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THE EFFECT OF TEMPERATURE GLIDE OF R407C REFRIGERANT ON THE POWER OF EVAPORATOR IN AIR REFRIGERATORS**WPLYW POŚLIZGU TEMPERATURY CZYNNIKA CHŁODNICZEGO R407C NA MOC PAROWNIKA CHŁODZIARKI POWIETRZA**

The article discusses the effect of the phenomenon of temperature glide of zeotropic refrigerants on thermal power of an evaporator in an air compression refrigerator. Zeotropic mixtures are subject to phase transitions, the process of which significantly differs from that of homogeneous refrigerants. In contrast to homogeneous refrigerants, where boiling and condensing processes take place at a constant temperature, for the zeotropic mixtures it is essential to know the vapor quality to unambiguously determine the temperature at which the evaporation process is initiated. The R407C refrigerant serves as an example to describe the method of determining the initial temperature of the evaporation process taking into account the effect of temperature glide. The developed formula (7) has been based on a proven linear course of isobars in the two-phase region (Fig. 5) and thus determining a polynomial describing their angle of inclination (8). In addition, temperature calculation formulas (9) and specific enthalpy (10) of dry saturated vapor of the R407C refrigerant have been presented as well. This approach allows to determine the temperature of the R407C refrigerant at the inlet to the evaporator without the required knowledge of its vapor quality. The previously used simplified methods for determining the temperature of a refrigerant at the inlet to the evaporator result in considerable deviations in calculated power of the evaporator compared with its actual value. The presented calculation example involving mine air compression refrigerator of TS-450P type shows that relative deviations of the evaporator thermal power may even exceed 20%. This example compares two simplified methods for determining zeotropic evaporating temperature of a refrigerant used in comparative calculations of refrigerants with the method presented in this article.

Keywords: R407C refrigerant, temperature glide, air refrigerator, evaporator

W artykule przedstawiono wpływ zjawiska poślizgu temperatury zeotropowych czynników chłodniczych na moc cieplną parownika sprężarkowej chłodziarki powietrza. Mieszaniny zeotropowe podlegają przemianom fazowym, których przebieg znacznie różni się od czynników jednorodnych. W odróżnieniu od jednorodnych czynników chłodniczych, których procesy wrzenia i skraplania odbywają się przy stałej temperaturze, dla mieszanin zeotropowych do jednoznacznego określenia temperatury początku procesu parowania niezbędna jest znajomość stopnia suchości pary. Na przykładzie czynnika chłodniczego R407C

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opisano metodę wyznaczania temperatury początkowej procesu parowania uwzględniającą zjawisko poślizgu temperatury. Opracowana zależność (7) powstała w oparciu o udowodniony liniowy przebieg izobar w obszarze pary mokrej (rys. 5) i określeniu na tej podstawie wielomianu opisującego ich kąt nachylenia (8). Dodatkowo przedstawiono wzory obliczeniowe temperatury (9) oraz entalpii właściwej (10) pary nasyconej suchej czynnika chłodniczego R407C. Takie podejście do problemu pozwala na wyznaczenie temperatury czynnika chłodniczego R407C na wlocie do parownika bez wymaganej znajomości stopnia suchości pary czynnika. Dotychczas stosowane uproszczone metody wyznaczania temperatury czynnika chłodniczego na wlocie do parownika powodują znaczne odstępstwa obliczonej na ich podstawie mocy parownika od jego wartości rzeczywistej. Przedstawiony przykład obliczeniowy dotyczący górniczej sprężarkowej chłodziarki powietrza pośredniego działania typu TS-450P pokazuje, że odchyłki względne mocy cieplnej parownika mogą przekraczać nawet ponad 20%. W przykładzie obliczeniowym porównano dwie uproszczone metody określenia temperatury parowania zeotropowego czynnika chłodniczego stosowane w obliczeniach porównawczych czynników chłodniczych z metodą zaprezentowaną w niniejszym artykule.

Słowa kluczowe: czynnik chłodniczy R407C, poślizg temperatury, chłodziarka powietrza, parownik

1. Introduction

In many sectors of the economy – such as in mining for example, the use of refrigerators determines not only the possibility of carrying out works or their duration, but it also has a direct impact on the safety of miners. Both the increasingly lower levels of mining and intensive exploitation make comfort of work worse in underground mines. Improving climatic conditions by increased ventilation of workings is limited, therefore better thermal conditions are achieved through the use of refrigeration devices. With respect to the manner the heat reception, refrigeration equipment most commonly used in mining can be divided into:

- direct action refrigeration equipment, in which the evaporator is directly used to cool the air,
- indirect action refrigeration equipment, in which the evaporator is used to cool the water supplied to gate road and longwall air refrigerators.

Regardless of the method of heat reception, refrigerators use working fluids with low boiling points referred to as refrigerants. In general, they are divided into homogeneous substances and mixtures, among which azeotropic and zeotropic solutions can be distinguished. Their choice is determined by thermodynamic, physico-chemical, economic as well as, increasingly, ecological aspects. The search for new, environment-friendly refrigerants is due to the destructive impact on the ozone layer and the contribution to the global warming effect of the refrigerants which belong to the CFC (chlorofluorocarbon) and HCFC (hydrochlorofluorocarbon) groups (e.g., R11, R12, R113, R114 and R22). Both for the devices already in service, as well as for new refrigeration systems, multicomponent mixtures with properties similar to those withdrawn working fluids were developed (Termo Schiessl, 2009). Both direct (TS-150÷450B) (Nowak et al, 2010) and indirect (TS-150÷450P) action Termospec refrigerators, widely used in underground mining, use chlorine-free three-component mixture of HFC (hydrofluorocarbon) group – R407C, which is a replacement for freon R22. It belongs to the group of zeotropic mixtures, whose components are subject to phase transitions separately, by their individual boiling points. Therefore, a constant temperature of evaporation (condensation) can not be attributed to a given pressure of evaporation (condensation), since this process occurs at a certain range of temperatures. This phenomenon is called temperature glide. It is a characteristic property of zeotropic mixtures and distinguishes them from homogeneous and azeotropic mixtures. Azeotropic solutions have such a composition that during boiling and condensation processes they behave as homogeneous refrigerants.

2. Temperature glide

Temperature glide of a refrigerant is referred to as the total change of its saturation temperature during the phase transition occurring at a constant pressure (Swinney et al., 1998). The value of this glide is dependent primarily upon the volatility of components of a mixture forming a given refrigerant and decreases with its increasing pressure. During the phase transitions, different boiling points of the individual components of a mixture result in changing the composition of the solution. Both of these phenomena have a significant impact on heat transfer (Rajapaksha, 2007).

The phenomenon of temperature glide can be used in countercurrent heat exchangers to improve the cooling system efficiency by appropriate glide matching of temperature profiles both of the refrigerant and of the fluid involved in the heat transfer (Mulroy et al., 1994; Lee et al., 2002; Gao & Zhao, 2008). Temperature profiles in a countercurrent heat exchanger for a single-component refrigerant and for a zeotropic mixture have been shown in Figure 1 (Rajapaksha, 2007; Jin & Zhang, 2011).

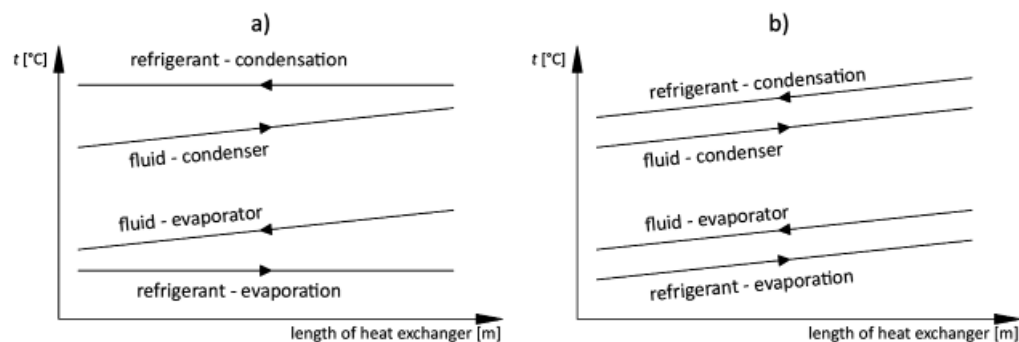


Fig. 1. Temperature profiles (t) in heat exchangers during phase transitions: (a) of a single-component refrigerant, (b) of a zeotropic mixture with temperature glide (Rajapaksha, 2007; Jin & Zhang, 2011)

For a zeotropic mixture, proper glide matching of the temperature profile (Fig. 1b) has such an effect that throughout the whole heat exchange process we receive equal or close difference in temperature of the fluids involved in the heat transfer. As demonstrated in the works (Mulroy et al., 1994) and (Lee et al., 2002), such a process leads to the increased value of the COP (Coefficient of Performance). Not always, however, the temperature profile of a refrigerant during phase transitions is linear. Depending on the composition of the mixture, the variation of both the temperature and the enthalpy in the two-phase region can be non-linear. If a more volatile component prevails in the solution, the nonlinearity is concave, and if a less volatile component predominates, nonlinearity is convex. Non-linear variation of the refrigerant parameters during phase transition gives rise to the so-called “pinch points”, or areas where the temperature difference of the refrigerant and the fluid is minimal – Figure 2 (Venkatarathnam et al., 1996; Venkatarathnam & Srinivasa Murthy, 1999; Jin & Zhang, 2011). “Pinch point” could occur within the heat exchanger – the authors (Venkatarathnam & Srinivasa Murthy, 1999) call it the type I “pinch point”, or at both ends of the heat exchanger – type II “pinch point”. The occurrence of

pinch points reduces the thermal power of the exchangers, which is connected with a need to increase the heat exchange surface to provide the desired capacity. Therefore, while selecting a refrigerant for a particular cooling system, it is necessary to know the value of the temperature glide and the nature of its variation in the two-phase region.

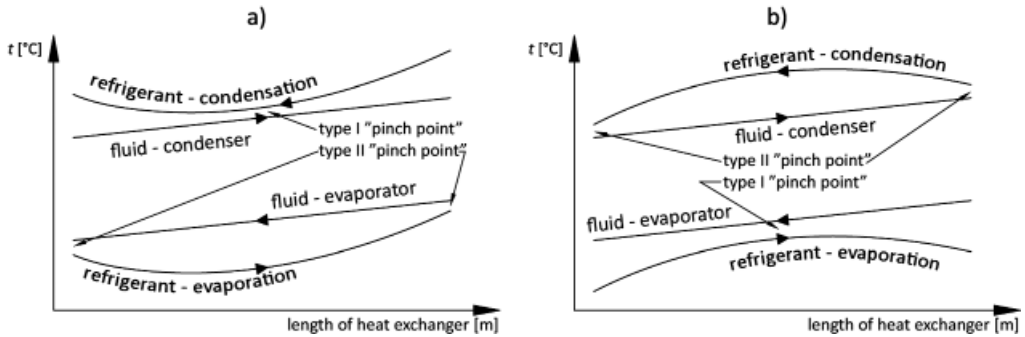


Fig. 2. Nonlinearity (a) concave and (b) convex of the temperature during phase transitions (Venkatarathnam & Srinivasa Murthy, 1999; Jin & Zhang, 2011)

For the purposes of comparison with homogeneous refrigerants (e.g., calculating the cooling capacity coefficient), condensation and evaporation temperatures of zeotropic mixtures are generally averaged or, in accordance with the standard PN-EN 12900 (2007), their values are assumed as those for dry saturated vapor. Such an approach, however, can not be applied in the calculation of the thermal power of heat exchangers, as this will result in too much divergence from the actual conditions. Therefore, while selecting an evaporator or a condenser to the refrigeration system, it is necessary to know the actual temperatures of a refrigerant, both at the inlet and the outlet of the heat exchanger. For the zeotropic mixtures, there is a problem to determine the temperature of a refrigerant at the evaporator inlet. This problem does not apply to condensers, where the actual process of condensation takes place from the state of dry saturated vapor until the refrigerant reaches the state of a saturated liquid. So the temperatures at the beginning and at the end of the condensation process are the functions of pressure, exclusively. However, in the case of evaporators, due to the fact that the process of boiling of a refrigerant initiates in the two-phase region, in order to unambiguously determine the state of the refrigerant at the inlet to the evaporator, it is essential to know the pressure and the vapor quality, or the temperature and the vapor quality.

The evaporator thermal power can be written down as heat flux penetrating the diaphragm of the heat exchanger from the heat transfer fluid to the refrigerant, from the widely known relationship:

$$\dot{Q}_o = k_p \cdot F_p \cdot \Delta t_{mp} \quad (1)$$

where:

\dot{Q}_o — evaporator thermal power [W],

k_p — heat transfer coefficient of the evaporator diaphragm [W/(m²·K)],

F_p — surface area of heat exchange in the evaporator [m²],
 Δt_{mp} — log mean temperature difference between heat transfer fluid and a refrigerant in the evaporator [K].

In the case of a countercurrent evaporator cooling water, log mean temperature difference is:

$$\Delta t_{mp} = \varepsilon_{\Delta t} \cdot \frac{\Delta t_{1p} - \Delta t_{2p}}{\ln \frac{\Delta t_{1p}}{\Delta t_{2p}}} \quad (2)$$

In the case of using a homogeneous refrigerant and a zeotropic mixture, the temperature differences Δt_{1p} and Δt_{2p} have been shown in Figure 3.

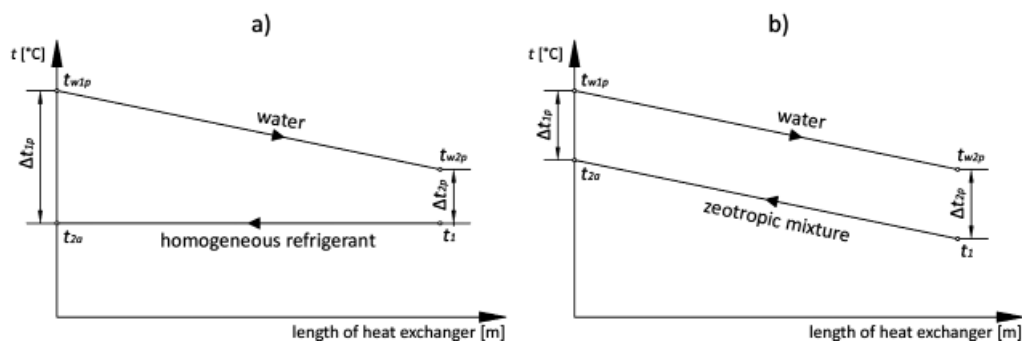


Fig. 3. Temperature changes of refrigerants in the evaporator for (a) a homogeneous refrigerant and (b) a zeotropic mixture

Additionally, in accordance with (Theodore, 2011), in the formula (2), a diminishing coefficient $\varepsilon_{\Delta t}$ which takes into account the cross flow of the refrigerants in the evaporator, was used. Its value depends upon the parameters P and R defined by the following relationships:

$$P = \frac{t_{2a} - t_1}{t_{w1p} - t_1} \quad (3)$$

$$R = \frac{t_{w1p} - t_{w2p}}{t_{2a} - t_1} \quad (4)$$

where:

- t_{w1p} — evaporator inlet water temperature [°C],
- t_{w2p} — evaporator outlet water temperature [°C],
- t_1 — refrigerant wet vapor temperature at the evaporator inlet [°C],
- t_{2a} — superheated refrigerant vapor temperature at the evaporator outlet [°C].

and it can be read from the following nomogram – Figure 4.

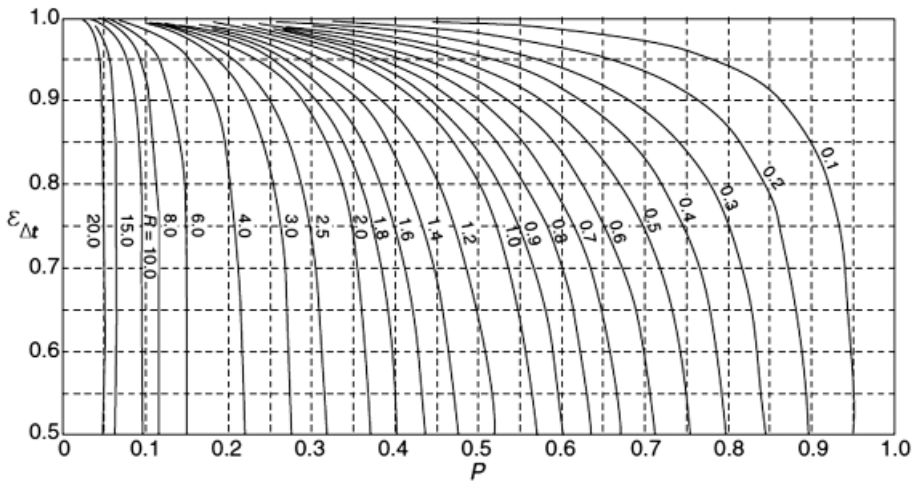


Fig. 4. Nomogram to determine the correction factor $\epsilon_{\Delta t}$ (Theodore, 2011)

Finally, the log mean temperature difference, taking into account the diminishing coefficient, can be determined by the following relationship (Theodore, 2011):

$$\Delta t'_{mp} = \frac{\sqrt{(t_{w1p} - t_{w2p})^2 + (t_{2a} - t_1)^2}}{\ln \left\{ \frac{t_{w1p} + t_{w2p} - t_1 - t_{2a} + \sqrt{(t_{w1p} - t_{w2p})^2 + (t_{2a} - t_1)^2}}{t_{w1p} + t_{w2p} - t_1 - t_{2a} - \sqrt{(t_{w1p} - t_{w2p})^2 + (t_{2a} - t_1)^2}} \right\}} \quad (5)$$

It is evident therefrom that, in order to calculate the thermal power of the evaporator working with a zeotropic refrigerant, it is necessary to know both the temperature of the refrigerant at the inlet as well as at the outlet of the evaporator. Assuming in calculations that the temperature of evaporation is the arithmetic mean of the temperatures of a saturated liquid and dry saturated vapor for a given pressure, or proceeding in accordance with the requirements of the standard PN-EN 12900, i.e. assuming the temperature of dry saturated vapor as the evaporation temperature, may result in significant deviations in the calculated value of the log mean temperature difference, and thus in the power of the evaporator.

3. Temperature of the R407C refrigerant at the evaporator inlet

The R407C refrigerant, compared to other zeotropic mixtures, is characterized by a considerable value of the temperature glide Δt , which, depending on the pressure of the refrigerant, varies from 2.4 to 7°C (in the pressure range of 1-40 bar) – Table 1. In the extreme case (the pressure of 1 bar), assuming for the calculation the mean value of the temperature of a refrigerant at a constant pressure as the temperature of evaporation, it is possible to overestimate the temperature at the

evaporator inlet by about 3°C (depending on the vapor quality), and following the guidelines of the standard PN-EN 12900, even by about 6°C. That is why, when assessing heat exchangers working with zeotropic mixtures with respect to their thermal power, it is necessary to know the changes in temperature of the circulating refrigerant during its phase transitions.

TABLE 1

The values of temperature glide of refrigerants (Lemmon et al., 2002)

P [bar]	R401A	R404A	R407A	R407B	R407C	R409A	R410A	R413A
1.0	5.74	0.75	6.42	4.29	7.00	8.50	0.08	2.76
5.0	5.06	0.54	5.65	3.62	6.21	7.75	0.10	1.59
10.0	4.55	0.44	5.09	3.19	5.63	7.10	0.11	1.11
15.0	4.11	0.37	4.61	2.84	5.13	6.51	0.12	0.84
20.0	3.70	0.32	4.16	2.51	4.66	5.93	0.12	0.66
25.0	3.29	0.26	3.70	2.18	4.18	5.34	0.12	0.52
30.0	2.86	0.21	3.22	1.82	3.66	4.71	0.11	0.39
35.0	2.39	0.13	2.67	1.39	3.09	4.01	0.10	0.27
40.0	1.82	-	1.98	-	2.40	3.18	0.08	-

To unambiguously determine the refrigerant temperature at the evaporator inlet (in the two-phase region) it is essential to know its pressure and vapor quality. The problem is to determine the vapor quality which, by definition, is the ratio of the mass of dry saturated vapor m'' to the total mass of wet vapor, equal to the sum of the mass of the fluid m' and the mass of dry saturated vapor m'' :

$$x = \frac{m''}{m' + m''} \quad (6)$$

where:

- m' — the mass of saturated fluid [kg],
- m'' — the mass of dry saturated vapor [kg].

To eliminate the necessity of knowing the vapor quality, this article presents a new method for determining the temperature of the R407C refrigerant at the evaporator inlet. It uses detailed data regarding the course of variance in the temperature of a refrigerant in the two-phase region as well as thermodynamic parameters on the boundary line of dry saturated vapor of the R407C zeotropic mixture. Taking into consideration the data obtained from the REFPROP 7 program developed by the National Institute of Standards and Technology, USA (Lemmon et al., 2002), the temperature profile of the R407C refrigerant for pressures in the range from 1 to 20 bar at 0.1 bar was drawn. The above-mentioned pressure range results from a typical operating range of compression refrigeration units (e.g. mine compression refrigerators of TS-300, TS-350 and TS-450P types). For a given pressure, temperature and specific enthalpy of wet vapor of a refrigerant was determined for the value range of vapor quality from 0 to 1 with a variance at 0.025. In this way, for each isobar drawn, 41 results of the temperature and specific enthalpy were obtained. The total of 7831 results were obtained. Figure 5 shows the course of the isobars of the R407C refrigerant in the two-phase region. For the reason of the Figure clarity, the representation of the

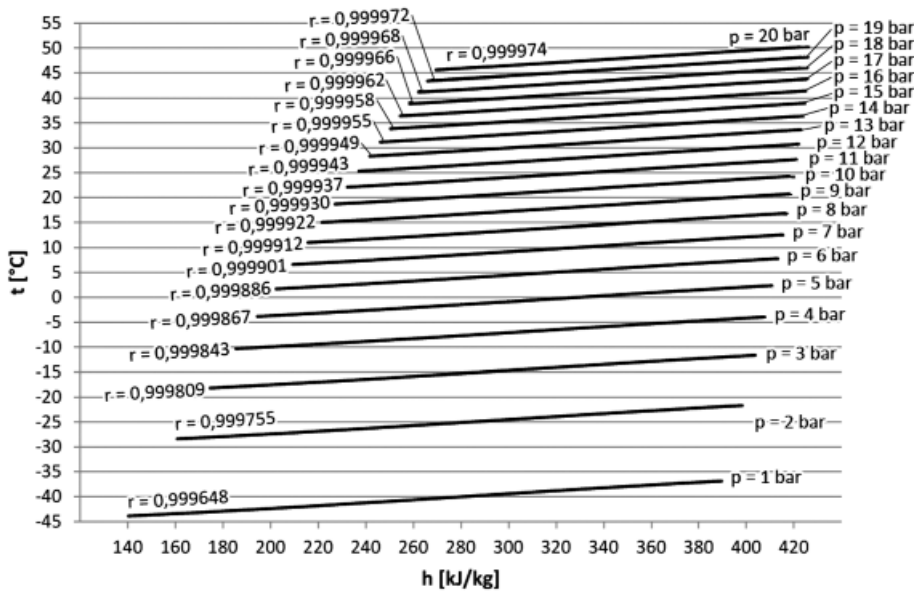


Fig. 5. The isobars of the R407C refrigerant in the two-phase region

isobars was limited to the R407C refrigerant in the two-phase region, in the system temperature – specific enthalpy for the refrigerant pressure variable at 1 bar.

Then, for each of the 191 isobars, Pearson correlation coefficients were determined. Arithmetical mean value of the obtained linear correlation coefficients for all the analyzed isobars is 0.99. These values prove a very strong linear relationship between the variables. It can therefore be concluded that the temperature profile of the R407C refrigerant during phase transitions is linear. This statement allows to determine the temperature t_1 of a refrigerant at the evaporator inlet, based on the knowledge of the isobar β , inclination angle, the temperature and specific enthalpy of dry saturated vapor of a refrigerant (h_2'', t_2''), as well as on the specific enthalpy of evaporation starting point h_1 – Figure 6. The required temperature of the refrigerant at the evaporator inlet is defined by the following formula:

$$t_1 = t_2'' - (h_2'' - h_1) \cdot \text{tg}\beta \quad (7)$$

where:

t_2'' — dry saturated vapor temperature for the evaporating pressure [°C],

h_2'' — specific enthalpy of dry saturated vapor for the evaporating pressure [kJ/kg],

h_1 — specific enthalpy of wet vapor at the evaporator inlet for the evaporating pressure [kJ/kg],

$\text{tg}\beta$ — tangent of the isobar inclination angle in the two-phase region [(°C·kg)/kJ].

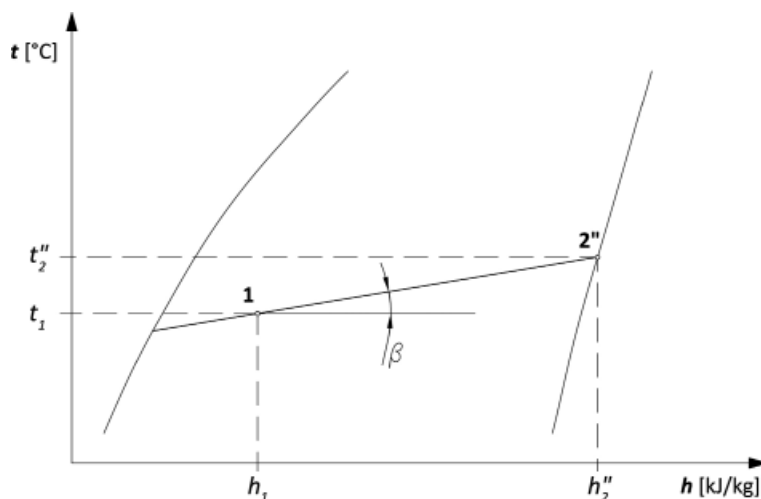


Fig. 6. Schematic diagram to determine the temperature of a refrigerant at the evaporator inlet

The tangent value of the inclination angle of the isobars in the two-phase region ($\text{tg}\beta$) were calculated for the isobars in the range of 1 to 20 bar, and presented by the sixth degree polynomial as a function of pressure:

$$\text{tg}\beta = \sum_{n=0}^6 a_n \cdot p^n \quad (8)$$

where:

- a_n — coefficient of a polynomial,
- p — evaporating pressure of a refrigerant [bar].

Coefficients of the polynomial were determined using the STATISTICA 10 program (StatSoft, 2011). For this purpose, one of the least squares nonlinear estimation method was used – the Levenberg – Marquardt method. The Levenberg – Marquardt method is a modification (extension) of the Gauss – Newton algorithm. Using a least squares loss function, as in the case of the Gauss -Newton algorithm, for finding the least squares parameter estimates, there is no need to calculate (or approximately estimate) partial derivatives of the second order. Instead, in each iteration, the algorithm solves a system of linear equations to calculate the gradient (StatSoft, 2006).

For the developed relationship, coefficients of correlation and determination were calculated, as well as absolute and relative deviations between the prescribed equation and the values obtained from the REFPROP program. The calculation results together with the determined coefficients of the polynomial have been shown in Table 2.

Coefficients of the polynomial, correlation coefficient, coefficient of determination and $\text{tg}\beta$ deviations from the data values

Quantity	Unit	Value	
Correlation coefficient R	[-]	0.9999954308	
Coefficient of determination R^2	[-]	0.9999908617	
Absolute deviation $\Delta\text{tg}\beta$	[(°C·kg)/kJ]·10 ⁶	mean	maximum
		0.720734	3.449888
Relative deviation $\delta\text{tg}\beta$	[%]	mean	maximum
		0.002443	0.011933
a_0	The unit of the polynomial coefficients results from the form of the equation (8) as well as values of the determined and determining parameters	2.88328290537497E-02	
a_1		7.02462704975242E-05	
a_2		5.67310045616829E-06	
a_3		-1.17114453303298E-06	
a_4		9.04271353192151E-08	
a_5		-3.36119011642961E-09	
a_6		4.87083236275841E-11	

The calculated coefficients of correlation and determination for the developed relationship amount to 0.9999954308 and 0.9999908617, accordingly. However, the maximum relative deviation slightly exceeded 0.01%. The resulting values of the coefficients of correlation and determination as well as absolute and relative deviations allow to evaluate highly the glide matching of the results obtained by the prescribed equation to the values from the REFPROP program. It can therefore be concluded that the inclination angle of a straight describing isobaric change of a refrigerant in the evaporator is a measure of the temperature glide.

Both the temperature and specific enthalpy of dry saturated vapor are functions of the refrigerant pressure. As in the work (Bonca et al, 1998) they are described by the sixth degree polynomials:

$$t_2^n = \sum_{n=0}^6 a_n \cdot \ln p^n \quad (9)$$

$$h_2^n = \sum_{n=0}^6 a_n \cdot \ln p^n \quad (10)$$

As in the case of the relationship for $\text{tg}\beta$, to determine the coefficients of the polynomials (9) and (10), STATISTICA 10 program was used. Statistical analysis was performed using data from the REFPROP 7 program. For pressures in the range of 1-20 bar with a variation of 0.05 bar, the values of corresponding temperature and specific enthalpy of dry saturated vapor were read out. A total of 381 data for each of the relationships were used for the calculations.

The determined coefficient polynomials, correlation coefficients, coefficients of determination, absolute and relative deviations for the relationships describing the temperature of dry saturated vapor and specific enthalpy of dry saturated vapor are to be found in Tables 3 and 4, respectively.

TABLE 3

Coefficients of the polynomial, correlation coefficient, coefficient of determination and temperature deviations of dry saturated vapor from the data values

Quantity	Unit	Value	
Correlation coefficient R	[-]	0.999	
Coefficient of determination R^2	[-]	0.999	
Absolute deviation $\Delta t_2''$	[°C]	mean	maximum
		0.000272	0.001107
Relative deviation $\delta t_2''$	[%]	mean	maximum
		0.003309	0.441741
a_0	The unit of the polynomial coefficients results from the form of the equation (9) as well as values of the determined and determining parameters	-3.68981071607199E+01	
a_1		2.03920218833464E+01	
a_2		2.04538194212078E+00	
a_3		3.00186534293768E-01	
a_4		-4.71214391231593E-02	
a_5		2.46601044587668E-02	
a_6		-3.49109664847636E-03	

TABLE 4

Coefficients of the polynomial, correlation coefficient, coefficient of determination and specific enthalpy deviations of dry saturated vapor from the data values

Quantity	Unit	Value	
Correlation coefficient R	[-]	0.999	
Coefficient of determination R^2	[-]	0.999	
Absolute deviation $\Delta h_2''$	[kJ/kg]	mean	maximum
		0.003163	0.011835
Relative deviation $\delta h_2''$	[%]	mean	maximum
		0.000759	0.003038
a_0	The unit of the polynomial coefficients results from the form of the equation (10) as well as values of the determined and determining parameters	3.89578164842409E+02	
a_1		1.20792696715190E+01	
a_2		7.86006021788295E-02	
a_3		1.19010164173223E+00	
a_4		-8.98718633786115E-01	
a_5		3.04637172534377E-01	
a_6		-4.63215077131162E-02	

Both relationships are characterized by very high values of correlation and regression coefficients close to unity. Both the absolute and the relative deviations are very small, and their maximum values for the temperature of dry saturated vapor are 0.001107°C and 0.441741%, respectively, and for specific enthalpy – 0.011835 kJ/kg and 0.003038%.

The value of specific enthalpy of wet vapor at the evaporator inlet h_1 can be calculated assuming the isenthalpic process of a refrigerant in the expansion valve.

Using the equations (7)-(10) it is possible to determine the actual value of the R407C refrigerant temperature at the evaporator inlet without having to know the vapor quality of a refrigerant. This procedure greatly simplifies thermal calculations for evaporators working with the R407C zeotropic mixture.

4. Calculation example

Calculation example involves a shell and tube evaporator used in mine indirect action refrigerators TS-450P, manufactured by Termospec. The calculations compared thermal power of the evaporator obtained depending on the manner in which the temperature of a refrigerant at the evaporator inlet was determined:

- temperature of a refrigerant calculated using the method presented in the article – variant 1,
- assumed as an arithmetic mean of the temperature of saturated fluid and saturated vapor of a refrigerant at a constant pressure – variant 2,
- assumed as the temperature of dry saturated vapor – variant 3.

The necessary structural data of the evaporator were obtained from its documentation (Termospec, 2008) and the required calculation parameters from (Życzkowski, 2012). The calculations assume a constant superheat of the refrigerant vapor at the outlet of the evaporator $\Delta t_p = 3\text{K}$.

The data for the calculations are presented in Table 5.

TABLE 5

Data for the calculation example

No.	Specification	Unit	Value
1	Evaporator inlet water temperature t_{w1p}	[°C]	14.2
2	Evaporator outlet water temperature t_{w2p}	[°C]	4.8
3	Refrigerant pressure p	[bar]	4.4
4	Wet vapor specific enthalpy at the evaporator inlet for evaporating pressure h_1	[kJ/kg]	241.16
5	Superheating of the refrigerant vapor Δt_p	[K]	3.0
6	Surface area of heat exchange in the evaporator F_p	[m ²]	50.48
7	Heat transfer coefficient of the evaporator diaphragm k_p	[W/(m ² ·K)]	795.47

In all the variants during the calculation of the temperature of dry saturated vapor, the equation (9) was used, and having taken into account superheating of the refrigerant, the temperature of the superheated refrigerant vapor at the evaporator outlet t_{2a} was determined. Further calculations involved determining the refrigerant temperature at the evaporator inlet, the logarithmic difference in temperature and power of the evaporator. The results have been summarized in Table 6.

TABLE 6

Results of calculations for specific variants

Calculation variant	t_2''	t_{2a}	h_2''	$\text{tg}\beta$	t_1	$\Delta t'_{mp}$	\dot{Q}_o
	[°C]	[°C]	[kJ/kg]	[(°C·kg)/kJ]	[°C]	[K]	[kW]
1	-1.31	1.69	408.87	0.0292	-6.20	10.59	425.4
2	-1.31	1.69	–	–	-4.46	9.84	394.9
3	-1.31	1.69	–	–	-1.31	8.36	335.6

When analyzing the obtained results, large differences between the temperatures of the refrigerant at the evaporator inlet can be noticed. Both the temperature calculated as an arithmetic mean of the temperatures of saturation of a liquid and the R407C refrigerant vapor (variant 2) as well as the temperature assumed for the calculations as the temperature of dry saturated vapor (variant 3) are clearly higher than the temperature determined by the method described by the authors (variant 1). Both approximate methods overestimate the temperature at the evaporator inlet by 1.74°C (variant 2) and 4.89°C (variant 3), which results in underestimation of the logarithmic value of the temperature difference by 0.75K (variant 2) and 2.23K and, consequently, in underestimating the power of the evaporator by 30.5 kW and 89.8 kW, respectively. In percentage, the power of the evaporator for variants 2 and 3 is lower by 7.2 and 21.1%, respectively. Thus, assuming for the calculations the temperature of a refrigerant at the evaporator inlet as the temperature of dry saturated vapor (variant 3) results in a significant reduction in its power.

5. Conclusions

The evaporation process of zeotropic refrigerants takes place at variable temperature, which in the thermal calculations requires the determination of the temperature of a refrigerant both at the inlet and at the outlet of the evaporator. Determination of the final temperature of the evaporation process requires only the knowledge of pressure (dry saturated vapor state). However, in the case of the parameters of a zeotropic refrigerant, unambiguous determination of thermodynamic parameters in the two-phase region (the beginning of the evaporation process) requires the knowledge of two independent variables (usually pressure and vapor quality). In this article, having stated the linearity of the isobars during phase transitions of the R407C refrigerant and having described the inclination angle of its isobars by the sixth degree polynomial as a function of pressure, a new method for determining the temperature of the R407C refrigerant at the evaporator inlet was presented. Such a procedure allows to take into account the phenomenon of temperature glide of a refrigerant in thermal calculations of the evaporator and to determine the temperature of a refrigerant at the evaporator inlet without knowing the vapor quality. Sample calculations show that the most commonly used simplified methods of determining the temperature of a refrigerant at the evaporator inlet underestimate its power, even by more than 20%.

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