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UNSTEADY STATES OF TEMPERATURES IN SURFACE COOLERS OF AIR

NIESTACJONARNE STANY TEMPERATUROWE W PRZEPONOWYCH CHŁODNICACH POWIETRZA

Work of mining surface coolers of air operating indirectly, where cooling medium is cold water, is discussed. Cases of both co-current and counter-current flow of both media through a cooler are taken into consideration.

Sets of equations describing changeable in time and along a cooler: temperature and specific humidity of cooled air and temperature of heat exchanger and cooling water are presented as well. Equations describing cooling of air, during which outdropping of water vapour contained in it does not occur and during which this process takes place, are given for each direction of flow of water. This way four different mathematical descriptions were obtained.

Conducted considerations are illustrated by two calculation examples, whose solutions are presented graphically.

Key words: mining, climatic conditions in mines, air cooling, surface coolers.

W artykule zajęto się oceną skuteczności działania górniczych przeponowych chłodnic powietrza o działaniu pośrednim, w którym czynnikiem chłodniczym jest zimna woda. Rozważono przypadki zarówno współprądowego, jak i przeciwprądowego przepływu obydwóch mediów, tj. powietrza i wody przez chłodnicę. Z uwagi na to, że kierunek przepływu wody przez chłodnicę ma istotne znaczenie dla efektu jej pracy, porównano w prezentowanym artykule wpływ charakteru jej pracy (współprądowa lub przeciwprądowa) na czasoprzestrzenne rozkłady temperatury i wilgotności właściwej chłodzonego powietrza oraz temperatury przepony wymiennika ciepła i temperatury wody chłodzącej, jak również na czasowe przebiegi tych wielkości. Uwzględniono chłodzenie suche, podczas którego nie dochodzi do skraplania pary wodnej w powietrzu, oraz chłodzenie mokre — ze skraplaniem pary. W ten sposób otrzymano cztery różne opisy matematyczne:

— dla suchego chłodzenia powietrza chłodnicą współprądową,

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— dla mokrego chłodzenia powietrza chłodnicą współprądową,
— dla suchego chłodzenia powietrza chłodnicą przeciwprądową,
— dla mokrego chłodzenia powietrza chłodnicą przeciwprądową.

Każdy z przypadków nie ustalonego chłodzenia suchego opisany jest układem równań różniczkowych cząstkowych pierwszego rzędu, natomiast przypadki nie ustalonego chłodzenia mokrego — dwoma układami równań różniczkowych cząstkowych pierwszego rzędu, powiązanych ze sobą warunkami granicznymi. Wyprowadzenie zamieszczonych tu równań było prezentowane we wcześniejszych pracach.

W celu zilustrowania omawianych zagadnień przedstawiono w pracy przykłady obliczeniowe, których rozwiązania podano w formie wykresów naniesionych wspólnie dla chłodnicy współprądowej i przeciwprądowej, co ułatwia ich porównywanie. Oddzielnie pokazane są wykresy dotyczące chłodzenia suchego, oddzielnie zaś odnoszące się do chłodzenia mokrego. Dla wszystkich badanych przypadków przedstawiono rozkłady wzdłuż chłodnicy temperatury ośrodków wymieniających ciepło i wilgotności właściwej powietrza dla różnych chwil czasowych oraz czasowe przebiegi tych wielkości dla wlotowego i wylotowego przekroju chłodnicy. Ponadto pokazano czasowe przebiegi temperatury i wilgotności powietrza schłodzonego, a dla chłodzenia mokrego dodatkowo przebieg położenia granicy stref chłodzenia suchego i mokrego w funkcji czasu.

Słowa kluczowe: górnictwo, klimatyzacja kopalń, chłodzenie powietrza, chłodnice przeponowe.

A LIST OF DESIGNATIONS USED IN THE TEXT

b — absolute pressure of air [Pa],
b_f — by-pass factor of a cooler [−],
c_c — specific heat of water [J/(kgK)],
c_p — specific heat of dry air at constant pressure [J/(kgK)],
c_t — specific heat of the material of a heat exchanger [J/(kgK)],
c_w — specific heat of water vapour at constant pressure [J/(kgK)],
F_{ch} — area of active cross-section of a cooler [m^2],
F_w — area of inside surface (from the side of water) of heat exchange in an exchanger [m^2],
F_z — area of outside surface (from the side of air) of heat exchange in an exchanger [m^2],
L — length of a cooler [m],
m_t — mass of a heat exchanger [kg],
s — current co-ordinate measured along a cooler in accordance with the direction of airflow [m],
s_w — boundary co-ordinate of zone, separating (during cooling with condensation of water vapour) the zone of dry cooling from the zone of wet cooling in a cooler [m],
T — time constant of an exponential falling of temperature of cooling water flowing into a cooler [s],
t_c — temperature of a cooled part of air [°C],
t_t — temperature of a heat exchanger [°C],
t_w — temperature of cooling water [°C],
t_{w2} — inflow temperature of cooling water in a co-current cooler [°C],
t_{w3} — inflow temperature of cooling water in a counter-current cooler [°C],
t_{w20} — initial (for $\tau = 0$) inflow temperature of cooling water in a co-current cooler [°C],
t_{w2u} — steady (for $\tau \to \infty$) inflow temperature of cooling water in a co-current cooler [°C],
t_{w30} — initial (for $\tau = 0$) inflow temperature of cooling water in a counter-current cooler [°C],
t_{w3u} — steady (for $\tau \to \infty$) inflow temperature of cooling water in a counter-current cooler [°C],
t_1 — temperature of air at the entry to a fan [°C],
t_2 — temperature of air at the entry to a cooler [°C],

t_3 — temperature of air at the outlet from a cooler, after cooled part gets mixed with non-cooled part [°C],
t_{pr} — temperature of dew point of air being cooled [°C],
x_c — specific humidity of cooled part of air [kg of vapour H_2O/kg of dry air],
$x_n(t)$ — specific humidity of air in a saturated state at temperature t [kg of vapour H_2O/kg of dry air],
x_1 — specific humidity of air at the entry to a fan [kg of vapour H_2O/kg of dry air],
x_2 — specific humidity of air at the entry to a cooler [kg of vapour H_2O/kg of dry air],
x_3 — specific humidity of air at the outlet from a cooler, after cooled part gets mixed with non-cooled part [kg of vapour H_2O/kg of dry air],
V_{cw} — capacity of pipes of a heat exchanger [m^3],
v — average velocity of airflow through a cooler [m/s],
v_{w0} — calculation velocity of flow of cooling water through a cooler [m/s],
α_w — co-efficient of taking up heat on the inside surface of an exchanger (from surface to water) [$W/(m^2K)$],
α_z — co-efficient of taking up heat on the outside surface of an exchanger (from air to surface) [$W/(m^2K)$],
ϱ — air density (related to dry air) [kg/m^3],
ϱ_w — water density [kg/m^3],
τ — time [s].

1. Introduction

Beside conventional methods for overcoming heat hazard present in mining galleries, artificial cooling of air, among others by means of surface coolers, is used more and more frequently. The shape of such coolers is usually of a channel with a heat exchanger installed in it. Cooling medium, cold water cooled outside a cooler, flows inside an exchanger, and air that is being cooled flows along its outside surface. The wall of an exchanger is a surface separating both media.

This work presents and compares two types of surface coolers operating indirectly: a co-current cooler and counter-current one. When the inflow of water is located close to the inflow of air and its outflow is located close to the outlet of air, both media flow in the same direction and in such a situation a cooler operates co-currently. (Fig. 1a). However, when water flows into a cooler from the side of the outflow of air and flows out close to its inflow, a cooler operates counter-currently. (Fig. 1b). As the direction of flow of water greatly influences its work, it was decided to discuss this problem in more detail.

Based on earlier works [Dziurzyński et al., 1999; Filek et al., 1998; Filek et al., 1999; Filek, Nowak, 2000; Holesz, 1997; Roszczynialski et al., 1992; Uhlig, 1982; Uhlig, 1983] sets of equations are presented here. These equations describe changeable in time (τ) distributions of temperature t_c and specific humidity x_c of cooled air (or rather its coolled part, which is connected with a notion of a by-pass factor b_f, discussed many times in the works quoted above) and also of temperatures of: a heat exchanger t_t and of cooling water t_w along a cooler, i.e. in the function of a co-ordinate s, whose axis was marked in figures 1a

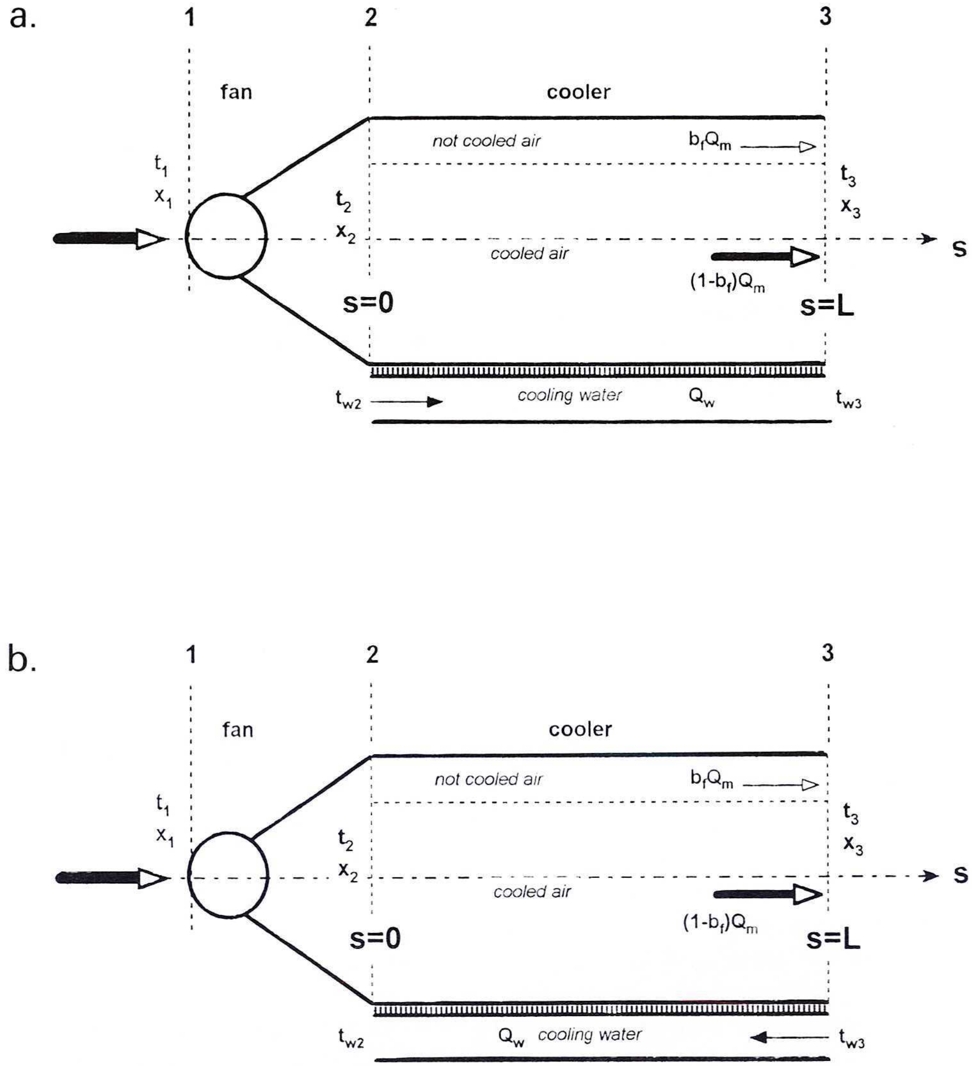


Fig. 1. The scheme of flow of cooled air and cooling water through a: a) co-current, b) counter-current cooler

and 1b. Two situations for each direction of flow of water were distinguished: dry cooling of air (without outdropping of water vapour contained in it) and wet cooling (with condensation of water vapour). This way four different mathematical descriptions were obtained:

— for dry cooling of air by means of a co-current cooler,
— fot wet cooling of air by means of a co-current cooler,
— for dry cooling of air by means of a counter-current cooler,
— for wet cooling of air by means of a counter-current cooler.

Conducted research is illustrated by calculation examples whose solutions are presented in the form of graphs, shown both for a co-current and counter-current cooler, which enables easy comparison.

Another significant, from practical point of view, problem of heat power of a surface cooler is going to be discussed in the next works.

2. Mathematical descriptions of cooling of air

Dry cooling of air can be described by one set of equations, comprising the whole length of a cooler. However, when outdropping of water occurs, a cooler is divided into two parts: zone I comprising the range of co-ordinate s from the entry to a cooler ($s = 0$) to the boundary cross-section ($s = s_w$), where condensation of water vapour does not occur and zone II from $s = s_w$ to $s = L$, where outdropping of vapour occurs.

While forming initial and boundary conditions for equations, it was assumed that:

- state of air at the entry (temperature t_1, specific humidity x_1 and pressure b) and also its mass flow rate Q_w are steady in time;
- temperature of cooling water flowing into a cooler falls exponentially according to (3) or (9), but its mass flow rate Q_w is steady;
- initial instant ($\tau = 0$) is determined by switching on an unit cooling water in a situation when air of steady temperature t_2 and specific humidity x_2, and also water of steady temperature t_{w20} or t_{w30} flows into a cooler.

Sets of equations, based on works [Holesz, 1997; Filek et al., 1998; Filek et al., 1999; Filek, Nowak, 2000] presenting mathematical models for different cases of cooling air, are given for situations under consideration.

A co-current cooler, dry cooling

The following set of equations is required:

$$\begin{cases} \dfrac{\partial t_c}{\partial \tau} + v\dfrac{\partial t_c}{\partial s} = -\dfrac{\alpha_z F_z}{\varrho F_{ch} L(1-b_f)(c_p + c_w x_2)}(t_c - t_t) \\ \dfrac{\partial t_t}{\partial \tau} = \dfrac{\alpha_z F_z}{m_t c_t}(t_c - t_t) - \dfrac{\alpha_w F_w}{m_t c_t}(t_t - t_w) \\ \dfrac{\partial t_w}{\partial \tau} + v_{w0}\dfrac{\partial t_w}{\partial s} = \dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c}(t_t - t_w) \\ x_c = x_2. \end{cases} \tag{1}$$

Boundary conditions for the equations above take the following form:

$$t_c(s = 0, \tau) = t_2 = \text{const} \tag{2}$$

$$t_w(s = 0, \tau) = t_{w2}(\tau) = (t_{w20} - t_{w2u})e^{-\tau/T} + t_{w2u}. \tag{3}$$

Moreover, initial conditions are determined based on the following set of equations:

$$\begin{cases} \dfrac{dt_c}{ds} = -\dfrac{\alpha_z F_z}{(1-b_f)(c_p + c_w v_\varrho F_{ch} L x_2)}(t_c - t_t) \\ \alpha_z F_z (t_c - t_t) = \alpha_w F_w (t_t - t_w) \\ \dfrac{dt_w}{ds} = \dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c v_{w0}}(t_t - t_w), \end{cases} \tag{4}$$

where boundary conditions are as follows:

$$t_c(s=0) = t_2 \tag{5}$$

$$t_w(s=0) = t_{w20}. \tag{6}$$

A counter-current cooler, dry cooling

A set of equations for a counter-current cooler differs from the one for cooling by means of a co-current cooler in a sign on the left side of the last equation referring to the temperature of water. This set takes the following form:

$$\begin{cases} \dfrac{\partial t_c}{\partial \tau} + v\dfrac{\partial t_c}{\partial s} = -\dfrac{\alpha_z F_z}{\varrho F_{ch} L(1-b_f)(c_p + c_w x_2)}(t_c - t_t) \\ \dfrac{\partial t_t}{\partial \tau} = \dfrac{\alpha_z F_z}{m_t c_t}(t_c - t_t) - \dfrac{\alpha_w F_w}{m_t c_t}(t_t - t_w) \\ \dfrac{\partial t_w}{\partial \tau} - v_{w0}\dfrac{\partial t_w}{\partial s} = \dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c}(t_t - t_w) \\ x_c = x_2. \end{cases} \tag{7}$$

Boundary conditions for the equations above take the following form:

$$t_c(s=0,\tau) = t_2 = \text{const} \tag{8}$$

$$t_w(s=L,\tau) = t_{w3}(\tau) = (t_{w30} - t_{w3u})e^{-\tau/T} + t_{w3u}. \tag{9}$$

Initial distributions of temperatures $t_c(s)$, $t_t(s)$ and $t_w(s)$ are determined based on the following set of equations:

$$\begin{cases} \dfrac{dt_c}{ds} = -\dfrac{\alpha_z F_z}{(1-b_f)(c_p + c_w x_2)}(t_c - t_t) \\ \alpha_z F_z (t_c - t_t) = \alpha_w F_w (t_t - t_w) \\ \dfrac{dt_w}{ds} = -\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c v_{w0}}(t_t - t_w) \end{cases} \tag{10}$$

with boundary conditions:

$$t_c(s=0)=t_2 \tag{11}$$

$$t_w(s=L)=t_{w30}. \tag{12}$$

A co-current cooler, wet cooling

As it was emphasised before, outdropping of water vapour carried by cooled air in a cooler occurs during wet cooling. This phenomenon takes place only in zone II of a cooler, where air is cooled enough. As there are differences in equations for each zone, searched values were additionally marked with „prim" in zone I and with „bis" in zone II in order to avoid misunderstanding.

In such a case, a mathematical description consists of two sets of equations connected with each other by means of boundary conditions.

In zone of dry cooling (I) — $0 \leqslant s \leqslant s_w$:

$$\begin{cases} \dfrac{\partial t_c'}{\partial \tau}+v\dfrac{\partial t_c'}{\partial s}=-\dfrac{\alpha_z F_z}{\varrho F_{ch} L(1-b_f)(c_p+c_w x_2)}(t_c'-t_t') \\ \dfrac{\partial t_t'}{\partial \tau}=\dfrac{\alpha_z F_z}{m_t c_t}(t_c'-t_t')-\dfrac{\alpha_w F_w}{m_t c_t}(t_t'-t_w') \\ \dfrac{\partial t_w'}{\partial \tau}+v_{w0}\dfrac{\partial t_w'}{\partial s}=\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c}(t_t'-t_w') \\ x_c'=x_2 \end{cases} \tag{13}$$

and in zone of wet cooling II — $s_w \leqslant s \leqslant L$:

$$\begin{cases} \left[c_p+c_w x_c''+(r_p+c_w t_c'')\Phi\right]\dfrac{\partial t_c''}{\partial \tau}+v\left[c_p+c_w x_c''+(r_p+c_w t_c''-c_c t_c'')\Phi\right]\dfrac{\partial t_c''}{\partial s}= \\ \qquad =-\dfrac{\alpha_z F_z}{\varrho F_{ch} L(1-b_f)}(t_c''-t_t'') \\ \dfrac{\partial t_t''}{\partial \tau}=\dfrac{\alpha_z F_z}{m_t c_t}(t_c''-t_t'')-\dfrac{\alpha_w F_w}{m_t c_t}(t_t''-t_w'') \\ \dfrac{\partial t_w''}{\partial \tau}+v_{w0}\dfrac{\partial t_w''}{\partial s}=\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c}(t_t''-t_w'') \\ x_c''=x_n(t_c'')=\dfrac{379.8\cdot 10^{\frac{7.5t_c''}{t_c''+237.29}}}{b-610.6\cdot 10^{\frac{7.5t_c''}{t_c''+237.29}}} \\ \Phi=\dfrac{dx_c''}{dt_c''}=\dfrac{379.8\cdot 237.29\cdot 7.5\cdot \ln 10\cdot b\cdot 10^{\frac{7.5t_c''}{t_c''+237.29}}}{\left[(t_c''+237.29)\left(b-610.6\cdot 10^{\frac{7.5t_c''}{t_c''+237.29}}\right)\right]^2}. \end{cases} \tag{14}$$

Boundary conditions for a set of equations (13) are the same as for dry cooling — dependencies (2) and (3). In accordance with the assumed designations they can be expressed in the following form:

$$t_c'(s=0,\tau)=t_2=\text{const} \tag{15}$$

$$t_w'(s=0,\tau)=t_{w2}(\tau)=\left(t_{w20}-t_{w2u}\right)e^{-\tau/T}+t_{w2u}. \tag{16}$$

Initial conditions for set (13) can be obtained from (17)

$$\begin{cases} \dfrac{dt_c'}{ds}=-\dfrac{\alpha_z F_z}{v\varrho F_{ch}L(1-b_f)(c_p+c_w x_2)}(t_c'-t_t') \\ \alpha_z F_z(t_c'-t_t')=\alpha_w F_w(t_t'-t_w') \\ \dfrac{dt_w'}{ds}=\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c v_{w0}}(t_t'-t_w') \end{cases} \tag{17}$$

with boundary conditions

$$t_c'(s=0)=t_2 \tag{18}$$

$$t_w'(s=0)=t_{w20}. \tag{19}$$

For a set of equations (14) obligatory in zone II of a cooler, boundary conditions must be given on boundary $s=s_w$ in the following form:

$$t_c''\left[s=s_w(\tau)\right]=t_{pr}=\text{const} \tag{20}$$

$$t_w''\left[s=s_w(\tau),\tau\right]=t_w'\left[s=s_w(\tau),\tau\right]. \tag{21}$$

Initial conditions are determined by solving a set of equations below, obtained from (14):

$$\begin{cases} \dfrac{dt_c''}{ds}=-\dfrac{\alpha_z F_z}{v\varrho F_{ch}L(1-b_f)\left[c_p+c_w x_c''+\left(r_p+c_w t_c''-c_c t_c''\right)\Phi\right]}(t_c''-t_t'') \qquad (22) \\ \alpha_z F_z(t_c''-t_t'')=\alpha_w F_w(t_t''-t_w'') \\ \dfrac{dt_w''}{ds}=\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c v_{w0}}(t_t''-t_w'') \\ \Phi=\dfrac{dx_c''}{dt_c''}=\dfrac{379.8\cdot 237.29\cdot 7.5\cdot \ln 10\cdot b\cdot 10^{\frac{7.5t_c''}{t_c''+237.29}}}{\left[(t_c''+237.29)\left(b-610.6\cdot 10^{\frac{7.5t_c''}{t_c''+237.29}}\right)\right]^2} \end{cases}$$

with boundary conditions

$$t_c''(s=s_w)=t_{pr} \tag{23}$$

$$t_w''(s=s_w)=t_w'(s=s_w). \tag{24}$$

Temperature of dew point present in dependence (20) is obtained from the following formula:

$$t_{pr} = \frac{237.29 \cdot \lg \dfrac{bx_2}{379.8 + 610.6\, x_2}}{7.5 - \lg \dfrac{bx_2}{379.8 + 610.6\, x_2}}. \tag{25}$$

It must be added here that co-ordinate s_w is obtained from condition (20).

A counter-current cooler, wet colling

As in the case of a co-current cooler, a mathematical description consists here of two sets of equations connected with each other by means of boundary conditions. The sign of the second component on the left side of an equation of balance of enthalpy of cooled water is different only. The following equations are obligatory for a counter-current cooler:

In zone of dry cooling (I) — $0 \leqslant s \leqslant s_w$:

$$\begin{cases} \dfrac{\partial t_c'}{\partial \tau} + v\dfrac{\partial t_c'}{\partial s} = -\dfrac{\alpha_z F_z}{\varrho F_{ch} L(1-b_f)(c_p + c_w x_2)}(t_c' - t_t') \\ \dfrac{\partial t_t'}{\partial \tau} = \dfrac{\alpha_z F_z}{m_t c_t}(t_c' - t_t') - \dfrac{\alpha_w F_w}{m_t c_t}(t_t' - t_w') \\ \dfrac{\partial t_w'}{\partial \tau} - v_{w0}\dfrac{\partial t_w'}{\partial s} = \dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c}(t_t' - t_w') \\ x_c' = x_2. \end{cases} \tag{26}$$

And in zone of wet cooling (II) — $s_w \leqslant s \leqslant L$:

$$\begin{cases} \left[c_p + c_w x_c'' + (r_p + c_w t_c'')\,\Phi\right]\dfrac{\partial t_c''}{\partial \tau} + v\left[c_p + c_w x_c'' + (r_p + c_w t_c'' - c_c t_c'')\,\Phi\right]\dfrac{\partial t_c''}{\partial s} = \\ \qquad = -\dfrac{\alpha_z F_z}{\varrho F_{ch} L(1-b_f)}(t_c'' - t_t'') \\ \dfrac{\partial t_t''}{\partial \tau} = \dfrac{\alpha_z F_z}{m_t c_t}(t_c'' - t_t'') - \dfrac{\alpha_w F_w}{m_t c_t}(t_t'' - t_w'') \\ \dfrac{\partial t_w''}{\partial \tau} - v_{w0}\dfrac{\partial t_w''}{\partial s} = \dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c}(t_t'' - t_w'') \\ x_c'' = x_n(t_c'') = \dfrac{379.8 \cdot 10^{\frac{7.5 t_c''}{t_c'' + 237.29}}}{b - 610.6 \cdot 10^{\frac{7.5 t_c''}{t_c'' + 237.29}}} \\ \Phi = \dfrac{dx_c''}{dt_c''} = \dfrac{379.8 \cdot 237.29 \cdot 7.5 \cdot \ln 10 \cdot b \cdot 10^{\frac{7.5 t_c''}{t_c'' + 237.29}}}{\left[(t_c'' + 237.29)\left(b - 610.6 \cdot 10^{\frac{7.5 t_c''}{t_c'' + 237.29}}\right)\right]^2}. \end{cases} \tag{27}$$

Here the problem of boundary conditions must be looked at from a slightly different perspective than in the case of a co-current cooler. In the case of a co-current cooler, for both changeables requiring the same conditions (temperature of air t_c and temperature of water t_w) conditions for zone I are given on the same boundary $s = 0$ and for zone II on the same boundary $s = s_w$. In the case of a counter-current cooler, boundary conditions are given on different boundaries. The following boundary conditions can be given for set (26):

$$t_c'(s = 0, \tau) = t_2 = \text{const} \tag{28}$$

$$t_w'(s = s_w, \tau) = t_w''(s = s_w, \tau). \tag{29}$$

As above, initial conditions for set (26) are determined based on a set of equations (30):

$$\begin{cases} \dfrac{dt_c'}{ds} = -\dfrac{\alpha_z F_z}{v\varrho F_{ch} L(1-b_f)(c_p + c_w x_2)}(t_c' - t_t') \\ \alpha_z F_z(t_c' - t_t') = \alpha_w F_w(t_t' - t_w') \\ \dfrac{dt_w'}{ds} = -\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c v_{w0}}(t_t' - t_w') \end{cases} \tag{30}$$

with boundary conditions

$$t_c'(s = 0) = t_2 \tag{31}$$

$$t_w'(s = s_w) = t_w''(s = s_w). \tag{32}$$

In addition, for a set of equations (27) used in zone II for a counter-current cooler, boundary conditions take the following form:

$$t_c''[s = s_w(\tau)] = t_{pr} = \text{const} \tag{33}$$

$$t_w''(s = L, \tau) = t_{w3}(\tau) = (t_{w30} - t_{w3u})e^{-\tau/T} + t_{w3u} \tag{34}$$

and initial conditions for it can be obtained from (35)

$$\begin{cases} \dfrac{dt_c''}{ds} = -\dfrac{\alpha_z F_z}{v\varrho F_{ch} L(1-b_f)\left[c_p + c_w x_c'' + (r_p + c_w t_c'' - c_c t_c'')\Phi\right]}(t_c'' - t_t'') \\ \alpha_z F_z(t_c'' - t_t'') = \alpha_w F_w(t_t'' - t_w'') \\ \dfrac{dt_w''}{ds} = -\dfrac{\alpha_w F_w}{\varrho_w V_{cw} c_c v_{w0}}(t_t'' - t_w'') \\ \Phi = \dfrac{dx_c''}{dt_c''} = \dfrac{379.8 \cdot 237.29 \cdot 7.5 \cdot \ln 10 \cdot b \cdot 10^{\frac{7.5t_c''}{t_c'' + 237.29}}}{\left[(t_c'' + 237.29)\left(b - 610.6 \cdot 10^{\frac{7.5t_c''}{t_c'' + 237.29}}\right)\right]^2} \end{cases} \tag{35}$$

with boundary conditions

$$t_c''(s = s_w) = t_{pr} \tag{36}$$

$$t_w''(s = L) = t_{w30}. \tag{37}$$

The following dependencies occur in all four cases discussed above:

$$t_2 = t_1 + \Delta t_{went} \tag{38}$$

$$x_2 = x_1, \tag{39}$$

where Δt_{went} is a rise in temperature of air in a fan [°C].

Cooled air of mass flow rate $Q_m(1-b_f)$ and not cooled air of mass flow rate $Q_m b_f$ get mixed at the outlet of a cooler, forming a mixture of temperature t_3 and specific humidity x_3. These values can be calculated based on the following set of equations:

$$\begin{cases} t_3 = \dfrac{(c_p t_{c3} + c_w x_{c3} t_{c3} + r_p x_{c3})(1-b_f) + (c_p t_2 + c_w x_2 t_2 + r_p x_2) b_f - r_p x_3}{c_p + c_w x_3} \\[2ex] x_3 = \begin{cases} x_{c3}(1-b_f) + x_2 b_f & \text{for} \quad x_{c3}(1-b_f) + x_2 b_f \leqslant x_n(t_3) \\ x_n(t_3) & \text{for} \quad x_{c3}(1-b_f) + x_2 b_f > x_n(t_3), \end{cases} \end{cases} \tag{40}$$

where:

$t_{c3} = t_c''(s = L)$,

$x_{c3} = x_c''(s = L) = x_n(t_{c3})$.

Specific humidity of air saturated with water vapour (x_n) is calculated from a dependence analogical to the fourth equation of the set (14) or (27).

For dry cooling, the above dependence is reduced to the following form:

$$\begin{cases} t_3 = t_{c3}(1-b_f) + t_2 b_f \\ x_3 = x_2. \end{cases} \tag{41}$$

3. Calculation examples

The results of exemplary calculations are presented below in order to show the differences in distributions along a cooler, in time courses of temperatures of media exchanging heat and in humidity of air in a co-current and counter-current cooler. These examples present cooling of air by means of a mining water surface cooler operating indirectly, type GCCP-115. The former example refers to cooling without outdropping of water, the latter one to cooling with outdropping.

The following data describing a cooler are necessary for calculations:

— specific heat of the material of a heat exchanger (copper): $c_t = 381$ J/(kgK),
— area of active cross-section of a cooler: $F_{ch} = 0.430$ m^2,
— area of inside surface of heat exchange in an exchanger: $F_w = 30.16$ m^2,
— area of outside surface of heat exchange in an exchanger: $F_z = 136{,}7$ m^2,
— length of a cooler: $L = 3$ m,
— mass of a heat exchanger: $m_t = 524$ kg,
— capacity of pipes of a heat exchanger: $V_{cw} = 0.0603$ m^3.

Example 1:

Both distributions of temperatures $t_c(s,\tau)$, $t_w(s,\tau)$ and $t_t(s,\tau)$, of humidity $x_c(s,\tau)$ and time course of temperature $t_3(\tau)$ and of specific humidity of air $x_3(\tau)$ at the outlet from a cooler, with unsteady dry cooling of air by means of cooler GCCP-115, while it operates both co-currently and counter-currently, must be found. Mass flow rate of air through a cooler (related to dry air) is equal to $Q_m = 10$ kg/s, its pressure $b = 105$ kPa, temperature at the entry to a fan $t_1 = 27$°C and specific humidity $x_1 = 11$ g/kg. Rise in temperature of air in a fan equals $\Delta t_{went} = 2$°C. Water flows through a cooler with mass flow rate equal to $Q_w = 3$ kg/s and its temperature at the entry falls exponentially with time constant $T = 360$s from $t_{w20} = 24$°C to $t_{w2u} = 12$°C from $t_{w30} = 24$°C to $t_{w3u} = 12$°C (in a counter-current cooler).

In situations with parameters above, condensation of water vapour does not occur, during both co-current and counter-current cooling.

Solution:

Using the equations presented above and dependencies given in [Gutkowski, 1972; Häussler, 1971; Kołodziejczyk, Rubik, 1969; Nowak, Łukosz, 1999], the problem presented above was solved numerically by means of computer programmes. The results were given in the form of graphs shown in figures 2—7, where distributions of temperatures of air (its cooled part), the heat exchanger and water along a cooler for three different time instants — Fig. 2 and 3, time courses of these temperatures at the entry and outlet from a cooler — Fig. 4 and 5, similar distributions of specific humidity of cooled part of air — Fig. 6, as well as time courses of parameters of cooled air — Fig. 7 were presented. As figure 2 is very clear, there was no need to put any legend under particular curves. Reciprocal position of curves — for a given cooler curve $t_c(s)$ is located in the highest point and $t_w(s)$ in the lowest — allows unique interpretation of graphs.

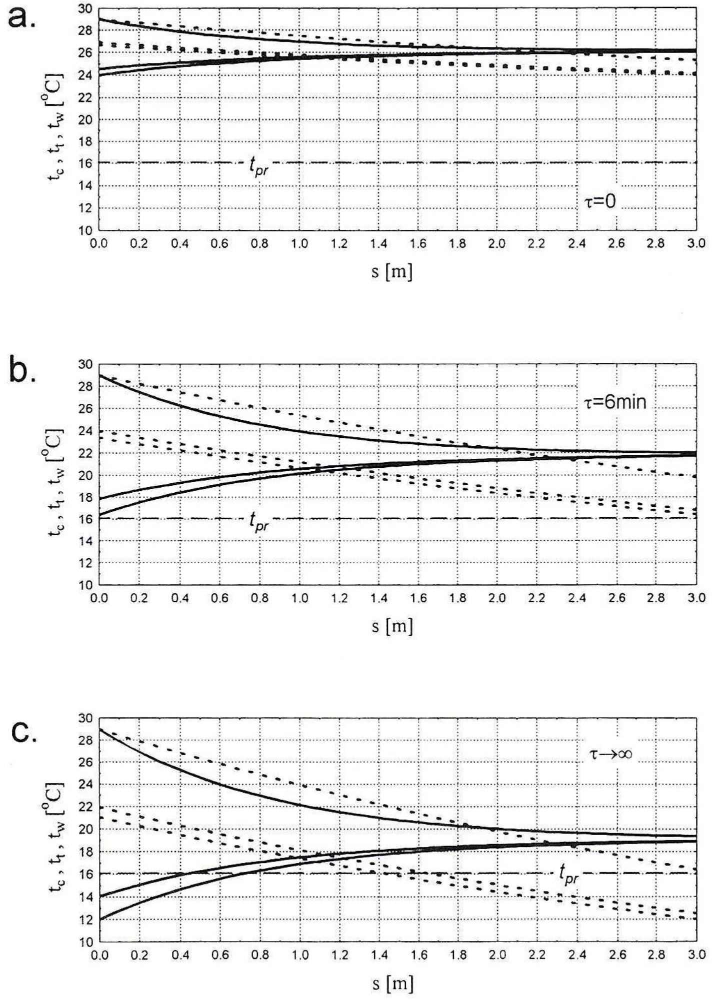


Fig. 2. Dry cooling of air. Distributions of temperatures of media exchanging heat in a co-current cooler (continuous lines) and a counter-current cooler (broken lines); a) in an initial state ($\tau = 0$), b) after 6 minutes, c) after the phenomenon gets steady ($\tau \to \infty$)

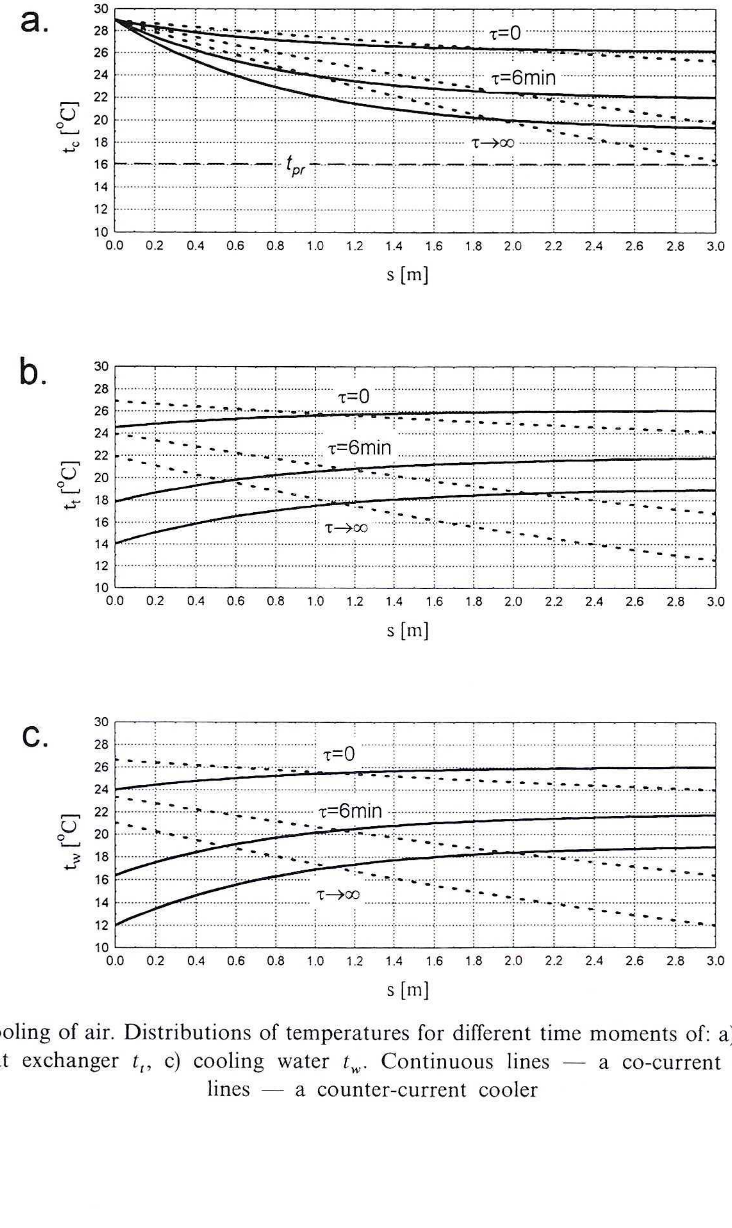


Fig. 3. Dry cooling of air. Distributions of temperatures for different time moments of: a) cooled part of air t_c, b) heat exchanger t_t, c) cooling water t_w. Continuous lines — a co-current cooler, broken lines — a counter-current cooler

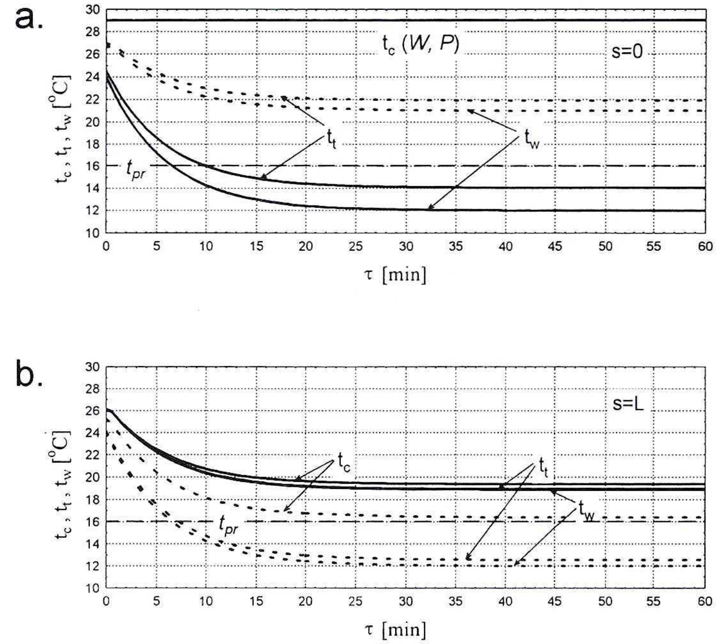


Fig. 4. Dry cooling of air. Time courses of temperatures of media exchanging heat in a co-current cooler (continuous lines) and a counter-current cooler (broken lines); a) in an entry cross-section of air ($s = 0$), b) in an outlet cross-section of air ($s = L$).

W — a co-current cooler, *P* — a counter-current cooler

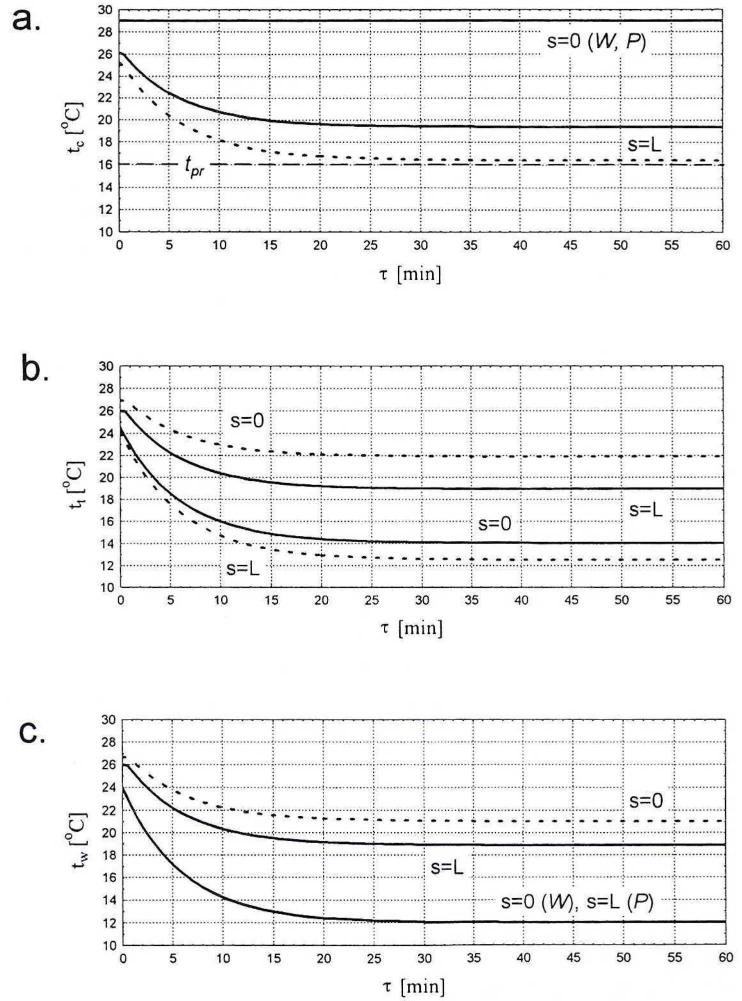


Fig. 5. Dry cooling of air. Time courses of temperatures for different cross-sections of co-current (continuous lines) and counter-current (broken lines) coolers of: a) cooled part of air t_c, b) heat exchanger t_t, c) cooling water t_w.
W — a co-current cooler, P — a counter-current cooler

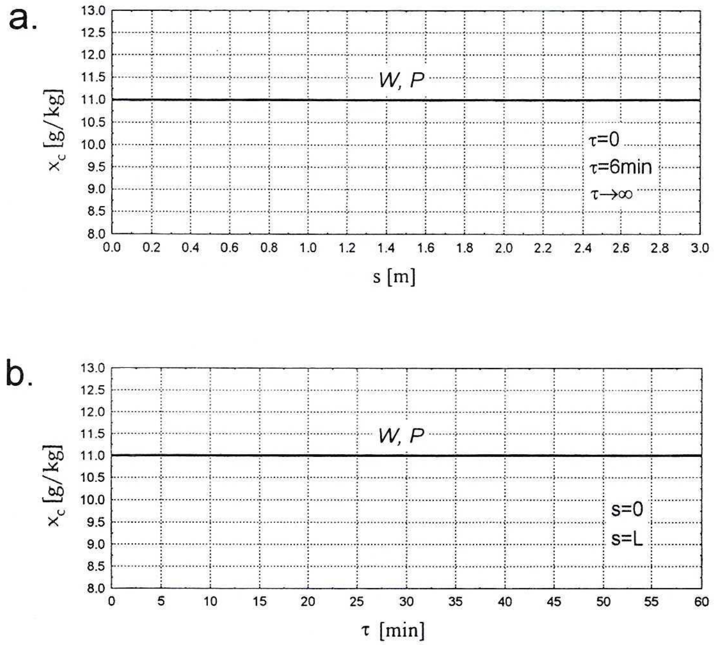


Fig. 6. Dry cooling of air a) distribution of specific humidity of cooled part of air x_c along a cooler for different time moments, b) time course of specific humidity of cooled part of air x_c in different cross-sections of a cooler.

W — a co-current cooler, P — a counter-current cooler

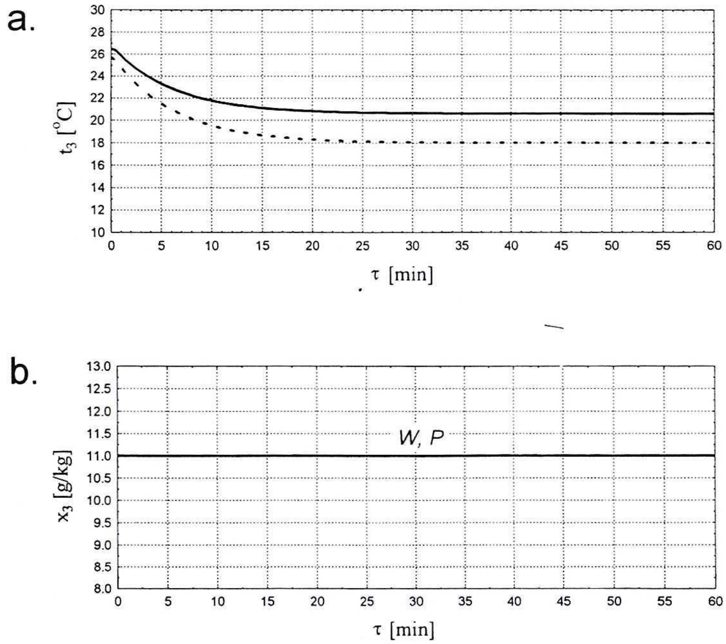


Fig. 7. Dry cooling of air. Time courses in a co-current (continuous lines) and a counter-current (broken lines) cooler of: a) temperature of cooled air t_3, b) specific humidity of cooled air x_3. W — a co-current cooler, P — a counter-current cooler

Example 2:

This example is nearly the same as example 1. The only difference is that higher specific humidity of air at the entry to a fan $x_1 = 16\,\mathrm{g/kg}$ was assumed. Condensation of water vapour in a cooler, fort both kinds of cooling, is ensured. Moreover, the results also include time course of the position of the boundary of zones I and II — $s_w(\tau)$.

Solution:

The results of calculations were also presented in the form of graphs — figures 8—13. The sequence of graphs is similar as in example 1. The comment above referring to figure 2 is also applicable to figure 8.

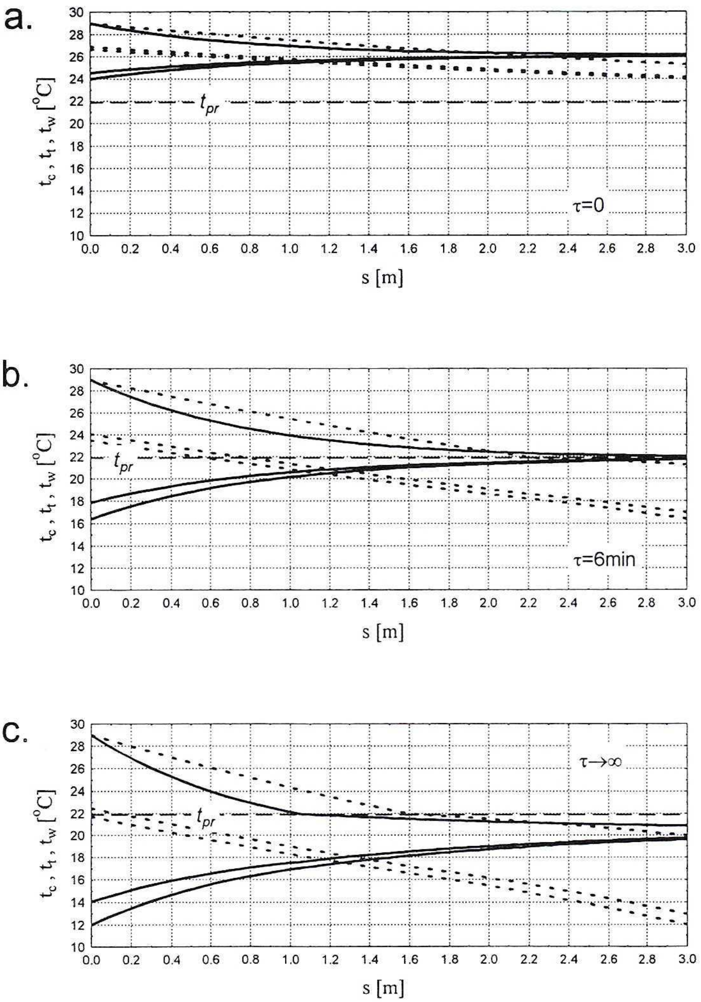


Fig. 8. Wet cooling of air. Distributions of temperature of media exchanging heat in a co-current (continuous lines) and counter-current (broken lines) cooler; a) in an initial state ($\tau = 0$), b) after 6 minutes, c) after the phenomenon gets steady ($\tau \to \infty$)

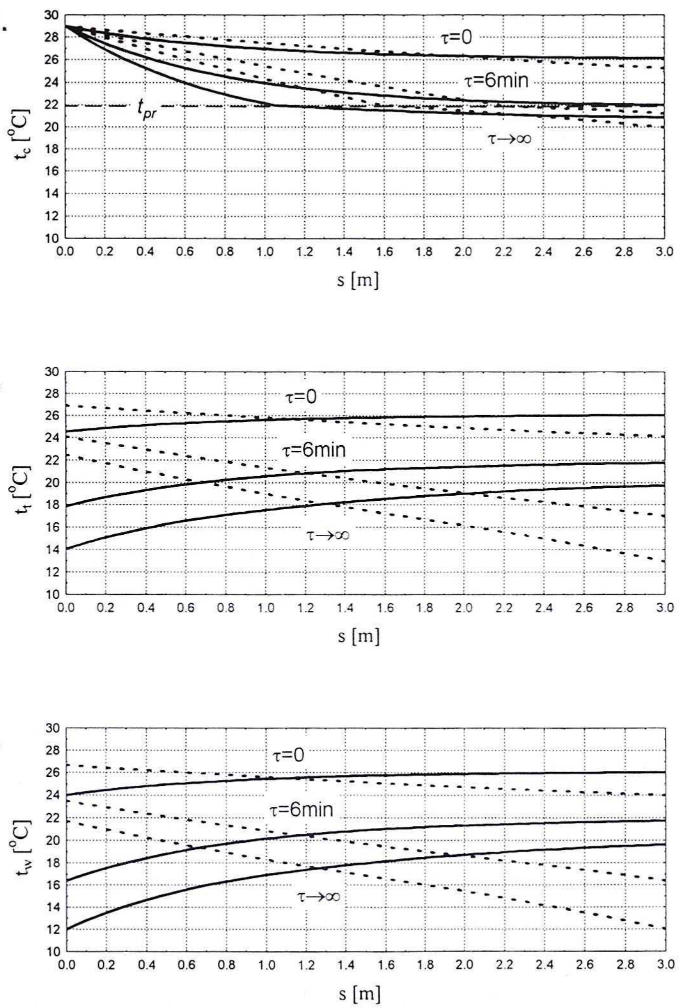
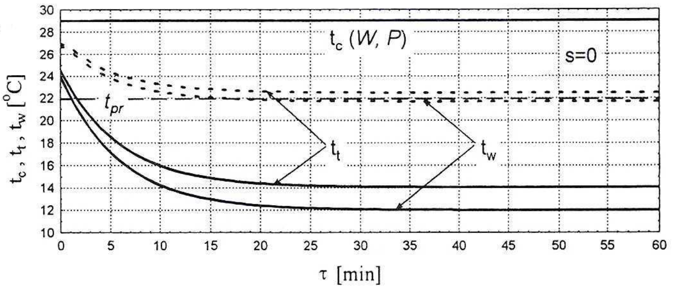


Fig. 9. Wet cooling of air. Distributions of temperatures for different time moments of: a) cooled part of air t_c, b) heat exchanger t_t, c) cooling water t_w. Continuous lines — a co-current cooler, broken lines — a counter-current cooler

a.



b.

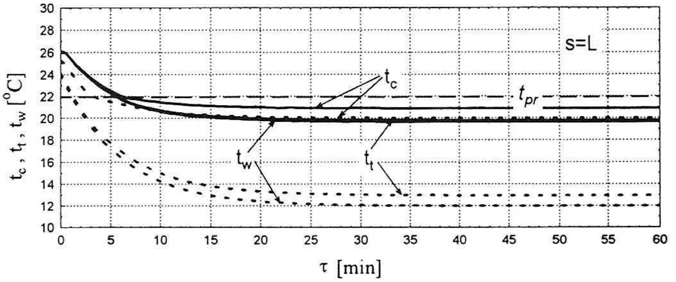


Fig. 10. Wet cooling of air. Time courses of temperatures of media exchanging heat in a co-current cooler (continuous lines) and a counter-current cooler (broken lines); a) in an entry cross-section of air ($s = 0$), b) in an outlet cross-section of air ($s = L$).
W — a co-current cooler, P — a counter-current cooler

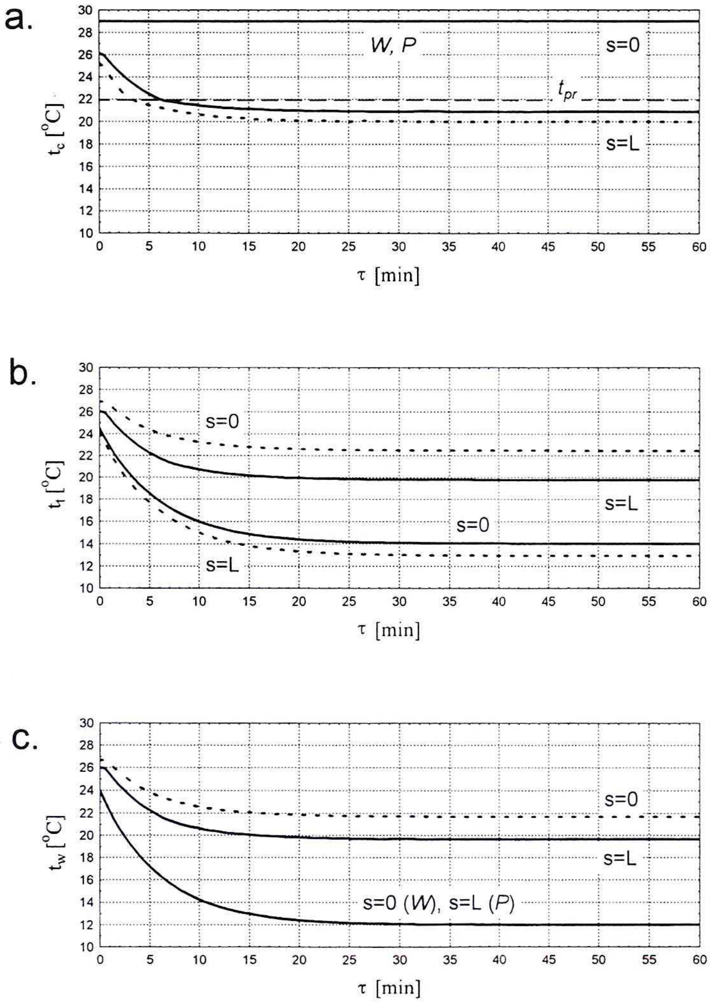
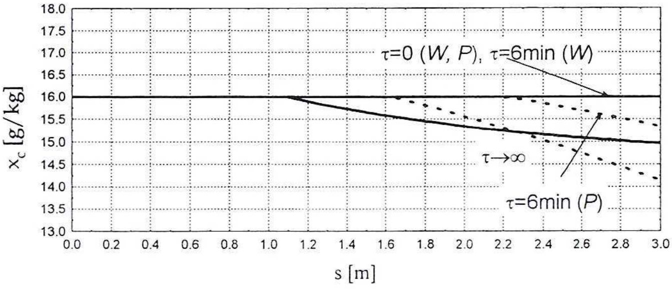


Fig. 11. Wet cooling of air. Time courses of temperatures for different cross-sections of co-current (continuous lines) and counter-current (broken lines) coolers of: a) cooled part of air t_c, b) heat exchanger t_t, c) cooling water t_w.
W — a co-current cooler, P — a counter-current cooler

a.



b.

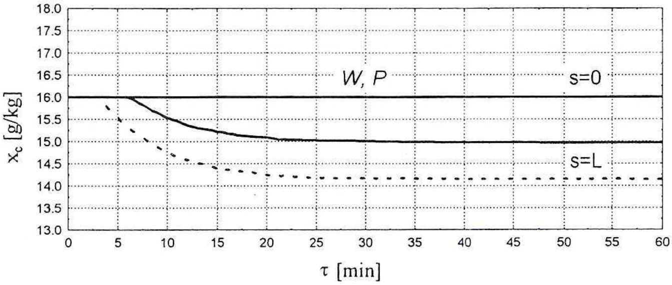


Fig. 12. Wet cooling of air a) distribution of specific humidity of cooled part of air x_c along a cooler for different time moments, b) time course of specific humidity of cooled part of air x_c in different cross-sections of a cooler. Continuous lines — a co-current cooler, broken lines — a counter-current cooler.

W — a co-current cooler, *P* — a counter-current cooler

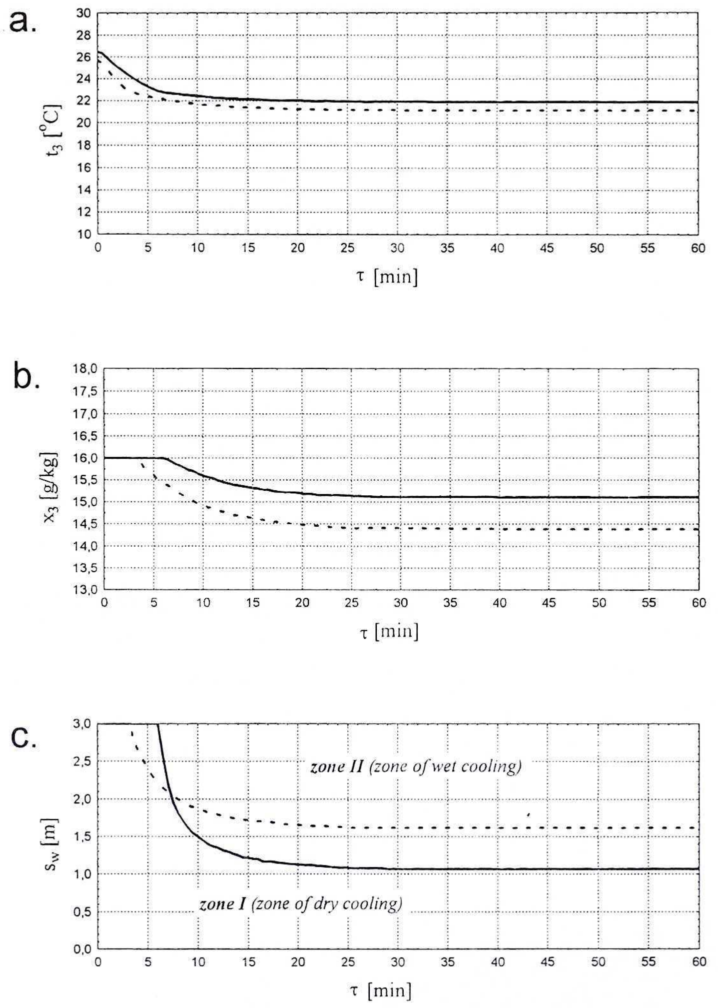


Fig. 13. Wet cooling of air. Time courses in a co-current (continuous lines) and a counter-current (broken lines) cooler of: a) temperature of cooled air t_3, b) specific humidity of cooled air x_3, c) position of boundary of zones of dry and wet cooling in a cooler — s_w.
W — a co-current cooler, P — a counter-current cooler

4. Conclusions

The analysis of graphs presented above allows drawing the following conclusions referring to cases under consideration:

— direction of flow of cooling water through a surface cooler greatly influences both space — time distributions of temperatures of media between which heat exchange occurs and humidity of air;

— in a counter-current cooler distributions of temperatures of air, heat exchanger and water in the function of co-ordinate s are similar to straight courses; while in a co-current cooler these courses are of an exponential character;

— changing slightly along axis s (compared to a co-current cooler) difference in temperatures of air and water in a counter-current cooler results in a more uniform cooling along its whole length;

— a counter-current cooler reduces temperature of air to a greater extent than a co-current cooler; therefore, drying of air by means of a counter-current cooler is greater (as far as specific humidity is concerned);

— monotonous character of functions $t_t(s)$ and $t_w(s)$ is obviously different for both kinds of coolers, which is connected with the direction of flow of a cooling medium;

— temperature of air in a co-current cooler falls more rapidly closer to its entry than in a counter-current cooler; as a result of which the zone of dry cooling in the former one is more narrow than in the latter one;

— however, in a co-current cooler closer to the outlet, slope of a curve of falling of temperature of air is smaller than in a counter-current one;

— character of time courses of all the values in both coolers is identical; there are only quantitative differences;

— condensation of vapour in a counter-current cooler begins after a shorter period of cooling than in a co-current cooler.

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