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BERNARD NOWAK*, ZBIGNIEW KUCZERA*

HEAT POWER DETERMINATION OF DV-290 REFRIGERATOR'S EVAPORATOR ON THE BASIS OF THERMODYNAMIC PARAMETERS OF INLET AIR

OKREŚLENIE MOCY CIEPLNEJ PAROWNIKA CHŁODZIARKI DV-290 NA PODSTAWIE PARAMETRÓW TERMODYNAMICZNYCH POWIETRZA WLOTOWEGO

The present paper introduces a method for calculating the thermal power of DV-290 mining air cooler's evaporator. The power usually differs from the nominal power given by the manufacturer. The thermodynamic parameters of cooled air are not obtained as a result of in situ measurements, but in indirect manner that is by determining the evaporation and condensation's pressure values of R407C refrigerant. The pressure dependencies formulated as a function of air enthalpy at the evaporator's inlet were obtained using calculations of a computer program which solves the system of equations describing heat and mass transfer in the refrigerator's compressor on the basis of previous measurements of air performed before and after its cooling. The obtained dependencies are demonstrated in a graphical (fig. 2 and fig. 3) and analytical (the regression equations (19) and (20)) manner, the values of correlation coefficients are also presented. For the known evaporation and condensation pressure values of the refrigerant, and thus for its basic physical parameters the complete thermal power of the evaporator was determined, that is its: air cooling overt power, dehumidification occult power, temperature, relative humidity and specific humidity of air after its cooling.

In addition, using the mentioned method, the capacity of DV-290 refrigerator's evaporator is provided for the given thermodynamic parameters of air before cooling, along with air thermodynamic parameters after cooling.

Keywords: air conditioning of mines, air cooling, compression refrigerator, thermal power

W pracy zaproponowano metodę obliczania mocy cieplnej parownika górniczej chłodziarki powietrza DV-290. Moc ta zazwyczaj jest różna od mocy znamionowej podanej przez jej producenta. Wymaganą znajomość parametrów termodynamicznych schłodzonego powietrza otrzymuje się, nie jak dotychczas w wyniku ich pomiarów *in situ*, lecz drogą pośrednią wyznaczając najpierw wartości ciśnień parowania i skraplania czynnika chłodniczego R407C. Odpowiednie zależności tych ciśnień w funkcji jednostkowej entalpii powietrza na wlocie parownika otrzymano, na podstawie wcześniejszych pomiarów parametrów powietrza przed i po jego schłodzeniu, z obliczeń utworzonym programem komputerowym rozwiązują-

* AGH UNIVERSITY OF SCIENCE AND TECHNOLOGY, A. MICKIEWICZA 30 AVE., 30-059 KRAKOW, POLAND

cym układ równań opisujący wymianę ciepła i masy w chłodziarce sprężarkowej. Uzyskane zależności przedstawiono w sposób graficzny (rys. 2 i rys. 3) oraz analityczny – równania regresji (19) i (20), podając też wartości współczynników korelacji. Dla znanych wartości ciśnień parowania i skraplania czynnika chłodniczego, a więc także i jego podstawowych parametrów fizycznych, korzystając z wymienionego programu komputerowego, wyznaczono, w funkcji jednostkowej entalpii powietrza na wlocie parownika, całkowitą jego moc cieplną z podziałem na moc jawną ochładzania powietrza, utajoną moc osuszania powietrza, temperaturę, wilgotność względną i wilgotność właściwą powietrza po jego ochłodzeniu.

Podano też, dla przykładowych zadanych parametrów termodynamicznych powietrza przed jego schłodzeniem obliczone wspomnianą metodą, moce parownika chłodziarki DV-290 oraz parametry termodynamiczne powietrza po schłodzeniu.

Słowa kluczowe: klimatyzacja kopalń, chłodzenie powietrza, chłodziarka sprężarkowa, moc cieplna

1. Introduction

In order to improve working conditions in terms of heat in underground mine headings, when ventilation methods are insufficient, active air cooling is used. For this purpose, compression refrigerators operating in direct or indirect cooling system are most commonly used. However, their operating efficiency is very frequently limited by their capability to transfer the heat removed from fresh air and produced by compressor's operation to the air consumed by an evaporative cooler. The efficiency of this air cooling process is determined by the efficiency of use of cooling capacity available in a given system. The assumption that the evaporator's rated heat power, given by the refrigerator manufacturer, is a constant value is in many cases groundless.

The heat power values of DV-290 refrigerator evaporator, measured in mine operating conditions with proper heat collection by DV-290 refrigerator condenser, the rated value of which was of 290 kW, were not constant and varied in the range from 210 to 308 kW. These fluctuations were accompanied by fluctuations of unit enthalpy of air at the inlet to the evaporator from 1162 to 1361 kW. The said evaporator of DV-290 refrigerator with R407C cooling agent was directly built into the principal pressure air duct. Fresh air was provided by two ES9 500/80 serially connected fans, arranged in a circulating stream.

In this paper, a method of determining the cooling capacity of a directly operating refrigerator evaporator, based only on a known unit enthalpy of air at the evaporator's inlet, is presented; and not, as it has been so far, on the product of mass flow rate of cooled air and of specific enthalpy difference between inlet and outlet of evaporator. So far the required specific enthalpy of cooled air has been calculated on the basis of experimentally determined thermodynamic parameters of air leaving an evaporator. In the method presented in this paper, the parameters of cooled air are determined indirectly by solving a system of equations which describe processes of heat and mass exchange running in refrigerator in a steady state. For this purpose, the vaporization and condensation pressure of a given cooling agent, which circulates through refrigerator and undergoes thermodynamic processes, have to be known so that the physical parameters of coolant may be determined on the basis of numerical values, specified in tables and corresponding to: cooling agent in a liquid phase, cooling agent at phase boundary and area of superheated steam. In air refrigerators used in mining sector these pressures are not always measured. Assuming that heat is sufficiently collected by the refrigerator's condenser, the values of the foregoing cooling agent pressures have been determined indirectly. For this purpose, at the same initial conditions, heat power of evaporator calculated on the basis of previous measurements made on site was compared with the heat power obtained in numeric solution of the said equation system forming mathematical model of air refrigerator operation. Superheating and supercooling of cooling agent were assumed at 5°C and – 2°C, respectively. In the mathematical description of the discussed air cooling system, a concentrated nature of its individual components (evaporator, condenser) was assumed, which causesthe parameters of air, cooling agent and water cooling the condensernot to change continuously, along these components, but in a stepwise manner. The said system of equations was introduced in the paper of (Nowak & Filek, 2009, 2010b), and the correctness of created mathematical descriptions of actual heat and mass exchange processes running in DV-290 and TS-300 refrigerators was evaluated in the paper (Nowak et al., 2010a).

2. Method of evaporator's heat power determination on the basis of unit enthalpy of air at evaporator inlet

The following system of equation, describing the operation of directly operating refrigerator in steady-state conditions, may be given after (Nowak & Filek, 2009, 2010b):

$$\ln\frac{t_1-t_{f0}}{t_{c2}-t_{f0}}=\frac{k_pF_p(t_1-t_{c2})}{Q_m(1-b_f)[c_p(t_1-t_{c2})+c_w(t_1x_1-t_{c2}x_{c2})+(r_p-c_c\,t_{c2})(x_1-x_{c2})]} \tag{1}$$

$$\ln\frac{t_1-t_{f0}}{t_{c2}-t_{f0}}=\frac{k_pF_p(t_1-t_{c2})}{Q_f[c_{pf0}(t_{f2}-t_{f0}\,\chi_{p1})+(r_{pf0}-c_{cf0}t_{f0})(1-\chi_{p1})]} \tag{2}$$

$$\ln\frac{t_{fk}-t_{w1}}{t_{fk}-t_{w2}}=\frac{k_sF_s}{Q_wc_c} \tag{3}$$

$$\ln\frac{t_{fk}-t_{w1}}{t_{fk}-t_{w2}}=\frac{k_sF_s(t_{w2}-t_{w1})}{Q_f(c_{pfk}t_{f1}+r_{pfk}-c_{cfk}t_{f2})} \tag{4}$$

$$t_{f2}=(t_{f1}+273{,}15)\left(\frac{p_k}{p_0}\right)^{\frac{\kappa-1}{\kappa}}-273{,}15 \tag{5}$$

$$\chi_{p1}=\frac{t_{f2}\,c_{cfk}-t_{f0}\,c_{cf0}}{t_{f0}\,(c_{pf0}-c_{cf0})+r_{pf0}} \tag{6}$$

$$x_{c2}=x_n(t_{c2})=\frac{379{,}8\cdot10^u}{b-610{,}6\cdot10^u}\qquad\text{where}\qquad u=\frac{7{,}5t_{c2}}{t_{c2}+237{,}29} \tag{7}$$

$$\begin{cases} t_2 = \dfrac{(c_p t_{c2} + c_w t_{c2} x_{c2} + r_p x_{c2})(1-b_f) + (c_p t_1 + c_w t_1 x_1 + r_p x_1) b_f - r_p x_2}{c_p + c_w x_2} \\ x_2 = \begin{cases} x_{c2}(1-b_f) + x_2 b_f & \text{for} \quad x_{c2}(1-b_f) + x_1 b_f \le x_n(t_2) \\ x_n(t_2) & \textit{for} \quad x_{c2}(1-b_f) + x_1 b_f > x_n(t_2) \end{cases} \end{cases} \tag{8}$$

The evaporator's heat power N_p [W] (divided into sensible air cooling power N_{pj} [W] and latent air drying power N_{pu} [W]) may be calculated using the following formulae:

$$N_{pj} = Q_m \left[c_p (t_1 - t_2) + c_w (t_1 x_1 - t_2 x_2) \right] \tag{9}$$

$$N_{pu} = Q_m (r_p - c_c t_2)(x_1 - x_2) \tag{10}$$

$$N_p = N_{pj} + N_{pu} \tag{11}$$

where:

- b_f — evaporator's bypass factor understood in accordance with (Kołodziejczyk & Rubik, 1976) as a ratio of a conventional mass of air being cooled down to its total mass [-],
- c_c, c_p, c_w — specific heat of: water, dry air at constant pressure and steam at constant pressure, respectively, [J/(kg·K)],
- c_{cf0}, c_{cfk}, c_{pf0}, c_{pfk} — specific heat of: liquid cooling agent in evaporator, liquid cooling agent in condenser, vapours of cooling agent at constant pressure in evaporator, vapours of cooling agent at constant pressure in condenser, respectively, [J/(kg·K)],
- F_p, F_s — surface of heat exchange in evaporator and condenser, respectively [m^2],
- k_p, k_s — coefficient of heat transfer in a membrane of evaporator and condenser, respectively, [W/(m^2·K)],
- p_0, p_k — absolute pressure of cooling agent in evaporator and condenser, respectively, [bar],
- Q_f, Q_m — mass flow of cooling agent and dry air in evaporator, respectively, [kg/s],
- r_p, r_{pf0}, $r_{pf\mathrm{k}}$ — vaporization/condensation heat: of water and cooling agent in evaporator and cooling agent in condenser, [J/kg],
- t_c, t_{c2} — temperature of portion of air being cooled in evaporator: average, at the outlet [°C]; at evaporator inlet $t_{c1} = t_1$,
- t_{f0}, t_{fk} — temperature of cooling agent vaporization in evaporator temperature of cooling agent condensation in condenser, [°C],
- t_{fp}, t_{fp2} — temperature of cooling agent in evaporator: average, at outlet [°C]; it was assumed $t_{fp1} = t_{f0}$,
- t_{f1}, t_{f2} — temperature of cooling agent in condenser: at inlet/outlet, respectively, [°C],
- t_1, t_2 — temperature of air at evaporator inlet/outlet, respectively, [°C],
- t_{w1}, t_{w2} — temperature of air at condenser inlet/outlet, respectively [°C],

x_{c2} — specific humidity of air portion being cooled down at evaporator outlet, [kg/kg],

x_1, x_2 — specific humidity of air at evaporator inlet and outlet, [kg/kg],

x_n — specific humidity of air saturated in the given temperature at evaporator outlet [kg/kg],

κ — isentropic exponent of cooling agent vapour [-],

χ_{p1} — dryness grade of cooling agent vapour at evaporator inlet [-].

Actual heat power of DV-290 refrigerator evaporator was determined for each of the 12 measurement variants of thermodynamic parameters of air, before and after cooling was carried out in mine conditions. In cross-section of air duct the following parameters were measured: dry-bulb (t_s) and wet-bulb (t_m) temperatures at evaporator inlet and outlet – with the use of Assmann's Aspirated Psychrometer, dynamic pressure at a distance of 1 m from the evaporator inlet – with the use of Pitot-Prandtl tube, and absolute air pressure (b) with the use of μBar – type barometer on the floor of a roadway in its section corresponding to the evaporator inlet. The measured and determined thermodynamic parameters of air at the evaporator inlet and outlet are given in Tables 1 and 2, and relationship between calculated values of evaporator's heat power N_p [kW] and unit air enthalpy h_1 [kW] at evaporator inlet is graphically plotted on Fig. 1 and described by the following equation (Kuczera, 2011):

$$N_p = -3136{,}8603 + 1095{,}3939 \cdot \log(h_1) \qquad (12)$$

In the latter case Statistica 8.0 (StatSoft, 2006) software was used. Correlation coefficient between the variables considered is of 0,8492, which demonstrates the significance of the relationship between them.

TABLE 1

Measured and calculated parameters of air at inlet to evaporator of DV-290 refrigerator

Item no.	**Air parameters at evaporator inlet**								
	Dry – bulb temperature t_s [°C]	Dry – bulb temperature t_m [°C]	Absolute air pressure b [Pa]	Volumetric flow rate of air V [m³/s]	Specific air humidity x [kg/kg]	Relative air humidity φ [%]	Mass flow rate of dry air Q_m [kg dry air/s]	Specific air enthalpy I [kJ/kg]	Unit air enthalpy h [kW]
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
1	32,2	27,2	110383	13,07	0,01882	67,43	16,04	80,58	1292,47
2	32,8	28,2	110130	12,69	0,02034	70,14	15,43	85,10	1313,08
3	32,4	28,4	109828	12,99	0,02094	73,58	15,78	86,22	1360,53
4	32,6	28,0	110282	12,42	0,02005	70,03	15,10	84,15	1270,62
5	32,8	27,8	109931	12,21	0,01968	67,80	14,85	83,41	1238,60
6	32,2	27,4	108977	12,10	0,01944	68,71	14,6	82,17	1199,63
7	32,6	27,2	110043	13,07	0,01871	65,35	15,91	80,71	1284,14
8	32,8	28,2	110097	12,69	0,02035	70,14	15,43	85,12	1313,47
9	32,4	28,2	109796	12,99	0,02059	72,36	15,78	85,32	1346,38
10	32,6	27,8	110205	12,42	0,01971	68,84	15,10	83,28	1257,46
11	32,8	27,6	109926	12,21	0,01933	66,64	14,85	82,51	1225,28
12	32,0	26,8	108982	12,10	0,01850	66,22	14,61	79,55	1162,23

TABLE 2

Measured and calculated parameters of air at outlet from evaporator of DV-290 refrigerator.

Item no.	Air parameters at evaporator outlet						
	Dry – bulb temperature t_s [°C]	Dry – bulb temperature t_m [°C]	Specific air humidity x [kg/kg]	Relative air humidity φ [%]	Specific air enthalpy I [kJ/kg]	Unit air enthalpy h [kW]	**Heat power of evaporator** $\mathbf{N_p}$ **[kW]**
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**
1	23,0	23,0	0,01624	100	64,43	1033,53	**254,96**
2	23,8	23,8	0,01711	100	67,48	1041,19	**266,92**
3	23,6	23,6	0,01694	100	66,84	1054,70	**299,59**
4	23,2	23,2	0,01646	100	65,20	984,54	**280,81**
5	23,4	23,4	0,01672	100	66,07	981,15	**253,14**
6	23,0	23,0	0,01645	100	64,97	948,54	**246,88**
7	23,2	23,2	0,01650	100	65,30	1038,98	**241,75**
8	23,6	23,6	0,01690	100	66,74	1029,74	**278,47**
9	23,2	23,2	0,01653	100	65,38	1031,69	**308,46**
10	23,2	23,2	0,01647	100	65,23	984,93	**267,78**
11	23,4	23,4	0,01672	100	66,07	981,15	**240,33**
12	23,0	23,0	0,01645	100	64,97	949,19	**210,15**

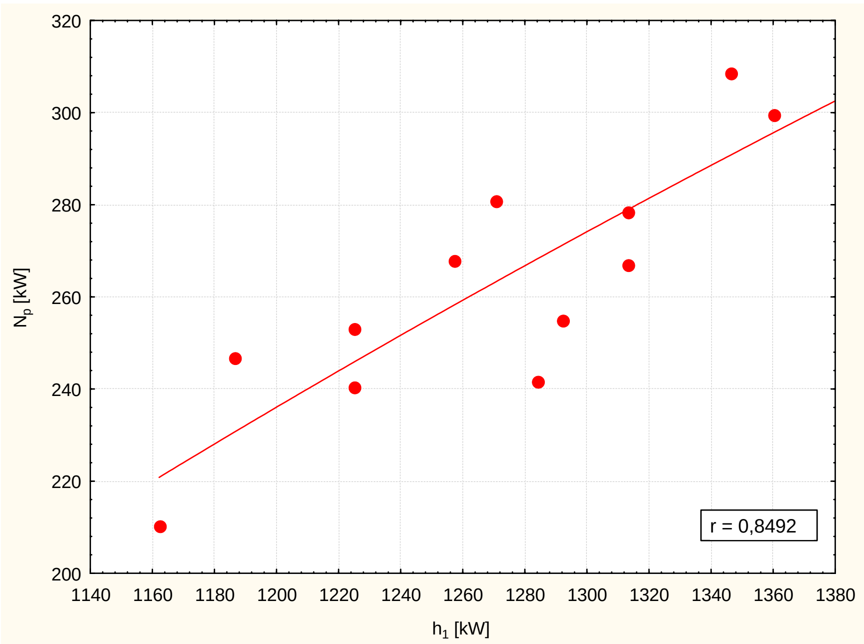

Fig. 1. Change of evaporator's heat power (N_p) in function of unit enthalpy of air at inlet to evaporator (h). Line – regression curve, circulars – measurement results

In addition to the foregoing, the following formulae were used (Pawiński et al., 2000; Roszczynialski et al., 1992):

– for calculation of air density ρ [kg/m^3]

$$\rho = \frac{0{,}003484}{t_s + 273{,}15} \cdot \left[b \; - 0{,}3781 \cdot \left(610{,}6 \cdot w - 6{,}6176 \cdot 10^{-4} (t_s - t_m) b \right) \right] \tag{13}$$

where:

b — absolute air pressure [Pa],
t_m — wet-bulb temperature of air [°C],
t_s — dry-bulb temperature of air [°C];

– for calculation of relative air humidity φ [-]

$$\varphi = \frac{610{,}6 \cdot w \; - 6{,}6176 \cdot 10^{-4} (t_s - t_m) \cdot b}{610{,}6 \cdot w'} \tag{14}$$

where: $w = 10^{\frac{7{,}5 \cdot t_m}{237{,}29 + t_m}}$ and $w' = 10^{\frac{7{,}5 \cdot t_s}{237{,}29 + t_s}}$

– for calculation of specific air humidity x [kg H_2O steam/kg dry air]

$$x = \frac{379{,}793 \cdot w - 4{,}1161 \cdot 10^{-4} (t_s - t_m) \cdot b}{b - 610{,}6 \cdot w + 6{,}6178 \cdot 10^{-4} (t_s - t_m) \cdot b} \tag{15}$$

– for calculation of dry air mass flow rate Q_m [kg dry air/s]

$$Q_m = \frac{V \cdot \rho}{1 + x} \tag{16}$$

where: V — volumetric air flow [m^3/s];

– for calculation of specific air enthalpy I [J/kg]

$$I = c_p \cdot t_s + x(c_w \cdot t_s + r_p) \tag{17}$$

– for calculation of unit (per time unit) enthalpy of air h [W]

$$h = Q_m \cdot I \tag{18}$$

On the basis of the foregoing mathematical description of operation of an air compression refrigerator for mining applications, the software was developed, which allows, among other things, to determethe vaporization pressure p_0 in the evaporator and condensation pressure p_k in the condenser of a given cooling agent, corresponding to the known heat power of evaporator. Using this software, the values of p_0 and p_k pressures were calculated for experimentally determined the evaporator's heat power values specified in Table 2. Vaporization pressure p_0 in the evaporator and condensation pressure p_k in the condenser of R407C cooling agent determined in this way were correlated with unit enthalpy of air at the evaporator inlet (h_1). The results of research are summarized in Table 3, graphically plotted on Fig. 2 & 3 and analytically presented (formulae 19 and 20).

$$p_0 = 42{,}855 - 11{,}463 \cdot \log(h_1) \tag{19}$$

$$p_\mathrm{k} = 28{,}3042 - 3{,}5156 \cdot \log(h_1) \tag{20}$$

where vaporization and condensation pressure (p_0 and p_k, respectively) is expressed in [bar], and unit enthalpy of air at evaporator inlet h_1 – in [kW]. The foregoing formulae were verified within the range of unit air enthalpy: 1100 kW < h_1 < 1400 kW. As previously, when creating the formulae (19) and (20), Statistica 8.0 software was used. Calculated correlation coefficients between the variables in the equations (19) and (20) amount to: –0,7369 and –0,9958, respectively.

TABLE 3

Calculated values of vaporization and condensation pressure (p_0/p_k, respectively) of R407C cooling agent

Item no.	**Calculated values of vaporization and condensation pressure of R407C cooling agent**			
	Heat power of evaporator N_p [kW]	Unit enthalpy of air at evaporator inlet h_1 [kW]	Vaporization pressure of cooling agent p_0 (bar)	Condensation pressure of cooling agent p_k (bar)
1	254,96	1292,47	7,38	17,37
2	266,92	1313,08	7,32	17,33
3	299,59	1360,53	6,87	17,29
4	280,81	1270,62	7,00	17,39
5	253,14	1225,28	7,38	17,46
6	246,88	1186,54	7,35	17,50
7	241,75	1284,14	7,63	17,37
8	278,47	1313,47	7,14	17,35
9	308,46	1346,38	6,71	17,31
10	267,78	1257,46	7,17	17,40
11	240,33	1225,28	7,57	17,44
12	210,15	1162,23	7,87	17,53

3. Numeric example

Considering the foregoing PC software and formulae (9)÷(20) exemplary calculations were made.

Parameters at the evaporator's inlet of a by-pass factor $b_f = 0{,}14$ of DV-290 air compression refrigerator are:

- air pressure $b = 110$ kPa;
- volumetric airflow rate $V = 12$ m^3/s;
- dry-bulb air temperature $t_s = 31$ °C;
- wet-bulb air temperature $t_m = 29$ °C;
- specific air humidity $x_1 = 0{,}22613$ kg/kg;

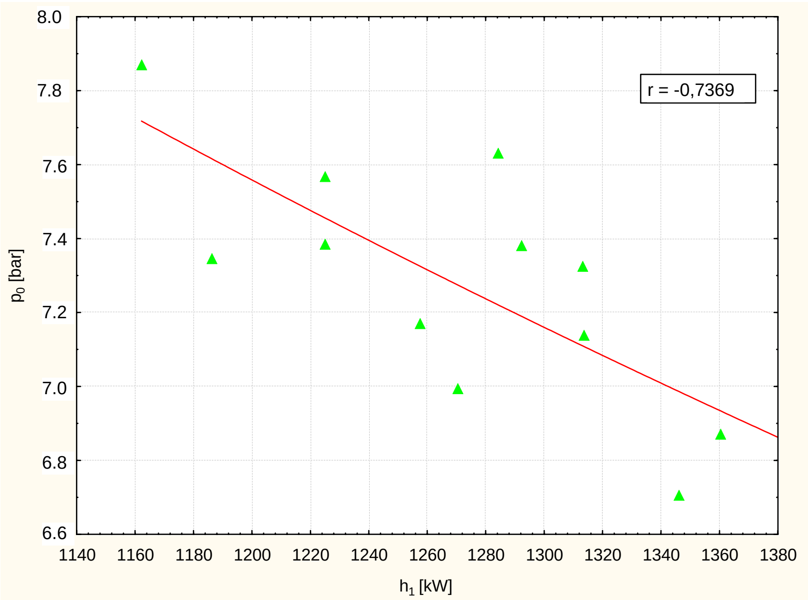

Fig. 2. Change of cooling agent vaporization pressure (p_0) in function of unit enthalpy of air (h) at evaporator inlet. Line – regression curve, triangles – measurement results

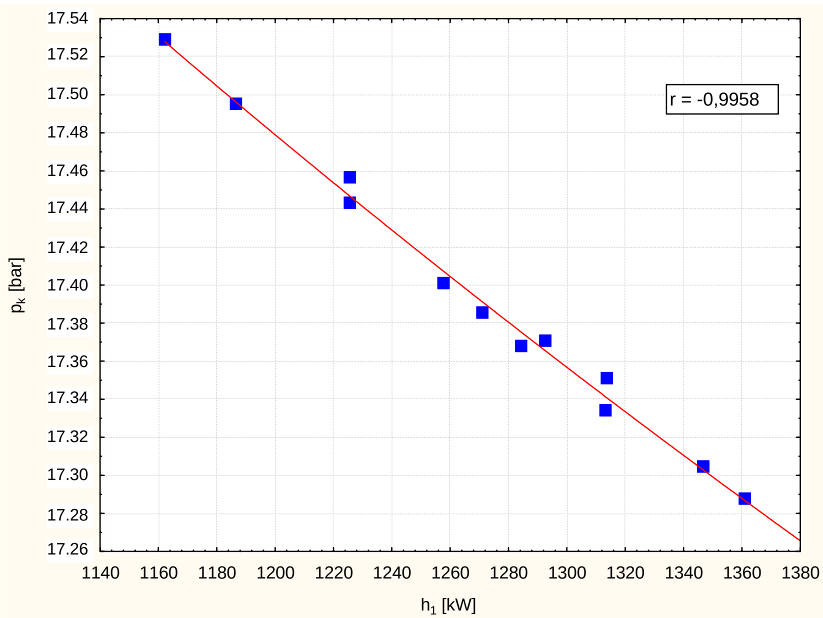

Fig. 3. Change of cooling agent condensation pressure (p_k) in function of unit enthalpy of air (h) at evaporator inlet. Line – regression curve, squares – measurement results

– relative air humidity $\varphi_1 = 85{,}9\%$;
– specific enthalpy of air calculated with the use of formula (17) $I_1 = 89{,}04$ kJ/kg;
– unit enthalpy of air calculated with the use of formula (18) $h_1 = 1299{,}07$ kW;
– vaporization pressure of R407C cooling agent in evaporator calculated with the use of formula (19) $p_0 = 7{,}16$ bar;
– condensation pressure of R407C cooling agent in condenser calculated with the use of formula (19) $p_0 = 17{,}36$ bar;

With the use of developed PC software the following parameters were calculated:
– total heat power of evaporator $N_p = 265{,}1$ kW;
– sensible air cooling power $N_{pj} = 98{,}7$ kW;
– latent air drying power $N_{pu} = 166{,}4$ kW;
– temperature of air after cooling-down $t_2 = 24{,}8$°C;
– specific humidity of air after cooling-down $x_2 = 0{,}17853$ kg/kg;
– relative humidity of air after cooling-down $\varphi_2 = 98{,}3\%$.

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