Archives of Mining Sciences **45**, 1 (2000) 71—88

KRZYSZTOF FILEK *, BERNARD NOWAK *, JANUSZ ROSZKOWSKI *

THE BEST SELECTION OF DUCT FAN CO-OPERATING WITH A SURFACE COOLER OR AIR

OPTYMALNY DOBÓR WENTYLATORA LUTNIOWEGO WSPÓŁPRACUJĄCEGO Z PRZEPONOWĄ CHŁODNICĄ POWIETRZA

This work takes into consideration the problem of the best selection of a fan forcing airflow through a mining counter-current cooler operating indirectly. The maximum thermal power of a fan — cooler system, which is equal to a steady state, was assumed as the criterion of such a cooperation. To determine this power a mathematical model (simplified, but taking into account condensation of water vapour) describing the distribution of temperature of air and its humidity, and temperature of a heat exchanger and cooled water along a cooler was used. Calculations were done for a Polish mining cooler of air GCCP-115 for two different states of air which was being cooled. This cooler cooperated with duct fans of four types: WLE-603A, WLE-804AM, WLE-1003B and WLE-1004A. The results of these calculations are presented in tables and graphs.

Key words: cooling of mines, air cooling, power of a cooler of air.

W pracy zajęto się zagadnieniem optymalnego doboru wentylatora wymuszającego przepływ przez górniczą wodną chłodnicę powietrza o działaniu pośrednim. Jako kryterium jakości takiej współpracy przyjęto maksimum, odpowiadającej stanowi ustalonemu, mocy cieplnej (N_u) układu wentylator-chłodnica.

Wentylator zamontowany na wlocie chłodnicy jest niezbędnym elementem takiego układu, lecz poprzez ogrzewanie przepływającego powietrza, pogarsza końcowy rezultat chłodzenia; mniejsza jest łączna moc całego układu (N_u), niż moc samej chłodnicy (N_c). Różne wentylatory, współpracujące z tą samą chłodnicą, powodują w ogólności różne przyrosty temperatury powietrza oraz różne prędkości przepływu powietrza przez chłodnicę.

Moc samej chłodnicy w funkcji prędkości powietrza opisuje krzywa o kształcie zbliżonym do wykładniczego — zależność ta jest rosnąca z wyraźnym nasyceniem przy dużych prędkościach. Z drugiej jednak strony, w celu uzyskania większej mocy chłodnicy przez zwiększenie prędkości powietrza, należy zastosować większy wentylator, o wyżej

* WYDZIAŁ GÓRNICZY, AKADEMIA GÓRNICZO-HUTNICZA, 30-059 KRAKÓW, AL. MICKIEWICZA 30

położonej charakterystyce spiętrzeniowej. To natomiast powoduje wzrost spiętrzenia wentylatora, a co za tym idzie zwiększenie przyrostu temperatury powietrza w wentylatorze i spadek mocy całego układu chłodzenia. Wynika z tego, że istnieje maksimum mocy N_u, odpowiadające pewnej pośredniej wartości prędkości przepływu chłodzonego powietrza.

Do określenia mocy chłodnicy N_c i mocy układu wentylator-chłodnica N_u należy znać wartości temperatury i wilgotności powietrza przed schłodzeniem i po schłodzeniu, jak również jego natężenie przepływu. Te pierwsze są zwykle znane, te drugie natomiast należy wyliczyć z bilansów ciepła i masy pary wodnej, odpowiadających procesowi chłodzenia. W niniejszej pracy zastosowano w tym celu uproszczony, ale uwzględniający zjawisko kondensacji pary wodnej, matematyczny model opisujący rozkłady temperatury i wilgotności powietrza, jak też temperatury pozostałych ośrodków uczestniczących w wymianie ciepła (wody chłodzącej i przepony wymiennika), wzdłuż chłodnicy.

Obliczenia przeprowadzono dla górniczej chłodnicy powietrza polskiej produkcji typu GCCP-115 przy dwóch różnych stanach powietrza poddawanego chłodzeniu — dla powietrza chłodniejszego i bardziej suchego oraz dla powietrza o wyższej temperaturze i wyższej wilgotności. Rozważono współpracę tej chłodnicy z wentylatorami lutniowymi czterech typów: WLE-603A, WLE-804AM, WLE-1003B i WLE-1004A. Wyniki przeprowadzonych obliczeń przedstawiono w tabelach i graficznie w formie wykresów. Jak wynika z ich obserwacji, najodpowiedniejszym wentylatorem dla chłodnicy GCCP-115 jest wentylator WLE-804AM.

Słowa kluczowe: klimatyzacja kopalń, chłodzenie powietrza, moc chłodnicy powietrza.

1. Introduction

The increase in a heat hazard in mining excavations, which is due both to the greater depths of location of useful minerals and concentration and mechanisation of works, encouraged many researches to take up this problem. Many different methods for forecasting temperature and humidity of air in mining excavations have been given and different ways of overcoming temperature hazards have been suggested. In deep mines more and more frequently it is necessary to use coolers of air in order to keep down the parameters of mining air within particular limits, in accordance with safety code. The use of coolers of air in such a situation is quite popular. The main criterion of their effectiveness is the range of obtained values of temperature and humidity of air where it is cooled, which is the result of the real thermal power of an exchanger. The differences in thermal powers of coolers of air given by manufacturers and real powers obtained from heat balances are first of all due to both parameters of cooled stream of air and parameters of cooling water, which are different from the assumed ones. Therefore, this article discusses the best selection of a duct fan forcing airflow through a mining, counter-current, surface air cooler operating indirectly. The best selection means the maximum thermal power of a fan — cooler system, which is equal to steady state.

In the cooler under consideration air is a cooled agent and cold water cooled outside a cooler is a cooling agent. Both air and water flow through a heat exchanger, without a direct contact. Water flows inside the pipes of an exchanger and air flows on the outside surface, which includes the pipes mentioned above and radiators increasing the area of the surface. Therefore, an exchanger is a cut-off wall, through which only the exchange of heat is possible.

Figure 1 presents schematically a fan — cooler system. Air which is cooled and having intensity Q (Q signifies here a mass flow rate in a cooler with reference to dry air) flows through a cooler having length L along a co-ordinate axis s. Its temperature at the entry to a fan is marked as t_1 and specific humidity as x_1, at the entry to a cooler, respectively as t_2 and x_2 and at its outlet as t_3 and x_3. Cooling water flows through a cooler in the opposite direction than the air. Its mass flow rate is marked as Q_w, entry temperature as t_{w0} and outlet temperature as t_{wp}.

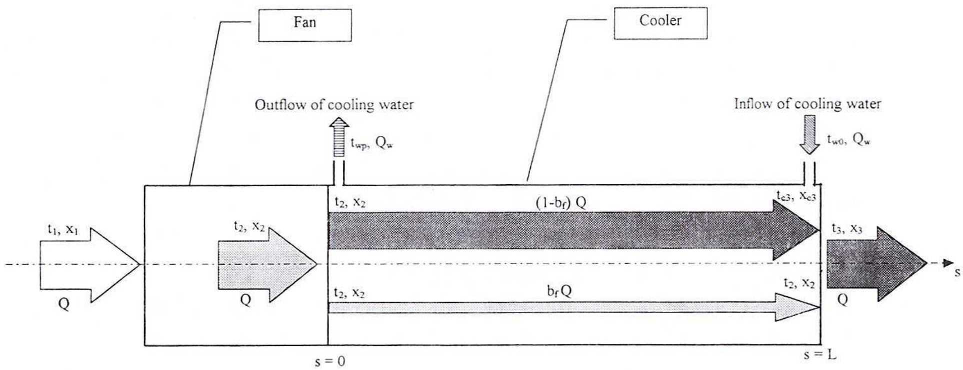

Fig. 1. The scheme of airflow through a system consisting of a duct fan and a counter-current cooler of air operating indirectly

While analysing cooling of air by-pass factor of a cooler (b_f) was used as well as the possibility of condensation of water vapour in the air which was cooled. However, it was assumed that out of the whole airflow, having mass flow rate Q, flowing through the cooler, only a part, having mass flow rate $(1-b_f)Q$, undergoes cooling and drainage (fall in temperature from t_2 to t_{c3} and in specific humidity from x_2 to x_{c3}) and the remaining part, having volumetric flow rate $b_f Q$, flows by assumed by-pass, without any change in temperature and humidity. It was also assumed that at the outlet from a cooler both the streams get ideally mixed, forming a mixture with temperature t_3 and specific humidity x_3. Changeable temperature and specific humidity of this part of air which exchanges heat and mass was marked as t_c and x_c, temperature of cooling water as t_w and of cut-off wall as t_t. Condensation of water vapour was taken into account as two zones in the cooler were distinguished: located closer to the entry of air the zone of dry cooling (zone I), where the temperature of air is so high that there is no condensation of vapour and located closer from the outlet of air the zone of cooling with condensation of vapour (zone II). A different mathematical description was used for each zone. Co-ordinate, which is the boundary of the zones, was marked as s_w. Value s_w for a particular cooler generally depends on parameters both of cooled air and of cooling water. It must be added here that this value is not known a priori; the method for its determination will be presented in the further part of this work.

2. Power of a cooler and of a cooler — fan system

In all the problems regarding designing and forecasting the work of a cooler thermal (cooling) power is one of the most important parameters characterizing this work. Here we mean real power exchanged between cooled air and cooling water, not nominal power given by manufacturers of coolers.

(Negative) power delivered to an exchanger together with cold water is generally used for cooling air and condensation of water vapour contained in it; in some specific cases condensation may not occur. The work of a cooler results in the fall in temperature of outflow air and a subsequent fall in its specific humidity and also an increase in temperature flowing out of the pipes of a water exchanger.

The parameters of outflow air of a cooler depend to a great extent on how the delivered power is distributed into these two effects, i. e. its cooling and drainage. It is related to the distribution of temperature of cooled part of air (t_c) along a cooler. In the zone of dry cooling (zone, I, $0 < s < s_w$) this temperature falls until it reaches the value of temperature of dew point t_{pr} in the cross-section of a cooler with co-ordinate s_w. The whole delivered power is used in this zone for cooling of air and specific humidity of air does not change. However, in zone II ($s_w < s < L$) simultaneous cooling of air and condensation of water vapour contained in it take place. Therefore, since only a part of delivered power is used for the former process, the drop of curve $t_c(s)$ is not as steep as in zone I. The remaining part of power is used for outdropping of vapour, which results in the fall in specific humidity of air. In the whole zone only the excess of vapour over the value equal to saturation is outdropped; therefore the cooled part of air is always here in a saturated state.

When the temperature of air at the entry to a cooler is not much higher than its temperature of dew point (high relative humidity of air), zone I is narrow and zone II is wide; condensation of water vapour takes place along nearly the whole cooler. At that time the fall in temperature of air in it is relatively small and the fall in specific humidity is high. In such a case the work of a cooler results to a large extent in drainage, not cooling of air. However, if condensation of water vapour in a cooler takes place close to the outlet of air, the temperature effect of the work of a cooler is great (steep fall of curve $t_c(s)$ in wide zone I and not as steep in zone II) and the effect of drainage of air is small.

We are going to take into consideration the power of cooler (N_c) and of fan — cooloer system (N_u), which is especially interesting from a practical point of view. It must be emphasised here that a fan forcing airflow through a cooler, although indispensable, makes the conditions of its work worse by heating the air that is to be cooled. As a result, the cooling power of the whole system is smaller than the power of the cooler itself.

Values N_c and N_u mentioned above can be calculated on the basis of the following dependencies:

$$N_c = (h_2 - h_3)Q \tag{1}$$

$$N_u = (h_1 - h_3)Q \tag{2}$$

where:

h_1, h_2 — specific enthalpy of air, respectively at the entry to a fan and at the entry to a cooler [J/kg],
h_3 — sum of specific enthalpy of air at the outlet from a cooler and enthalpy of outdropped water [J/kg],
Q — mass flow rate of dry air in a cooler [kg/s].

Specific enthalpy of air for the three cross-sections mentioned above can be obtained in the following way [Gutkowski 1972; Häussler 1971; Pawiński et al. 1995; Wacławik et al. 1995]:

$$h_1 = c_p t_1 + c_w x_1 t_1 + r x_1 \tag{3}$$

$$h_2 = c_p t_2 + c_w x_2 t_2 + r x_2 \tag{4}$$

$$h_3 = c_p t_3 + c_w x_3 t_3 + r x_3 + c_c t_3 (x_2 - x_3) \tag{5}$$

It was assumed here that the temperature of outdropped water flowing out of a cooler is equal to the temperature of air t_3 flowing out of a cooler.

Additionally, taking into consideration the fact that airflow in a fan increases its temperature by Δt_{went}, but does not change its specific humidity.

$$t_2 = t_1 + \Delta t_{went} \tag{6}$$

$$x_2 = x_1 \tag{7}$$

N_c and N_u can be expressed in the following way:

$$N_c = \{c_p(t_1 + \Delta t_{went} - t_3) + c_w[x_1(t_1 + \Delta t_{went}) - x_3 t_3] + (x_1 - x_3)(r - c_c t_3)\} Q \tag{8}$$

$$N_u = \{c_p(t_1 - t_3) + c_w(x_1 t_1 - x_3 t_3) + (x_1 - x_3)(r - c_c t_3)\} Q \tag{9}$$

where:

Δt_{went} — increase in temperature of air in a fan [°C],
c_p, c_w, c_c — specific heat, respectively of air at constant pressure, vapour at constant pressure, water [J/(kgK)],
r — latent heat of water vapourisation [J/kg].

Therefore, temperature and humidity of air at the entry to a fan and at the outlet from a cooler, increase in temperature of air in a fan and the amount of cooled air must be known in order to determine the power of a cooler and of a fan — cooler system. The state of air at the entry (t_1, x_1) is usually known, Δt_{went} can be obtained from a well-known dependence (10) [Informacja 1984; Markefka, Stefanowicz 1986;

Nowak, Filek 1996; Holesz 1997] and the method for determining the state of air at the outlet from a cooler is going to be discussed in the following sub-chapter.

$$\Delta t_{\text{went}} = \frac{\Delta p}{\varrho c_n \eta_w} \tag{10}$$

where:

Δp — total pressure of a fan [Pa],
ϱ — density of air, with reference to dry air [kg/m^3],
η_w — efficiency of a fan [−].

3. The change in the state of air in a cooler

The basic problem while testing the influence of a cooler on air is the determination of the state (i. e. temperature and humidity) of outflow air from a cooler on the basis of the state of inflow air, temperature of inflow water and the amount of both these agents. The mathematical model of time-space changes in temperature of air, its humidity and temperature of the cut-off wall of a heat exchanger and cooling water was presented in: [Holesz 1997; Filek et al. 1998a; Filek et al. 1998b; Filek et al. 1998c; Filek et al. 1999a]. From practical point of view the values of temperature and humidity of a cooled agent — air (at steady state) flowing out of a heat exchanger are of great interest. The mathematical model mentioned above was adapted for such a state. Further on, this model is called a full model or a model without any simplifications and it consists of sets of non-linear differential equations, linked with each other by means of boundary conditions. The equations of this description were obtained from balances of enthalpy of cooled air, the cut-off wall of a heat exchanger and cooling water and also from a balance of mass of water vapour in the air. They were obtained on the basis of the following assumptions:

— influence of changes in density (ϱ) and pressure (b) of air were neglected,
— uniform distribution of mass of a heat exchanger, its outside and inside surface and volume occupied in a cooler by cooled air and cooling water along a cooler was assumed,
— ideal leak tightness of side-walls of a cooler was assumed,
— heat transmission by means of radiation was neglected,
— heat conduction in the direction parallel to axis s was neglected,
— it was assumed that all the outdropped water is taken out of the system under consideration and does not remain in the air e.g. in the form of fog.

Solving the equations of the full model requires the use of a sophisticated computer programme; therefore it is not appropriate for common use in practice. The authors of this work simplified this model by assuming a constant value of co-efficient α_w (calculated for temperature t_{w0}) and value Φ (calculated for temperature t_{pr}) and by neglecting some components in denominator of the right side of the first equation from set (12), which enables obtaining an analytical solution easily. The method for simplifying the full model and obtained results are presented in: [Filek et al. 1999b].

For steady state the mathematical description, before simplifying (the full model) takes a form of non-linear sets of differential equations (11) and (12) with boundary conditions (13)—(16). To avoid ambiguity the values which play a role of unknowns in equations are marked with „'" in zone I and with „"" in zone II.

For zone I ($0 \leqslant s \leqslant s_w$):

$$\begin{cases} \dfrac{dt'_c}{ds} = -\dfrac{\alpha_z F_z}{\varrho F_{ch} Lv(1-b_f)(c_p + c_w x'_c)}(t'_c - t'_t) \\ \alpha_z F_z (t'_c - t'_t) = \alpha_w(t'_w) F_w (t'_t - t'_w) \\ \dfrac{dt'_w}{ds} = -\dfrac{\alpha_w(t'_w) F_w}{\varrho_w V_{cw} c_c v_{w0}}(t'_t - t'_w) \\ \alpha_w(t'_w) = (22.5\, t'_w + 1430)\, v_w^{0.8} d_w^{-0.2} \\ x'_c = x_2 \end{cases} \tag{11}$$

For zone II ($s_w \leqslant s \leqslant L$):

$$\begin{cases} \dfrac{dt''_c}{ds} = -\dfrac{\alpha_z F_z}{\varrho F_{ch} Lv(1-b_f)\left[c_p + c_w x''_c + (r + c_w t''_c - c_c t''_c)\,\Phi(t''_c)\right]}(t''_c - t''_t) \\ \alpha_z F_z (t''_c - t''_t) = \alpha_w(t''_w) F_w (t''_t - t''_w) \\ \dfrac{dt''_w}{ds} = -\dfrac{\alpha_w(t''_w) F_w}{\varrho_w V_{cw} c_c v_{w0}}(t''_t - t''_w) \\ x''_c = x_n(t''_c) = \dfrac{379.8 \cdot 10^{\frac{7.5\, t''_c}{t''_c + 237.29}}}{b - 610.6 \cdot 10^{\frac{7.5\, t''_c}{t''_c + 237.29}}} \\ \Phi(t''_c) = \dfrac{dx''_c}{dt''_c} = \dfrac{379.8 \cdot 237.29 \cdot 7.5 \cdot \ln 10 \cdot b \cdot 10^{\frac{7.5\, t''_c}{t''_c + 237.29}}}{\left[(t''_c + 237.29)\left(b - 610.6 \cdot 10^{\frac{7.5\, t''_c}{t''_c + 237.29}}\right)\right]^2} \\ \alpha_w(t''_w) = (22.5\, t''_w + 1430)\, v_w^{0.8} d_w^{-0.2} \end{cases} \tag{12}$$

Boundary conditions for the above equations are as follows:

$$t'_c(s=0)=t_2 \tag{13}$$

$$t''_c(s=s_w)=t'_c(s=s_w)=t_{pr} \tag{14}$$

$$t'_w(s=s_w)=t''_w(s=s_w) \tag{15}$$

$$t''_w(s=L)=t_{w0} \tag{16}$$

SYMBOL USED IN (11)—(16) STAND FOR:

b — absolute pressure of air [Pa],
b_f — by-pass factor of a cooler [−],
d_w — inside diameter of a pipe of a heat exchanger [m],
F_{ch} — area of effective cross-section of a cooler [m^2],
F_w — area of inside surface of heat exchange in an exchanger [m^2],
F_z — total area of outside surface of a heat exchanger [m^2],
L — length of a cooler [m],
s — current co-ordinate [m],
s_w — boundary co-ordinate of zone of dry cooling and condensation of water vapour in a cooler [m],
t'_c, t''_c — searched temperature of cooled part of air, respectively in a dry zone and with condensation of water vapour in a cooler [°C],
t_{pr} — temperature of dew point of inflow air in a cooler [°C],
t'_t, t''_t — searched temperature of a cut-off wall, respectively in a dry zone and with condensation of water vapour in a cooler [°C],
t'_w, t''_w — searched temperature of cooling water, respectively in a dry zone and with condensation of water vapour in a cooler [°C],
t_{w0} — temperature of cooling inflow water in a cooler [°C],
t_1, t_2 — temperature of inflow air, respectively in a fan and a cooler [°C],
V_{cw} — capacity of pipes of a heat exchanger [m^3],
v — average velocity of airflow through a cooler [m/s],
v_w — true average velocity of flow of cooling water through a pipe of a heat exchanger [m/s],
v_{w0} — average calculation velocity of flow of cooling water through a cooler [m/s],
x'_c, x''_c — searched specific humidity of cooled part of air, respectively in a dry zone and with condensation of water vapour in a cooler [1 kg of H_2O vapour/1 kg of dry air],
$x_n(t''_c)$ — specific humidity of air in saturation state at temperature t''_c [1 kg of H_2O vapour/1 kg of dry air],
x_1, x_2 — specific humidity of air flowing into, respectively a fan and a cooler [1 kg of H_2O vapour/1 kg of dry air],
α_w, α_z — coefficients of taking up heat, respectively on the inside and outside surface of a heat exchanger [$W/(m^2K)$],
ϱ — air density in a cooler (with reference to dry air) [kg/m^3],
ϱ_w — water density [kg/m^3].

As a result of simplifying the full model, which does not influence greatly the obtained results [Filek et al. 1999b, Filek et al. 1999c], the description in the form of equations (17)—(22) was obtained. Sets of differential equations (17) and (19) are linear; it is possible to solve them analytically. The simplified description takes the following form:

For zone I ($0 \leqslant s \leqslant s_w$):

$$\left\{\begin{aligned} \frac{dt'_c}{ds} &= K_1(t'_w - t'_c) \\ \frac{dt'_w}{ds} &= K_2(t'_w - t'_c) \end{aligned}\right. \tag{17}$$

For zone II ($s_w \leqslant s \leqslant L$):

$$\left\{\begin{aligned} \frac{dt''_c}{ds} &= K_3(t''_w - t''_c) \\ \frac{dt''_w}{ds} &= K_2(t''_w - t''_c) \end{aligned}\right. \tag{18}$$

with boundary conditions:

$$t'_c(s=0) = t_1 \qquad t'_w(s=0) = t_{wp} \tag{19}$$

$$t''_c(s=s_w) = t_{pr} \qquad t''_w(s=L) = t_{w0} \tag{20}$$

Boundary conditions (13), (14) and (16) remained unchanged, but it is more convenient to assume a condition for variable t'_w not at boundary $s = s_w$, as in (15), but at boundary $s = 0$ (second condition (19)). Value t_{wp} is an unknown a priori temperature of outflow water from a cooler. The method for determining it, as well as s_w, will be presented further on a set of algebraic equations (29) and (30).

Solving the above sets of equations (17) and (18) and using unchanged, as a result of simplifications, equations for the temperature of the cut-off wall and humidity of air, the following is obtained:
For the zone of dry cooling ($0 \leqslant s \leqslant s_w$):

$$t'_c(s) = \frac{1}{K_4}\left[K_2 t_2 - K_1 t_{wp} - K_1(t_2 - t_{wp})e^{K_4 s}\right] \tag{21}$$

$$t'_t(s) = \frac{1}{K_4}\left[K_2 t_1 - K_1 t_{wp} - K_6(t_1 - t_{wp})e^{K_4 s}\right] \tag{22}$$

$$t'_w(s) = \frac{1}{K_4}\left[K_2 t_2 - K_1 t_{wp} - K_2(t_2 - t_{wp})e^{K_4 s}\right] \tag{23}$$

$$x'_c(s) = x_1 = x_2 \tag{24}$$

For the zone of cooling with condensation of water vapour ($s_w \leqslant s \leqslant L$):

$$t_c''(s) = \frac{K_2 t_{pr} e^{K_5 L} - K_3 t_{w0} e^{K_5 s_w} - K_3(t_{pr} - t_{w0}) e^{K_5 s}}{K_2 e^{K_5 L} - K_3 e^{K_5 s_w}} \tag{25}$$

$$t_t''(s) = \frac{K_2 t_{pr} e^{K_5 L} - K_3 t_{w0} e^{K_5 s_w} - K_7(t_{pr} - t_{w0}) e^{K_5 s}}{K_2 e^{K_5 L} - K_3 e^{K_5 s_w}} \tag{26}$$

$$t_w''(s) = \frac{K_2 t_{pr} e^{K_5 L} - K_3 t_{w0} e^{K_5 s_w} - K_2(t_{pr} - t_{w0}) e^{K_5 s}}{K_2 e^{K_5 L} - K_3 e^{K_5 s_w}} \tag{27}$$

$$x_c'' = x_n(t_c'') = \frac{379.8 \cdot 10^{\frac{7.5 t_c''}{t_c'' + 237.29}}}{b - 610.6 \cdot 10^{\frac{7.5 t_c''}{t_c'' + 237.29}}} \tag{28}$$

Unknown values of a co-ordinate of boundary section s_w and of temperature of outflow water from a cooler t_{wp} are determined on the basis of continuity of temperatures of air and cooling water at the boundary of two zones of cooling mentioned above — (14) and (15). After inserting $s = s_w$ and equating (21) to t_{pr} and (23) to (27) the appropriate set of equations (29), (30) is obtained. This set can be solved numerically or graphically and values s_w and t_{wp} calculated in such a way can be inserted into dependencies (21)—(23) and (25)—(27), which finishes the procedure of calculation of temperatures. Knowing t_c in zone II (marked as t_c''), on the basis of (28), the distribution of specific humidity of air in this zone can be obtained.

$$t_{wp} = \frac{K_4 t_{pr} - K_2 t_2 + K_1 t_2 e^{K_4 s_w}}{K_1(e^{K_4 s_w} - 1)} \tag{29}$$

$$t_{wp} = \frac{K_2 t_2(e^{K_4 s_w} - 1)}{K_2 e^{K_4 s_w} - K_1} + K_4 \frac{K_2 t_{pr}(e^{K_5 L} - e^{K_5 s_w}) + K_5 t_{w0} e^{K_5 s_w}}{(K_2 e^{K_5 L} - K_3 e^{K_5 s_w})(K_2 e^{K_4 s_w} - K_1)} \tag{30}$$

Constants $K_1 - K_7$ occuring in equations (17)—(30) are equal to:

$$K_1 = \frac{C_1 \alpha_w(t_{w0}) F_w}{\alpha_w(t_{w0}) F_w + \alpha_z F_z} \tag{31}$$

$$K_2 = \frac{C_2 \alpha_z F_z}{\alpha_w(t_{w0}) F_w + \alpha_z F_z} \tag{32}$$

$$K_3 = \frac{C_3 \alpha_w(t_{w0}) F_w}{\alpha_w(t_{w0}) F_w + \alpha_z F_z} \tag{33}$$

$$K_4 = K_2 - K_1 \tag{34}$$

$$K_5 = K_2 - K_3 \tag{35}$$

$$K_6 = \frac{K_2 \alpha_w(t_{w0}) F_w + K_1 \alpha_z F_z}{\alpha_w(t_{w0}) F_w + \alpha_z F_z} \tag{36}$$

$$K_7 = \frac{K_2\alpha_w(t_{w0})F_w + K_3\alpha_z F_z}{\alpha_w(t_{w0})F_w + \alpha_z F_z} \tag{37}$$

and:

$$C_1 = \frac{\alpha_z F_z}{\varrho F_{ch} L v (1-b_f)(c_p + c_w x_2)} \tag{38}$$

$$C_2 = \frac{\alpha_w(t_{w0})F_w}{\varrho_w V_{cw} c_c v_{w0}} \tag{39}$$

$$C_3 = \frac{\alpha_z F_z}{\varrho F_{ch} L v (1-b_f)\left[c_p + \Phi(t_{pr}) r\right]} \tag{40}$$

$$v = \frac{Q}{\varrho F_{ch}} \tag{41}$$

$$v_w = \frac{4\,Q_w}{\pi d_w^2 \varrho_w n} \tag{42}$$

$$v_{w0} = \frac{Q_w L}{V_{cw}\varrho_w} \tag{43}$$

where:

n — number of pipes in a heat exchanger of a cooler [–],

Q_w — mass flow rate of cooling water in a cooler [kg/s].

The temperature of dew point can be calculated on the basis of expression [Roszczynialski et al. 1992]:

$$t_{pr} = \frac{237.29\ \lg\dfrac{bx_2}{379.8 + 610.6\,x_2}}{7.5 - \lg\dfrac{bx_2}{379.8 + 610.6\,x_2}} \tag{44}$$

As it has already been said, the temperature and humidity of air flowing out of a cooler (t_3, x_3), i.e. for $s = L$, are determined with the assumption that a cooled part of air and not cooled one are mixed ideally, which can be expressed in the following form:

$$t_3 = \frac{(c_p t_{c3} + c_w x_{c3} t_{c3} + r x_{c3})(1-b_f) + (c_p t_2 + c_w x_2 t_2 + r x_2) b_f - r x_3}{c_p + c_w x_3} \tag{45}$$

$$x_3 = \begin{cases} x_{c3}(1-b_f) + x_2 b_f & \text{for} \quad x_{c3}(1-b_f) + x_2 b_f \leqslant x_n(t_3) \\ x_n(t_3) & \text{for} \quad x_{c3}(1-b_f) + x_2 b_f > x_n(t_3) \end{cases} \tag{46}$$

where:

$$t_{c3} = t_c''(s = L)$$
$$x_{c3} = x_n(t_{c3}) \tag{47}$$

In the above set of equations the authors took into consideration the possibility of additional condensation of water vapour while mixing streams of air mentioned above if the following condition is fulfilled:

$$x_{c3}(1-b_f)+x_2 b_f > x_n(t_{c3}) \tag{48}$$

If there is no condensation of water vapour in a cooler, distributions of temperatures t_c, t_t and t_w are obtained from a set of equations (11) with the following conditions:

$$t_c(s=0) = t_2; \qquad t_w(s=L) = t_{w0} \tag{49}$$

However, specific humidity of air remains steady and is equal to x_1. Such a case, together with a calculation example, was discussed in [Filek et al. 1998a].

The procedure of calculation of temperature and humidity of air at the outlet of a coller was finished in such a way.

4. The best selection of a fan for a cooler

The following data must be known in order to start a theoretical calculation of an energetic effect of linking a particular cooler with a given fan into one system:

— constants characterizing a cooler, occurring in equations of the previous sub-chapter,

— data describing air at the entry to a fan, except volumetric airflow rate,

— data describing water at the entry to a cooler,

— resistance of a cooler R [kg/m^7],

— characteristics of a fan $\Delta p(V_m)$ and $\eta_w(V_m)$.

V_m stands for a volumetric airflow rate in a cooler [m^3/s].

Values connected with geometry of a cooler (L, F_w, F_z, F_{ch}, V_{cw}, d_w) are usually known. We suggest that a by-pass factor b_f for the cooler under consideration should be determined on the basis of work by [Nowak, Łukosz 1999]. These authors present for cooler GCCP-115 the following dependence, based on the results of measurements:

$$b_f(V_m) = 1-\exp(-0.01603\, V_m) \tag{50}$$

The determination of the amount of air blown into a cooler can be done graphically by means of finding an intersection point between total pressure characteristics of a fan $\Delta p(V_m)$ and curve expressed by the following equation:

$$\Delta p = RV_m^2 \tag{51}$$

For a volumetric airflow rate V_m determined in such a way, on the basis of efficiency characteristics η_w can be obtained. The same results can be obtained by means of calculations, assuming that characteristics of a fan can be expressed e.g. by polynomials in the following form:

$$\Delta p(V_m) = a_0 + a_1 V_m + a_2 V_m^2 + a_3 V_m^3 \tag{52}$$

$$\eta_w(V_m) = a_4 + a_5 V_m + a_6 V_m^2 + a_7 V_m^3 \tag{53}$$

where:

a_0, a_1, a_2, ..., a_7 — constants.

A volumetric airflow rate in a cooler V_m is determined by solving algebraic equation (54), total pressure of a fan on the basis of (51) or (52) and its efficiency on the basis of (53).

$$RV_m^2 = a_0 + a_1 V_m + a_2 V_m^2 + a_3 V_m^3 \tag{54}$$

The following dependence (for calculations density of air in front of a fan was assumed) occurs between a mass flow rate of dry air Q and a volumetric flow rate of humid air V_m:

$$V_m = \frac{Q(1+x_1)}{\varrho_{m1}} \tag{55}$$

where:

ϱ_{m1} — density of humid air at the entry to a fan [kg/m^3],

while [Roszczynialski et al. 1992, Wacławik et al. 1995]:

$$\varrho_{m1} = \frac{0.0021672 \cdot b(1+x_1)}{(273+t_1)(0.622+x_1)} \tag{56}$$

Increase in temperature of air in a fan expressed by dependence (10) can be expressed in the following form:

$$\Delta t_{\text{went}} = \frac{RQ^2(1+x_1)^2}{\varrho \varrho_{m1}^2 c_p \eta_w} \tag{57}$$

If we know thermodynamic parameters of inflow air in a fan, in a cooler and outflow air, we can determine its thermal power N_c, as well as thermal power N_u of a fan — cooler system — (8), (9). Therefore, on the basis of the mathematical model given above, it is possible to make the best selection of fan cooperating with a particular cooler of air so that for a steady state thermal power of a fan — cooler system could have the maximum value. Exemplary calculations were done for two different states of inflow air for four different types of duct fans cooperating with cooler GCCP-115. Fans WLE-603A, WLE-804AM, WLE-1003B and WLE-1004A were used. The results of calculations were presented in tables 1 and 2 and also graphically in figures 2 and 3, where black points refer to a fan — cooler system and empty points (circles) refer to a cooler itself cooperating with a particular type of a fan.

In order to make the comparison of cooperation between a particular fan and a given cooler for two cases: cooler and less humid inflow air and warmer and wetter inflow air possible, all the calculations were done on the basis of the same assumptions of absolute pressure of air as $b = 1080\,\text{hPa}$ and temperature of cooling water at its entry to a cooler as $t_w(s = L) = t_{w0} = 10°\text{C}$. Intensity of cold water flow was assumed as equal to $Q_w = 2.5\,\text{kg/s}$. As it has already been mentioned, two different states of air at the entry to a fan were tested:
$A: t_1 = 24°\text{C}$ and $\varphi_1 = 80\%$, and $B: t_1 = 32°\text{C}$ and $\varphi_1 = 90\%$; where φ_1 signifies relative humidity of air at the entry to a fan.

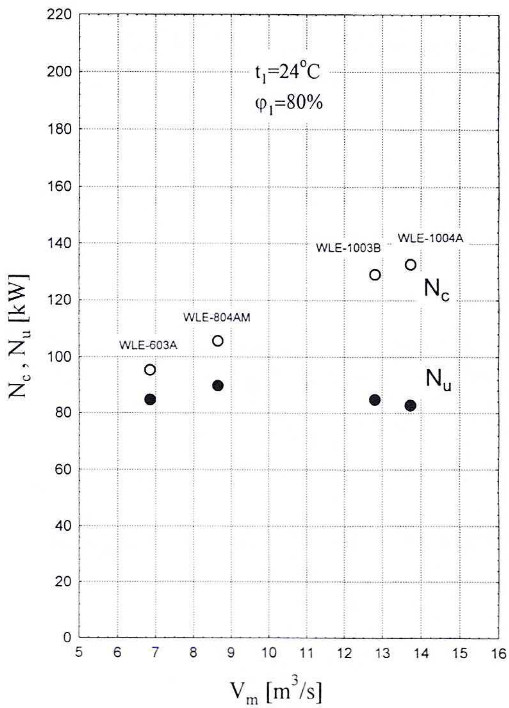

Fig. 2. The results of calculations of thermal power of cooler (N_c) and a cooler — fan system (N_u) for different types of duct fans for the state of inflow air $A: t_1 = 24°\text{C}$, $\varphi_1 = 80\%$

TABLE 1

The results of calculations of power of cooler (N_c) and a cooler — fan system (N_c) for configurations under consideration, for the state of inflow air $A: t_1 = 24°C$, $\varphi_1 = 80\%$

b	hPa	1080			
t_1	°C	24			
φ_1	%	80			
x_1	kg/kg	0,01405			
h_1	kJ/kg	59,90			
t_{pr}	°C	20,33			
ϱ	kg/m^3	1,2390			
ϱ_{m1}	kg/m^3	1,2564			
t_{w0}	°C	10			
R	kg/m^7	10,72			
Q_w	kg/s	2,5			
Fan		WLE-603A	WLE-804AM	WLE-1003B	WLE-1004A
V_m	m^3/s	6,85	8,65	12,80	13,73
Q	kg/s	8,49	10,72	15,86	17,01
Δp	Pa	503	802	1756	2021
η_w	%	33	45	52	57
Δt_{went}	°C	1,22	1,43	2,71	2,85
b_f	—	0,104	0,129	0,186	0,198
t_2	°C	25,22	25,43	26,71	26,85
h_2	kJ/kg	61,17	61,38	62,70	62,84
s_w	m	1,29	1,56	2,20	2,30
t_{wp}	°C	19,39	20,32	22,45	22,78
t_{e3}	°C	17,89	18,37	19,24	19,38
x_{e3}	kg/kg	0,01203	0,01241	0,01312	0,01323
t_3	°C	18,65	19,29	20,63	20,86
x_3	kg/kg	0,01224	0,01262	0,01329	0,01339
h_3	kJ/kg	49,92	51,52	54,56	55,03
N_c	kW	95,43	105,67	129,23	132,84
N_u	kW	84,71	89,84	84,83	82,85

TABLE 2

The results of calculations of power of cooler (N_c) and a cooler — fan system (N_u) for configurations under consideration, for the state of inflow air $B: t_1 = 32°C$, $\varphi_1 = 90\%$

b	hPa	1080			
t_1	°C	32			
φ_1	%	90			
x_1	kg/kg	0,02565			
h_1	kJ/kg	97,88			
t_{pr}	°C	30,15			
ϱ	kg/m^3	1,1849			
ϱ_{m1}	kg/m^3	1,2153			
t_{w0}	°C	10			
R	kg/m^7	10,72			
Q_w	kg/s	2,5			
Fan		WLE-603A	WLE-804AM	WLE-1003B	WLE-1004A
V_m	m^3/s	6,85	8,65	12,80	13,73
Q	kg/s	8,12	10,25	15,17	16,27
Δp	Pa	503	802	1756	2021
η_w	%	33	45	52	57
Δt_{went}	°C	1,28	1,50	2,84	2,98
b_f	—	0,104	0,129	0,186	0,198
t_2	°C	33,28	33,50	34,84	34,98
h_2	kJ/kg	99,23	99,45	100,87	101,01
s_w	m	0,61	0,79	1,34	1,43
t_{wp}	°C	25,89	27,02	29,26	29,61
t_{c3}	°C	26,09	26,74	27,79	27,95
x_{c3}	kg/kg	0,02009	0,02090	0,02227	0,02248
t_3	°C	26,85	27,62	29,10	29,34
x_3	kg/kg	0,02067	0,02151	0,02290	0,02311
h_3	kJ/kg	80,28	83,17	88,11	88,87
N_c	kW	153,75	166,89	193,43	197,57
N_u	kW	142,80	150,72	148,08	146,50

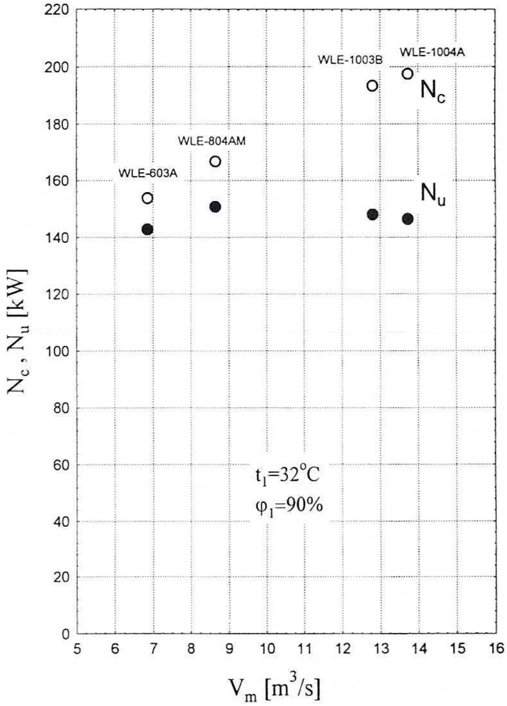

Fig. 3. The results of calculations of thermal power of cooler (N_c) and a cooler — fan system (N_u) for different types of duct fans for the state of inflow air $B: t_1 = 32°C$, $\varphi_1 = 90\%$

In table 1 the results of calculations of power N_c and N_u, as well as some indirect results, for the state of inflow air *A* were presented for four types of fans; in table 2 similar data were given for state *B*.

Comparing the obtained results for different cooler — fan systems it can be concluded that the best effects, as far as heat in concerned, both for the state of air at the entry *A* and *B* are ensured by linking cooler GCCP-115 with fan WLE-804AM. Fans, cooperating with a particular cooler, of greater and greater flow rate, increase thermal power of cooler N_c (for the unchanged remaining parameters of air and water) and in case of the power of the whole system N_u initial increase can be observed and after reaching its maximum value, a fall which is due to great increases in temperature of air caused by the work of a fan.

When designing and using such systems of cooling air, it should be remembered that there is the best, as far as the greatest power of a fan — cooler system is concerned, value of the quantity of cooled air.

Artykuł powstał w ramach projektu badawczego nr 9 T12A 052 12 finansowanego przez Komitet Badań Naukowych.

REFERENCES

Filek K., Holesz K., Nowak B., Roszkowski J., 1998a. Termodynamiczny model chłodzenia powietrza górniczą chłodnicą przeponową bez kondensacji pary wodnej. Archives of Mining Sciences, 43, 1, 19—34.

Filek K., Holesz K., Nowak B., Roszkowski J., 1998b. Weryfikacja opisu działania górniczej chłodnicy przeponowej bez kondensacji pary wodnej. Archives of Mining Sciences, 43, 4, 503—515.

Filek K., Holesz K., Nowak B., Roszkowski J. 1998c. Eksperymentalne badania górniczej chłodnicy przeponowej przy chłodzeniu powietrza z kondensacją pary wodnej. Wydawnictwa AGH, Górnictwo, Kraków, vol. IV, 249—263.

Filek K., Holesz K., Nowak B., Roszkowski J., 1999a. Chłodzenie powietrza górniczą chłodnicą przeponową z kondensacją pary wodnej. Archives of Mining Sciences, 44, 1, 3—21.

Filek K., Nowak B., Roszkowski J., 1999b. Przybliżona metoda obliczania parametrów termodynamicznych czynnika chłodzonego i chłodzącego w przeciwprądowej chłodnicy powietrza. Materiały Międzynarodowej Konferencji na temat: „Najnowsze osiągnięcia w zakresie przewietrzania kopalń oraz zwalczania zagrożeń pożarowych, gazowych i klimatycznych", Szczyrk, 241—252.

Filek K., Nowak B., Roszkowski J., 1999c. Moc cieplna układu wentylator — przeponowa chłodnica powietrza. Materiały Międzynarodowej Konferencji na temat: „Górnictwo — praktyka i nauka. Spojrzenie w XXI wiek", Kraków.

Gutkowski K., 1972. Chłodnictwo. Wybrane zagadnienia obliczeniowe. Wydawnictwa Naukowo-Techniczne, Warszawa.

Häussler W., 1971. Zastosowanie wykresu *i-x* w inżynierii sanitarnej. Arkady, Warszawa.

Holesz K., 1997. Temperatura i wilgotność powietrza w górniczych chłodnicach przeponowych w stanach nieustalonych. Praca doktorska, Kraków.

Informacja zespołu d/s klimatyzacji i przewietrzania głębokich kopalń, 1984. Główne Biuro Studiów i Projektów Górniczych, nr 2.

Markefka P., Stefanowicz T., 1986. Wentylacyjne sposoby i środki zapobiegania nadmiernym przyrostom temperatury powietrza w drążonych wyrobiskach korytarzowych. Budownictwo Węglowe. Projekty — problemy, nr 6.

Nowak B., Filek K., 1996. Temperatura i wilgotność chłodzonego powietrza w podziemnych wyrobiskach górniczych przewietrzanych wentylacją opływową. Wydawnictwa AGH. Rozprawy, Monografie, 48.

Nowak B., Łukosz M., 1999. Wpływ natężenia przepływu powietrza przez górniczą przeciwprądową chłodnicę przeponową o działaniu pośrednim na jej współczynnik bocznikowania. Wydawnictwa AGH, Górnictwo, Kraków, vol. III (w druku).

Pawiński J., Roszkowski J., Strzemiński J., 1995. Przewietrzanie kopalń. Śląskie Wydawnictwo Techniczne, Katowice.

Roszczynialski W., Trutwin W., Wacławik J., 1992. Kopalniane pomiary wentylacyjne. Wydawnictwo „Śląsk", Katowice.

Wacławik J., Cygankiewicz J., Knechtel J., 1995. Warunki klimatyczne w kopalniach głębokich. Poradnik. Biblioteka Szkoły Eksploatacji Podziemnej. Kraków.

REVIEW BY: PROF. DR HAB. INŻ. ZYGMUNT KOLENDA, KRAKÓW

Received: 30 August 1999.