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EFFICIENCY OF REFRIGERATION PLANTS OPERATION TAKING INTO ACCOUNT AMBIENT TEMPERATURE CHANGE

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ABSTRACT

An entropic and statistical method of analysis allows calculating losses in different refrigeration plant components, comparing operational efficiency of different refrigeration systems.

This method of analysis was applied in order to investigate refrigeration systems with two working temperature levels (evaporation temperatures minus 35°C and minus 10°C) for retail application, taking into account ambient temperature change. A description of the calculation method was given.

A comparison of efficiency refrigeration systems using R404A as a refrigerant with a system working on a transcritical CO_2 cycle was made.

The degree of thermodynamic efficiency was calculated.

A schematic diagram, schedules of losses, COP, and the degree of thermodynamic efficiency depending on ambient temperature are given.

This result could not be achieved with a traditional refrigeration system comparison method using a coefficient of performance (COP) or seasonal efficiency as the only efficiency criterion.

Keywords: entropic and statistical method of analysis, CO_2, refrigeration cycle, transcritical refrigeration plant, degree of thermodynamic efficiency.

1. INTRODUCTION

One of the ways of reducing power consumption is by using a floating condensing temperature algorithm. According to this algorithm, condensing temperature changes depending on ambient temperature changing.

Applying this algorithm to the transcritical CO2 cycle allows the operation point to be changed from transcritical to subcritical, thus increasing energy efficiency.

Transition from HFC to nature refrigerants is very important for retail application (Schalenbourg, 2019) and (Skacanova and de Ona, 2019). Analysis of two main refrigeration cycles for retail application – a transcritical cycle with CO2 as a refrigerant with parallel compression and ejector (System 1) is shown on Fig. 1, and a refrigeration plant with two cycles with R404A which are independent from each other - one stage refrigeration cycle with single throttling and cycle with economizer (System 2) is shown on Fig. 2 – was made taking into account ambient temperature change and, as a result, discharge pressure change.

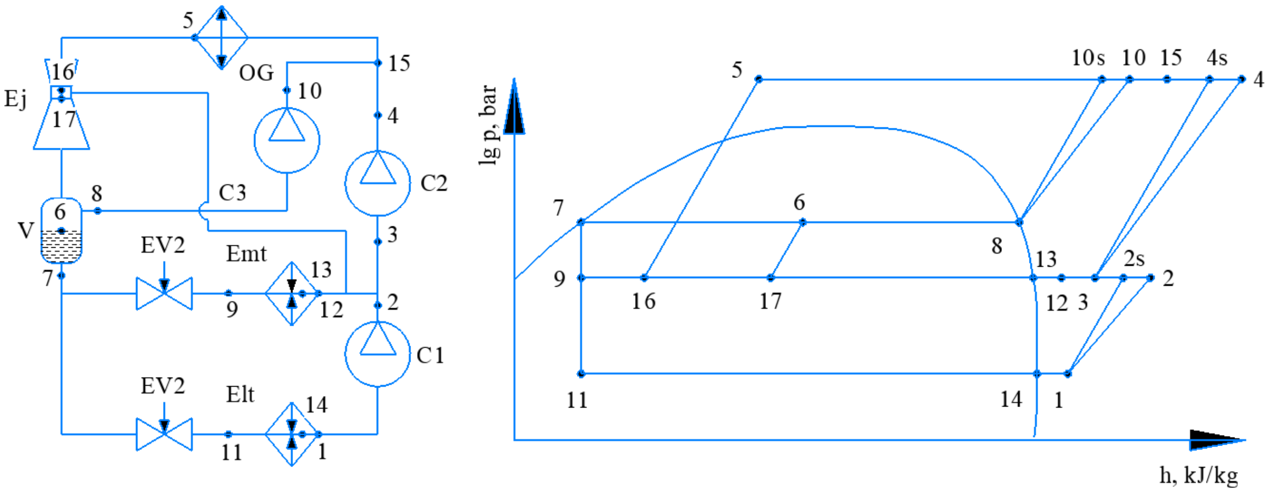

Figure 1: Transcritical refrigeration system with CO_2 parallel compression and ejector (System 1)

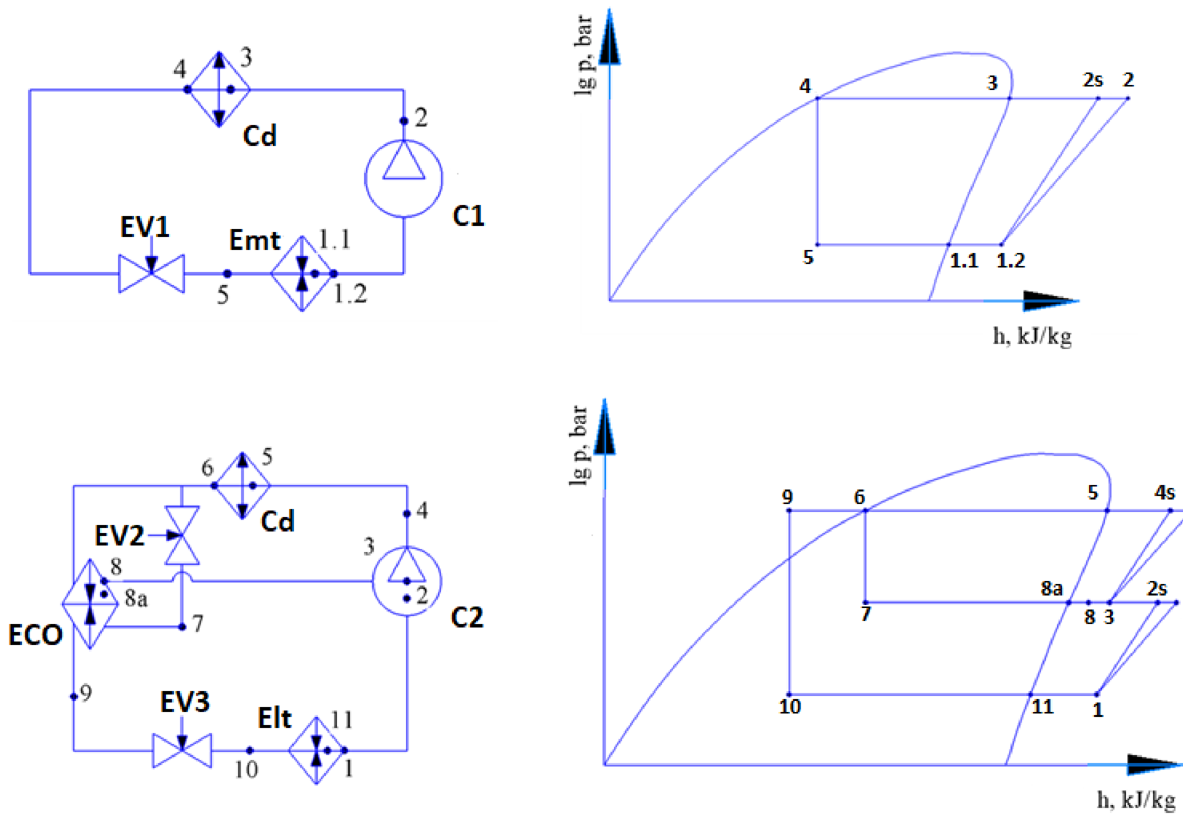

Figure 2: Refrigeration system with R404A (System 2)

2. ENTROPIC AND STATISTICAL ANALYSIS OF REFRIGERATION PLANTS

2.1. Initial data

Initial data is given in Table1.

Table1. Initial data for analysis

Evaporation temperature at low temperature (LT) level, °C (K)	-35 (238)
Evaporation temperature at medium temperature (MT) level, °C (K)	-10 (263)
Air temperature at LT level T_c^{LT}, °C (K)	-20 (253)
Air temperature at MT level T_c^{MT}, °C (K)	0 (273)
Cooling capacity at LT level Q_{LT}, kW	70
Cooling capacity at MT level Q_{MT}, kW	360

The city where the refrigeration plant is to be installed is Moscow. Data on annual ambient temperature changes was taken from Meteonorm software (Fig. 3).

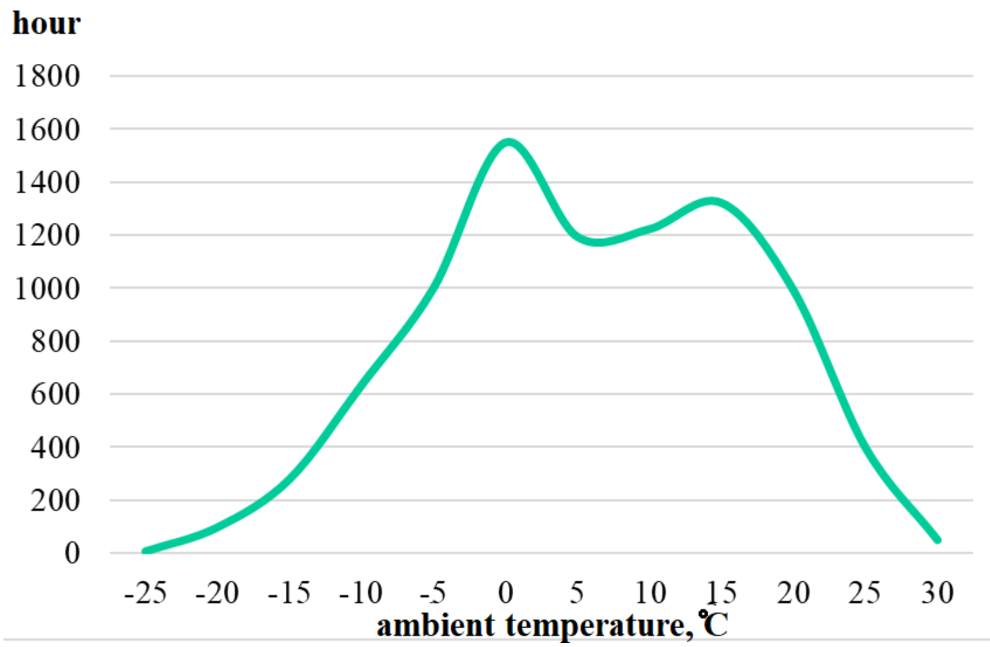

Figure 3: Ambient temperature

Minimum condensing temperatures were taken based on allowable working range of compressor (data was provided by the manufacturers):

LT refrigeration plant with R404A – plus 5°C;

MT refrigeration plant with R404A – plus 10°C;

Refrigeration plant with R744 – plus 14 °C.

Isentropic efficiency values were taken based on statistical data for the corresponding working parameters.

2.2. Calculation method

The entropic and statistical method of analysis was used as calculation method (Arkharov, 2011) and (Arkharov, 2014).

The following equations were used in the analysis (Arkharov and Shishov, 2014) and (Shishov and Talyzin, 2019):

Minimum specific work which is necessary for cold generation

$$l_{min} = q_o \times \frac{T_{env} - T_c}{T_c} \qquad \text{Eq.(1)}$$

Adiabatic compression work

$$l_s^{total} = \sum_{i=1}^{n} \left(l_{s_i} \times g_i \right) \qquad \text{Eq.(2)}$$

Actual specific compression work

$$l_{comp}^{total} = \sum_{i=1}^{n} \left(l_{comp_i} \times g_i \right) \qquad \text{Eq.(3)}$$

Degree of thermodynamic efficiency

$$\eta_{therm} = \frac{l_{min}}{l_{comp}} \qquad \text{Eq.(4)}$$

COP at adiabatic compression

$$\varepsilon_s = \frac{q_o}{l_s} \qquad \text{Eq.(5)}$$

Actual value of COP

$$\varepsilon_{act} = \frac{q_o}{l_{comp}} \qquad \text{Eq.(6)}$$

Part of compression work required to compensate for entropy production in condenser:

$$\Delta l_{cond} = (h_s - h_{dew}) - T_{env} \times (s_s - s_{dew}) + T_{env} \times (h_{dew} - h_{bubble}) \times \left(\frac{1}{T_{env}} - \frac{1}{T_{cond}} \right) \qquad \text{Eq.(7)}$$

Part of compression work required to compensate entropy production in gas cooler (energetic losses):

$$\Delta l_{gc} = (h_s - h_{out}) - T_{env} \times (s_s - s_{out}) \qquad \text{Eq.(8)}$$

Part of compression work required to compensate for entropy production in throttling processes (energetic losses):

$$\Delta l_{thr} = T_{env} \times (s_{thr\_out} - s_{thr\_in}) \qquad \text{Eq.(9)}$$

Part of compression work required to compensate for entropy production in evaporator (energetic losses):

$$\Delta l_{evap} = \left(h_{evap\_dew} - h_{evap\_in} \right) \times T_{env} \times \frac{T_c - T_o}{T_o \times T_c} + T_c \times (s_{evap\_sh} - s_{evap\_dew}) - (h_{evap\_sh} - h_{evap\_dew}) \qquad \text{Eq.(10)}$$

Specific adiabatic compression work is a sum of parts of compression work for compensation of entropy production in all processes of refrigeration cycle (energetic losses):

$$l_{s.c} = l_{min} + \Delta l_{gc} + \Delta l_{thr} + \Delta l_{evap} + \Delta l_{other} \quad \text{Eq.(11)}$$

Energetic losses in compressor:

$$\Delta l_{comp} = l_{comp} - l_{s.c} \quad \text{Eq.(12)}$$

Rated compression work:

$$l_{comp.c} = l_{s.c} + \Delta l_{comp} \quad \text{Eq.(13)}$$

Values of mass flows in every process were taking into account.

2.3. Analysys results

Graphs of refrigeration plant parameter changes are shown on Fig. 4 – Fig. 6.

The data about isentropic efficiency of compressor were taking as statistical date from selection software Select v. 7.14 provided by Emerson Climate Technologies (distributed for free).

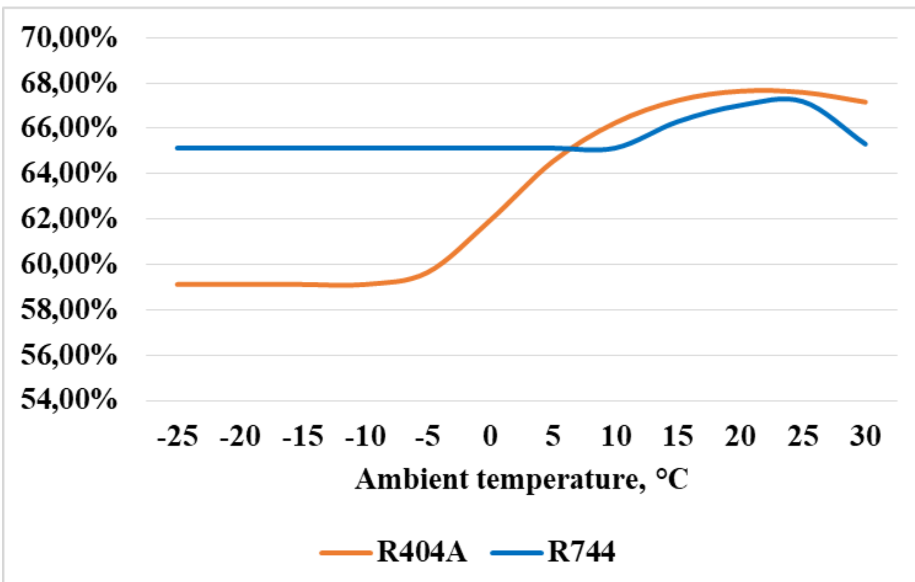

Figure 4: Isentropic efficiency of compressors.

The average parameters for the period under review are shown in Table 2.

Table 2. Efficiency parameters of refrigeration plants

	System 1 (R744)	System 2 (R404A)
COP at adiabatic compression	6,57	7,33
Actual value of COP	4,3	4,19
Degree of thermodynamic efficiency, %	18,82	17,21

Average specific energetic losses at system components for the period under review are shown on Fig. 7.

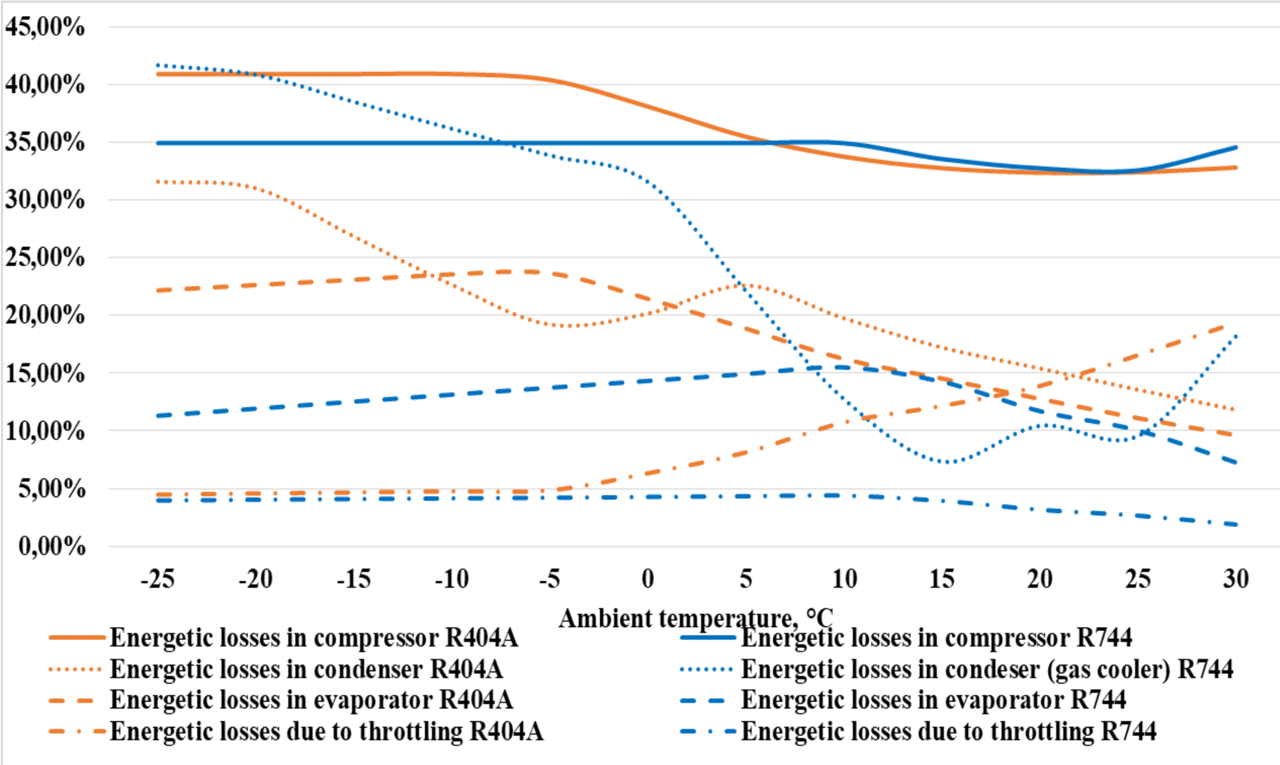

Figure 5: Energetic losses in % of compression work.

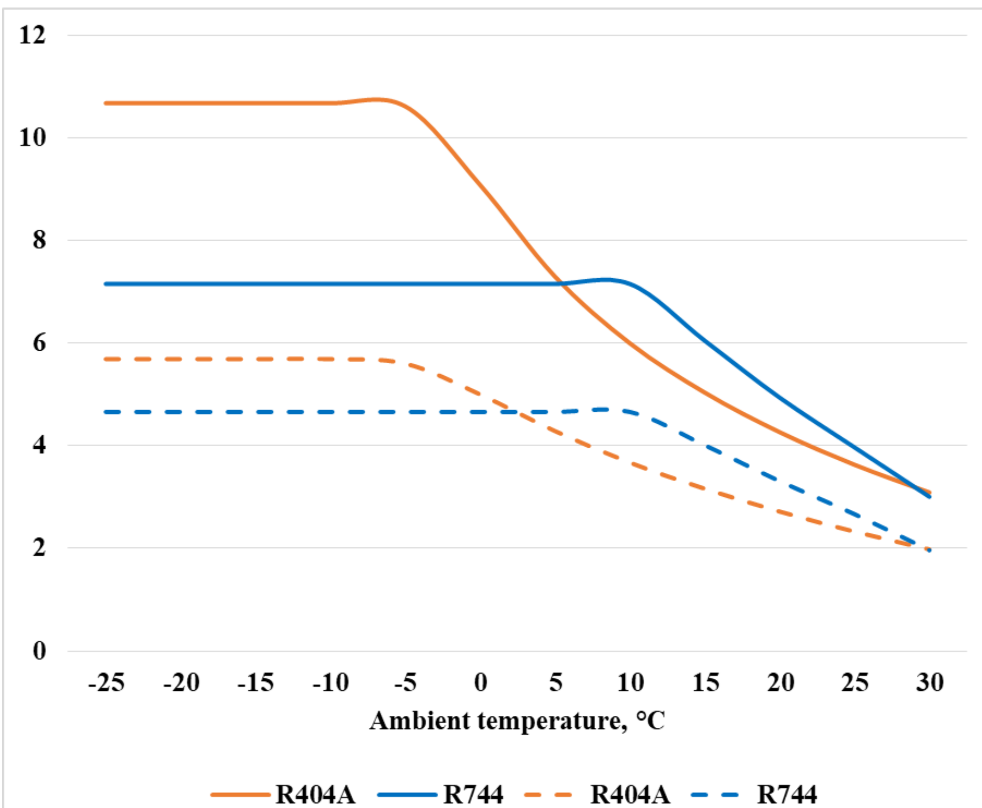

Figure 6: COP at adiabatic compression and actual value (dotted lines).

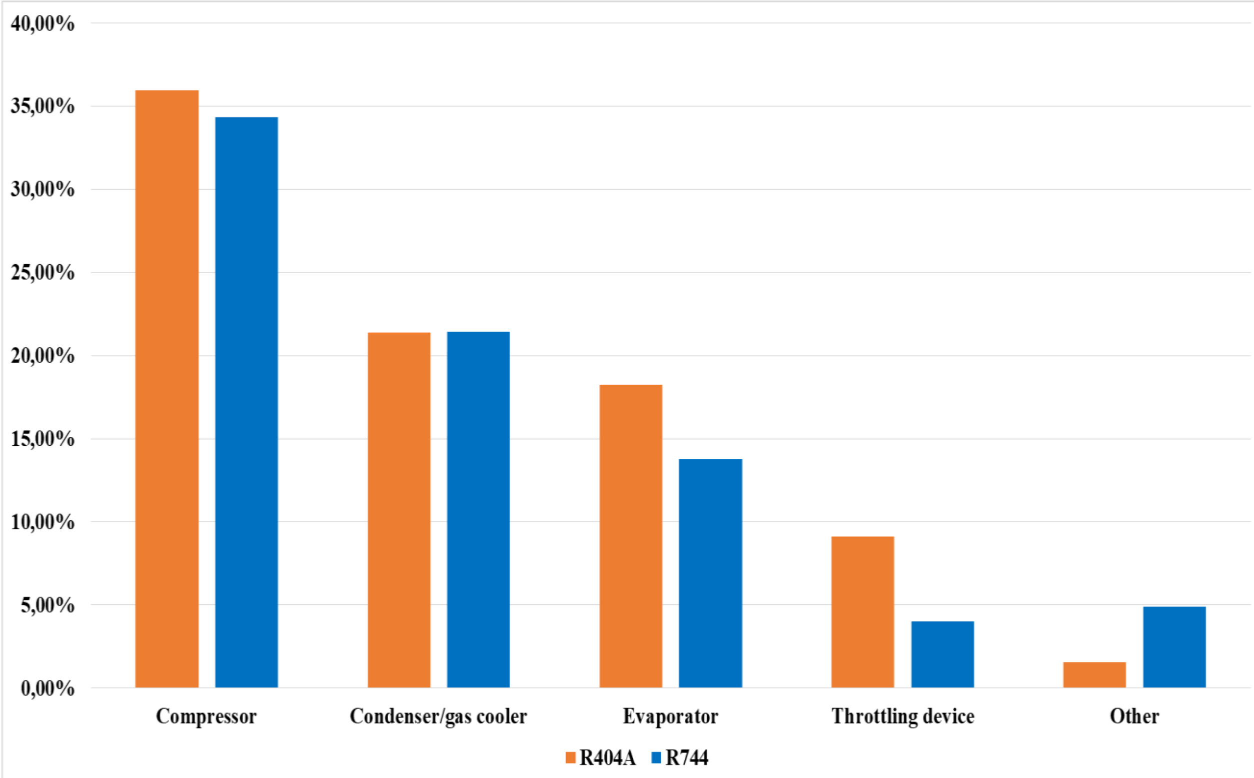

Figure 7: Average specific energetic losses at system components (in % of compression work).

3. CONCLUSIONS

The application of this entropic and statistical method of the analysis allows for cooling plants' energy efficiency to be increased based on information about losses in different refrigeration plant components and for measures to be taken to increase their operational efficiency;

The average actual values of COP for the period under review for System 1 (R744) and System 2 (R404A) are comparable. The difference between these values is no more than 2,61%;

Average meaning of degree of thermodynamic efficiency for transcritical refrigeration plant CO2 more than for refrigeration plant works with R404A at 8,58% for the period under review.

NOMENCLATURE

LT	Low temperature level	T_c	Air temperature in cooling volume (K)
MT	Medium temperature level	t_c	Air temperature in cooling volume (°C)
C	Compressor	T_{env}	Environmental temperature (K)
GC	Gas cooler	t_{env}	Environmental temperature (°C)
Elt	Evaporator LT	q_o	Specific mass cooling capacity (kJ/kg)
Emt	Evaporator MT	l_{min}	Minimum specific work which is necessary for cold generation (kJ/kg)
V	Intermediate vessel	l_s	Adiabatic compression work (kJ/kg)
EV	Expansion valve	l_{comp}	Actual specific compression work (kJ/kg)
Ej	Ejector	Δl_{comp}	Energetic losses in compressor (kJ/kg)
Cond	Condenser	Δl_{evap}	Energetic losses in evaporator (kJ/kg)
Cond-evap	Condenser-evaporator	Δl_{gc}	Energetic losses in gas cooler (kJ/kg)

p	pressure (bar(abs))	Δl_{thr}	Energetic losses in throttling devices (kJ/kg)
ε_s	COP at adiabatic compression	h_{out}	specific enthalpy at outlet, kJ/kg
ε_{act}	Actual value of COP	s_{out}	specific entropy at outlet, kJ/kg
η_{therm}	The degree of thermodynamic efficiency	s_{thr_in}	specific entropy at inlet of throttling device, kJ/kg
s	Specific entropy (kJ/kg/K)	s_{thr_out}	specific entropy at outlet of throttling device, kJ/kg
h	Specific enthalpy (kJ/kg/K)	h_{evap_dew}	specific enthalpy at dew point (evaporation pressure), kJ/kg
T_o	Evaporation temperature (K)	s_{evap_dew}	specific entropy at dew point (evaporation pressure), kJ/kg
t_o	Evaporation temperature (°C)	h_{evap_sh}	specific enthalpy of superheated vapor at evaporation pressure (superheat in evaporator)
T_{cond}	Condensation temperature (K)	s_{evap_sh}	specific entropy of superheated vapor at evaporation pressure (superheat in evaporator)
h_{evap_in}	specific enthalpy at evaporator inlet, kJ/kg	Δl_{other}	Energetic losses in processes which do not mentioned above (superheat, mixing etc.)
s_{dew}	specific entropy at dew point, kJ/kg	s_{bubble}	specific entropy at bubble point, kJ/kg
h_{dew}	specific enthalpy at dew point, kJ/kg	h_{bubble}	specific enthalpy at bubble point, kJ/kg

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