance, the measured mass (crosses) can be negative. This inconsistency is due to wind-induced lifting force that acts on the condenser plate and creates the "negative mass" effect. While the upper part of the plate is open, its lower part is protected by a metal box that confines the balance, see Fig. 1 in Muselli and Beysens [2001] (in this book). The difference of the air velocities above and below the plate creates a lifting force similar to that used by airplane wing. The correlation of the negative mass and the wind velocity can be seen in Fig. 2 between 20 and 22h.

A turbulent air motion caused by frequently varying wind velocity over such a condenser creates sometimes spurious balance oscillations that can manifest themselves in an artificial increase or decrease of the measured mass. This is illustrated in Fig. 3, where the apparently 'wrong' mass measurements correlate

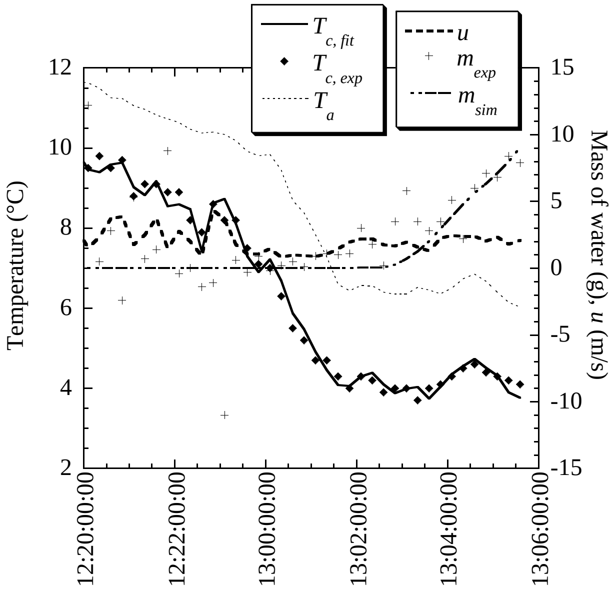

Figure 3: The data fit for the night of February, 12–13, 2000.

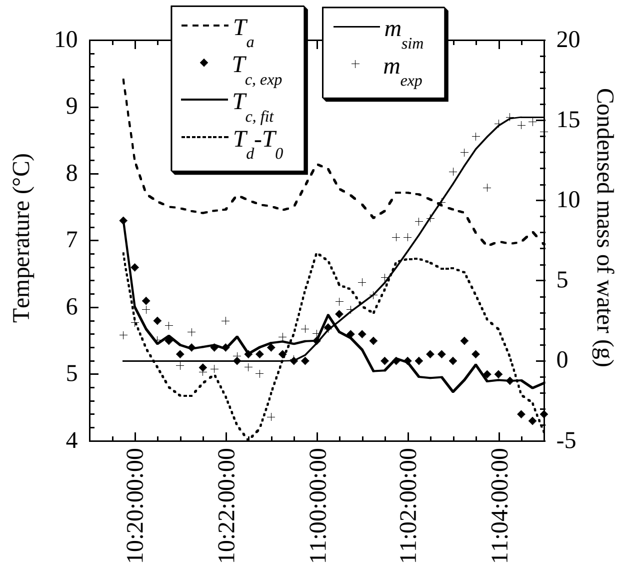

Figure 4: The data fit for the night of January, 10–11, 2000.

with the wind oscillations. While the temperature fit ($k = 3.41$) itself is very good due to the persistent wind, it is quite difficult to infer the actual value of the condensed mass. An abnormally high value for $g = 0.661$ follows.

A fit to other data calculated with $k = 3.59$ and $g = 0.434$ is presented in Fig. 4. The condensation begins at about 23h and ends at about 4h30. During this interval of time, T_c stays below the condensation temperature $T_d - T_0$. One can see that the condenser temperature stays several degrees cooler than air due to the radiation losses. The simulated temperature stays close to $T_{c,exp}$ during all the period until the end of the dew formation, the variance of $T_c - T_{c,exp}$ being less than 0.34°C. The fit to m is equally very good.

4 CONCLUSIONS

We elaborated an algorithm and a set of the PC software that permit to predict the amount of condensed water on the condenser plate. The model predicts also the evolution of the plate temperature during night. This software was verified against the experimental data obtained by Muselli and Beysens [2001] (in this book) during 16 months. As a rule, the variance of the difference between the measured and simulated temperatures of the condenser plate does not exceed 0.5°C. The model has three adjustable empirically parameters that depend on the position and material of the condenser. They are proved to be independent of time.

However, this model has some limitation. It does not work well when the wind is weak (less than 1 m/s) for a long period of time. Additional work is thus needed to improve this model.

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