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IMPROVEMENT OF THE THERMODYNAMIC CYCLE OF SINGLE-STAGE STEAM COMPRESSOR REFRIGERATORS

Alexey G. Slinko[1], Artem S. Boychuk[1], Serhii V. Kozlovskyi[1*], Georg K. Lavrenchenko[2]
[1]Odessa National Maritime University, 34, Mechnikova str., Odesa, 65029, Ukraine
[2]LLC «Institute of Low Temperature Energy Technologies», p/b 285, Odesa, 65023, Ukraine
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Abstract
The most common device used by mankind are refrigerators. Their number on the planet Earth is significantly inferior to the number of gadgets, but significantly exceeds the number of no less common cars, the main device of which, most often, are internal combustion engines. They, producing mechanical/electrical energy, pollute the Earth's atmosphere with carcinogenic gases and heat, and refrigeration plants, using this energy, pollute the atmosphere with heat. The efficiency of both the direct (energy) cycle and the reverse (refrigeration) cycle depends on the thermodynamic cycle on which they are built. Therefore, several improvements to the thermodynamic cycle of single-stage refrigerating machines are proposed: from the simplest to the most complex, respectively, from less to more efficient. The performed comparative calculations showed that the excess of the main efficiency indicators of the refrigeration cycle with limited regenerative heat exchange over the indicators of the simplest basic cycle (without surface supercooling of the liquefied refrigerant and superheating of saturated steam) do not exceed 6 %. Exceeding similar indicators of the cycle with maximum (limit) regenerative isobaric steam overheating and polytropic process of its compression over the indicators of the cycle with limited regenerative heat exchange reach tens of percent. At the same time, exceeding the indicators of a similar cycle, but with isochoric regenerative steam overheating, reach 100 %.
Keywords: throttling; compression work; volumetric and energy losses of the compressor; cooling coefficient; thermal pollution of the atmosphere.

УДОСКОНАЛЕННЯ ТЕРМОДИНАМІЧНОГО ЦИКЛУ ОДНОСТУПЕНЕВИХ ПАРОКОМПРЕСОРНИХ ХОЛОДИЛЬНИХ УСТАНОВОК

Олексій Г. Слинько[1], Артем С. Бойчук[1], Сергій В. Козловський[1], Георгій К. Лавренченко[2]
[1]Одеський національний морський університет, вул. Мечникова, 34, Одеса, Україна 65029
[2]ТОВ «Інститут низькотемпературних енерготехнологій», а/с 285, Одеса, Україна, 65023

Анотація
Найпоширенішим пристроєм, яке використовується Людством, є холодильні машини. Їх кількість на планеті Земля суттєво уступає кількості гаджетів, але значно перевищує кількість не менш поширених автомобілів, головним пристроєм яких частіше за все є двигуни внутрішнього згоряння. Вони, виробляючи механічну/електричну енергію, забруднюють атмосферу Землі канцерогенними газами і теплотою, а холодильні установки, використовуючи цю енергію, забруднюють атмосферу теплотою. Ефективність як прямого (енергетичного) циклу, так і зворотного (холодильного) багато в чому залежить від термодинамічного циклу, на якому вони побудовані. Тому пропонується декілька удосконалень термодинамічного циклу одноступеневих холодильних машин: від самих простих до більш складних, відповідно, від менш до більш ефективних. Виконані порівняльні розрахунки показали, що перевищення головних показників ефективності холодильного циклу з обмеженим регенеративним теплообміном над показниками найпростішого базового циклу (без поверхневих переохолодження зрідженого холодоагенту і перегріву насиченої пари) не перевищують 6 %. Перевищення аналогічних показників циклу з максимальним (граничним) регенеративним ізобарним перегрівом пари і політропним процесом її стиснення над показниками циклу з обмеженим регенеративним теплообміном досягають десятки відсотків, а показники аналогічного циклу, але з ізохорним регенеративним перегрівом пари, досягають 100 %.
Ключові слова: дроселювання; робота стиснення; об'ємні і енергетичні втрати компресору; холодильний коефіцієнт; теплове забруднення атмосфери.

**Corresponding author: e-mail: nemio@ukr.net*

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Introduction

A large pile of reliable knowledge on any issue becomes science with a capital letter only when the main idea is crystallized from this pile in the form of the so-called "golden" rule. In thermodynamics, this is "all processes are irreversible" [1], and in its subdivisions: energy (on the conversion of heat into work) it is "...to increase the efficiency of the process of converting heat into work, cool the working body before compression, and heat it before expansion " [2; 3], and in the refrigerating cycle (after obtaining cold due to mechanical/electrical energy) it is - "...to increase the efficiency of the process of obtaining cold, heat the working body regeneratively before compression, and cool it before expansion" [4].

Refrigeration units are now the most common appliances used by mankind. Home refrigerators, devices for comfortable, technical and technological air conditioning, large city refrigerators, factories, refrigerated ships and gas carriers, this is far from a complete list of industries that widely use cold, which is obtained artificially, as a rule, with the help of steam compressor refrigeration units [5–9]. A significant majority of such refrigerating units work on high-tech energy: mechanical/electric, during the production of which heat and gas pollution of the Earth's atmosphere takes place, which is now an actual problem. In addition, the refrigeration units themselves are significant thermal pollutants of the atmosphere. Therefore, improving the thermodynamic cycle of steam-compressor refrigerating units is now an urgent task of modern times and will always remain so [10–14].

The article shows possible ways of its improvement using the example of the thermodynamic cycle of a single-stage refrigerating unit. For this, the most important thermal energy characteristics of three improved types of thermodynamic cycle of single-stage refrigerating units were calculated and compared. Freon R125, which is used in the temperature range - condensation t_{cond} = 40 ℃ and evaporation t_{evap} = -20 ℃, is used as the working fluid of the compared cycles [15–19].

Results and discussion

Thermodynamic cycle of a basic single-stage refrigerating machine

In fig. 1 the *T,s*-diagram shows the thermodynamic cycle of a simple basic refrigeration unit without subcooling of the liquefied refrigerant and superheating of the steam. The use of such a simple cycle causes large throttling losses (process 4-5), which reduce the specific mass cooling capacity, and the compression of saturated steam (process 1-2) threatens with a hydraulic shock in the compressor. Taken together, the indicated shortcomings determine the low value of the cooling coefficient of such a cycle.

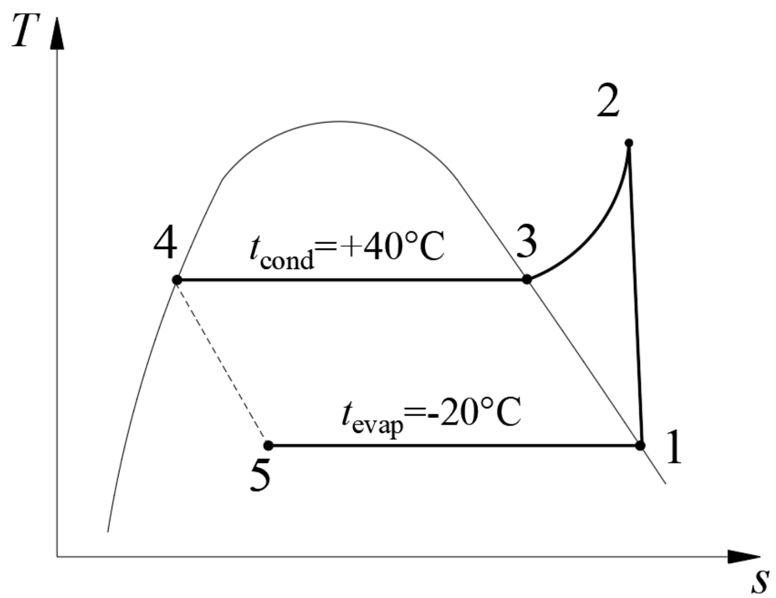

Fig. 1. The thermodynamic cycle of the refrigeration unit without any subcooling of the liquefied refrigerant and overheating of the steam

Using the REFPROP program [20], the thermodynamic properties of refrigerant R125 were calculated at all characteristic points of the basic cycle. The determined thermodynamic properties are given in table 1.

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Table 1

Thermodynamic properties of the refrigerant (R125) at characteristic points of a simple basic refrigeration cycle

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	-20	3.3733	21,331	0.046880	307.22	323.03	1.4906
2	43.336	20.085	136.06	0.0073501	336.39	351.15	1.4906
3	40	20.085	142.52	0.00701640	332.60	346.69	1.4764
4	40	20.085	1088.4	0.00091878	252.82	254.67	1.1826
5	-20	3.3733	39.264	0.025469	246.08	254.67	1.2205

Using tabular data, calculated:

– specific mass cooling capacity

$$q_0 = h_1 - h_5 = 323.03 - 254.67 = 68.36 \text{ kJ/kg};$$

– specific work of compression

$$l = h_2 - h_1 = 351.15 - 323.03 = 28.12 \text{ kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon = \frac{q_0}{l} = \frac{68.36}{28.12} = 2.431.$$

Conditionally accepting the cooling capacity of the installation $Q_0 = 200$ kW, calculated:

– consumption of refrigerant R125, which circulates in the system

$$M_{\text{R125}} = \frac{Q}{q_0} = \frac{200}{68.36} = 2.9257 \text{ kg/sec};$$

– the theoretical power of the installation

$$N_{\text{T}} = M_{\text{R125}} \cdot l = 2.9257 \cdot 28.12 = 82.27 \text{ kW};$$

– the heating factor of the refrigerant during its compression

$$\lambda_{\text{W}} = \frac{T_0}{T_{\text{k}} + 40} = \frac{253.15}{313.15 + 40} = 0.7168;$$

– indicator efficiency of the theoretical cycle

$$\eta_{\text{i}} = \lambda_{\text{W}} + bt_0 = 0.7168 + 0.0025 \cdot (-20) = 0.6668;$$

– indicator power of the compressor

$$N_{\text{i}} = \frac{N_{\text{T}}}{\eta_{\text{i}}} = \frac{82.27}{0.6668} = 123.38 \text{ kW},$$

– feed rate

$$\lambda_{\text{V}} = \lambda_{\text{C}} \cdot \lambda_{\text{throttling}} \cdot \lambda_{\text{W}} \cdot \lambda_{\text{d}} = 0.8514 \cdot 0.98 \cdot 0.7168 \cdot 0.98 = 0.5861,$$

where the coefficient of influence of the harmful volume of the compressor cylinder on the supply of steam to the compressor

$$\lambda_{\text{C}} = 1 - \text{c}\left(\frac{p_{\text{boil}}}{p_{\text{evap}}} - 1\right) = 1 - 0{,}03\left(\frac{20.085}{3.3733} - 1\right) = 0.8514;$$

– the actual volume of steam sucked in and compressed by the compressor

$$V_{\text{act}} = \frac{M_{\text{R125}}}{\rho_1} = \frac{2.9257}{21.331} = 0.1372 \text{ m}^3/\text{sec};$$

– hourly volume of the compressor

$$V_{\text{h}} = \frac{V_{\text{act}}}{\lambda_{\text{V}}} = \frac{0.1372}{0.5861} = 0.2341 \text{ m}^3/\text{sec};$$

– the power spent on friction in the kinematic pairs of the compressor

$$N_{\text{friction}} = V_{\text{h}} \cdot p_{\text{friction}} = 0.2341 \cdot 50 = 11.705 \text{ kW};$$

– the actual (effective) capacity of the compressor

$$N_{\text{e}} = N_{\text{i}} + N_{\text{friction}} = 123.38 + 11.705 = 135.09 \text{ kW};$$

– the effective (effective) cooling coefficient

$$\varepsilon_{\text{E}} = \frac{Q_0}{N_{\text{e}}} = \frac{200}{135.09} = 1.4805;$$

– heat load of the condenser (heat pollution of the atmosphere)

$$Q_{\text{k}} = Q_0 + N_{\text{i}} = 200 + 135.09 = 335.09 \text{ kW}.$$

Thermodynamic cycle of a single-stage refrigerating machine with limited regenerative heat exchange

Such a thermodynamic cycle of a single-stage refrigerating machine is shown in fig. 2 and proposed a long time ago to increase the specific cooling capacity, prevent hydraulic shock in the compressor and prevent the formation of steam during the transportation of liquefied refrigerant on the way from the condenser to the temperature control valve. In such a cycle, not only the specific mass cooling capacity (h_7-h_6) increases, but also the specific work of compression (h_2-h_1). The relative ratio of these characteristics depends on the type of refrigerant used and the temperature range of the cycle. Looking ahead, in the considered cycle, limited regenerative heat exchange reduces the efficiency of the refrigeration cycle. When constructing such a cycle, the superheat temperature of the steam is usually set in the range of 5...10 °C. In the case under study, 10 °C was taken. Then, when the thermodynamic properties of R125 are determined at point 1, it is accepted $t_1 = -10$ °C, $p_1 = 3.3733$ bar.

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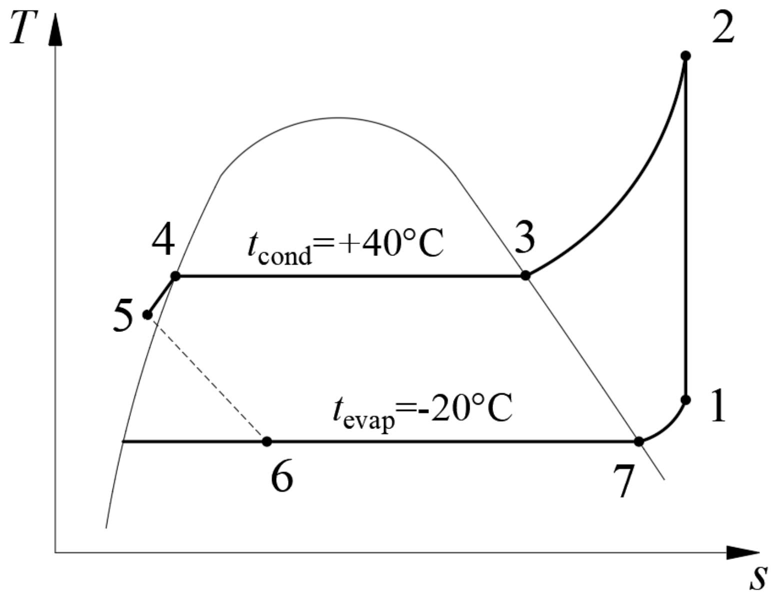

Fig. 2. Thermodynamic cycle of a single-stage refrigerating unit with limited regenerative heat exchange

The enthalpy of the regeneratively supercooled liquid refrigerant (point 5) was determined from the heat balance of the regenerative heat exchanger.

Using the REFPROP program [20], the thermodynamic properties of refrigerant R125 were calculated at all characteristic points of this cycle. The determined thermodynamic properties are given in table 2.

Table 2

Thermodynamic properties of the working medium (R125) at characteristic points of the thermodynamic cycle of a single-stage refrigerating unit with limited regenerative heat exchange

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	-10	3.3733	20.207	0.0494870	314.26	330.95	1.5213
2	51.469	20.085	124.41	0.0080380	344.84	360.99	1.5213
3	40	20.085	142.52	0.00701640	332.60	346.69	1.4764
4	40	20.085	1088.4	0.00091878	252.82	254.67	1.1826
5	34.886	20.085	1131.1	0.00088412	244.97	246.75	1.1571
6	-20	3.3733	43.50	0.022988	239.00	246.75	1.1892
7	-20	3.3733	21.331	0.046880	307.22	323.03	1.4906

Using tabular data, calculated:

- superheat of steam

$\Delta h = h_1 - h_7 = 330.95 - 323.03 = 7.92\,\text{kJ/kg}$;

– enthalpy of liquefied supercooled refrigerant (point 5)

$h_5 = h_4 - \Delta h = 254.67 - 7.92 = 246.75\ \text{kJ/kg}$;

– specific mass cooling capacity

$q_0 = h_7 - h_5 = 323.03 - 246.75 = 76.28\ \text{kJ/kg}$;

– specific work of steam compression

$l = h_2 - h_1 = 360.99 - 330.95 = 30.04\ \text{kJ/kg}$;

– theoretical cooling coefficient

$$\varepsilon = \frac{q_0}{l} = \frac{76.28}{30.04} = 2.5393.$$

When acceptedabove the cooling capacity of the installation is calculated:

– the consumption of refrigerant R125, which circulates in the system

$$M_{\text{R125}} = \frac{Q_0}{q_0} = \frac{200}{76.28} = 2.6219\ \text{kg/sec},$$

where $Q_0 = 200$ kW;

– the theoretical capacity of the refrigerating machine

$N_{\text{T}} = M_{\text{R125}} \cdot l = 2.6219 \cdot 30.04 = 78.76\,\text{kW}$;

– the indicator power of the machine (provided that the indicator efficiency remained the same as in the previous version of the cycle)

$$N_{\text{i}} = \frac{N_{\text{T}}}{\eta_{\text{i}}} = \frac{78.76}{0.6668} = 118.12\,\text{kW};$$

– the actual hourly volume of steam that must be pumped through the system by the compressor

$$V_{\text{act}} = \frac{M_{\text{R125}}}{\rho_1} = \frac{2.6219}{20.207} = 0.1298\ \text{m}^3/\text{sec};$$

– hourly volume of the compressor (provided that the compressor supply ratio remains the same as in the previous version of the cycle)

$$V_{\text{h}} = \frac{V_{\text{act}}}{\lambda_{\text{V}}} = \frac{0.1298}{0.5861} = 0.2214\ \text{m}^3/\text{sec};$$

– the power spent by the compressor on internal friction

$N_{\text{friction}} = V_{\text{h}} \cdot p_{\text{fiction}} = 0.2214 \cdot 50 = 11.07\,\text{kW}$;

– the actual (effective) capacity of the compressor

$$N_\mathrm{e} = N_\mathrm{i} + N_\mathrm{friction} = 118.12 + 11.07 = 129.19 \text{ kW};$$

– the effective (effective) cooling coefficient

$$\varepsilon_\mathrm{E} = \frac{Q_0}{N_\mathrm{e}} = \frac{200}{129.19} = 1.5481;$$

– heat load of the condenser (heat pollution of the atmosphere)

$$Q_\mathrm{k} = Q_0 + N_\mathrm{i} = 200 + 129.19 = 329.19 \text{ kW}.$$

Termodynamic cycle of a single-stage refrigerating machine with the limit regenerative isobaric superheating of steam and the polytropic process of its compression

The thermodynamic cycle of such a refrigerating machine is shown in Fig. 3.

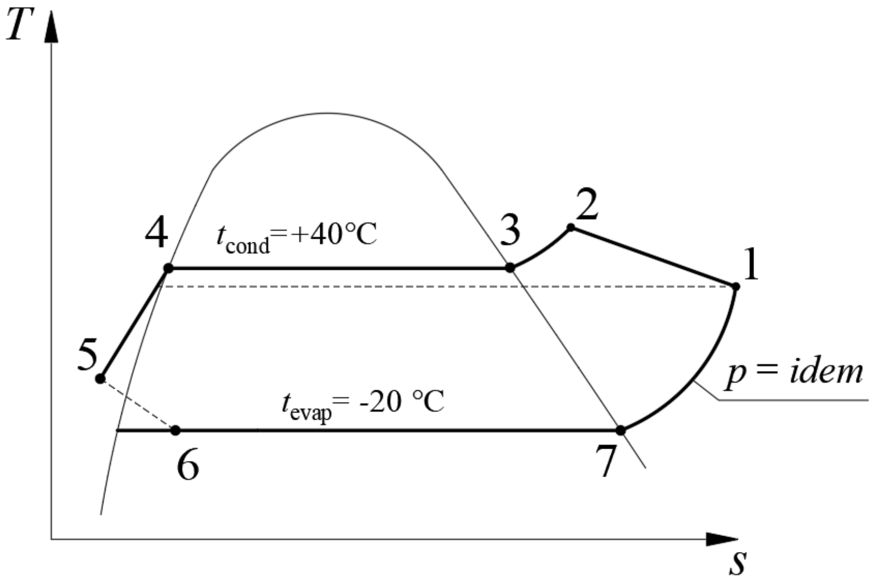

Fig. 3. Thermodynamic cycle of a single-stage refrigerating unit with a limiting isobaric regenerative overheating of steam and the polytropic process of its compression

The limiting regenerative (maximum possible) steam superheat is represented by process 7-1. Overheating of the steam is carried out by liquefied saturated refrigerant, which ensures, firstly, its overheating to the temperature of the surrounding medium (dashed horizontal line), secondly, supercooling of the liquefied refrigerant (process 4-6). The latter increases the specific mass cooling capacity. To reduce the value of the specific work of steam compression, the adiabatic compression process was replaced by the polytropic process 1-2. To prevent hydraulic shock in the compressor, the temperature of the steam at the end of compression (point 2) is taken to be 10 °C higher than the condensation temperature. The compressor is cooled by the same coolant that cools the condenser. The determined thermodynamic properties of the refrigerant R125 at the characteristic points of the studied cycle are given in the table 3.

Table 3

Thermodynamic properties of refrigerant R125 at characteristic points of the thermodynamic cyclesingle-stage refrigerating unit with extreme isobaric regenerative superheating of steam and polytropic compression process

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	30	3.3733	16.895	0.059188	343.34	363.30	1.6357
2	50	20.085	163.23	0.0079222	343.36	359.27	1.5160
3	40	20.085	142.52	0.00701640	332.60	346.69	1.4764
4	40	20.085	1088.4	0.00091878	252.82	254.67	1.1826
5	11.138	20.085	1279.3	0.00078354	212.53	214.10	1.0469
6	-20	3.3733	43.149	0.023176	239.53	247.35	1.1916
7	-20	3.3733	21.331	0.046880	307.22	323.03	1.4906

Using tabular data, calculated:

– specific work of the polytropic steam compression process (process 1-2)

$$l_\mathrm{k} = T_\mathrm{aver}(s_2 - s_1) - (h_2 - h_1) = 313.15(1.5160 - 1.6357) - (359.27 - 363.30) = -33.45 \text{ kJ/kg};$$

– heat of marginal regenerative overheating of steam (process 7-1)

$$\Delta q_\mathrm{superheat} = h_1 - h_7 = 363.30 - 323.03 = 40.27 \text{ kJ/kg};$$

– the enthalpy of the liquefied supercooled refrigerant at point 5

$h_5 = h_4 - \Delta h = 254.67 - 40.57 = 214.10$ kJ/kg;

– specific mass cooling capacity

$$q_0 = h_7 - h_5 = 323.03 - 214.10 = 108.93 \text{ kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon = \frac{q_0}{|l_k|} = \frac{108.93}{33.45} = 3.2565.$$

With the same cooling capacity of the installation, the following is calculated:

$$M_{R125} = \frac{Q_0}{q_0} = \frac{200}{108.93} = 1.8360 \text{ kg/sec};$$

– theoretical power of the compressor

$$N_T = M_{R125} \cdot l_k = 1.8360 \cdot 33.45 = 61.41 \text{ kW};$$

– indicator power of the compressor

$$N_i = \frac{N_T}{\eta_i} = \frac{61.41}{0.95} = 64.64 \text{ kW};$$

where the indicator efficiency of the compressor is equal to

$$\eta_i = \lambda_w + b \cdot t = 1 + 0.0025 \cdot (-20) = 0.95;$$

where the heating factor $\lambda_w = 1$ (there is no heating of the refrigerant vapor in the compressor cylinders)

– feed rate

$$\lambda_V = \lambda_c \cdot \lambda_{throttling} \cdot \lambda_w \cdot \lambda_d = 0.8514 \cdot 0.98 \cdot 1.0 \cdot 0.98 = 0.8177;$$

– the actual hourly volume of steam that must be pumped through the system by the compressor

$$V_{act} = \frac{M_{R125}}{\rho_1} = \frac{1.8360}{16.895} = 0.1087 \text{ m}^3/\text{sec};$$

– hourly volume of the compressor

$$V_h = \frac{V_{act}}{\lambda_V} = \frac{0.1087}{0.8177} = 0.1329;$$

– the power spent by the compressor on its internal friction

$$N_{friction} = V_h \cdot p_{friction} = 0.1329 \cdot 50 = 6.65 \text{ kW};$$

– the actual (effective) capacity of the compressor

$$N_e = N_i + N_{friction} = 64.64 + 6.65 = 71.29 \text{ kW};$$

– the effective (effective) cooling coefficient

$$\varepsilon_E = \frac{Q_0}{N_e} = \frac{200}{71.29} = 2.810;$$

– heat load of the condenser (heat pollution of the atmosphere)

$$Q_k = Q_0 + N_i = 200 + 64.64 = 264.64 \text{ kW}.$$

Termodynamic cycle of a single-stage refrigerating machine with limiting regenerative isochoric superheating of steam and a polytropic compression process

The thermodynamic cycle of such a refrigerating machine is shown in Fig. 4. Limit regenerative (maximum possible) superheating of the steam is carried out according to the isochoric process (curve 7–1). From the previous (discussed above) thermodynamic cycle of a single-stage refrigerating unit, superheating of steam is carried out according to the isochoric process 1–7, which significantly reduces the amount of heat spent on its superheating, which is bad from the point of view of regenerative subcooling of the liquefied refrigerant, but very effective from the point of view of operation its further compression by a compressor. This thermodynamic cycle does not differ in any other way from the one considered in the previous section of the article.

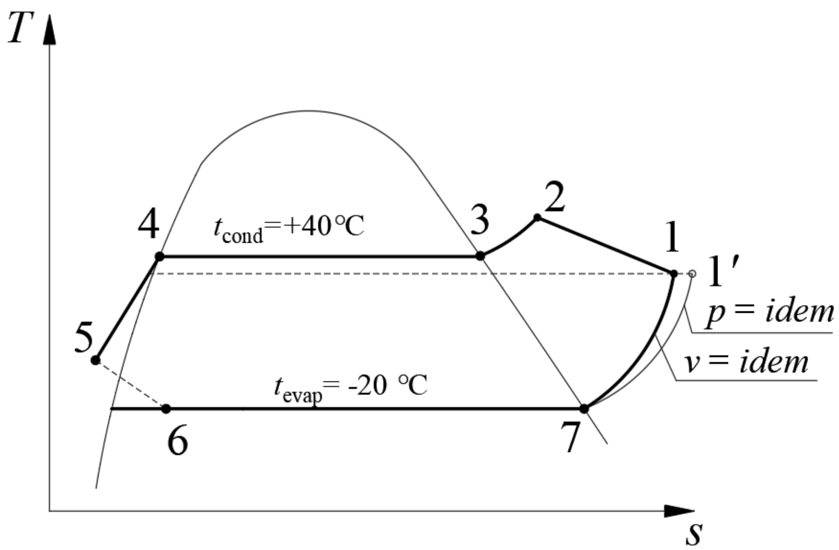

Fig. 4. Thermodynamic cycle of a single-stage refrigerating unit with limiting isochoric regenerative superheating of steam and a polytropic compression process

The determined thermodynamic properties of the refrigerant R125 at the characteristic points of this cycle are given in the table 4.

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Table 4

Thermodynamic properties of refrigerant R125 at characteristic pointsof the thermodynamic cycle of a single-stage refrigerating unit with a limiting isochoric regenerative superheating of steam and a polytropic compression process

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	30	4.2017	21.331	0.046880	342.65	362.35	1.6181
2	50	20.085	163.23	0.0079222	343.36	359.27	1.5160
3	40	20.085	142.52	0.00701640	332.60	346.69	1.4764
4	40	20.085	1088.4	0.00091878	252.82	254.67	1.1826
5	15.093	20.085	1255.6	0.00079641	217.64	219.24	1.0648
6	-20	3.3733	43.149	0.023176	239.53	247.35	1.1916
7	-20	3.3733	21.331	0.046880	307.22	323.03	1.4906

Using tabular data, calculated:

– heat consumption for isochoric regenerative superheating of steam

$$q_{\text{superheat}} = u_1 - u_7 = 342.65 - 307.22 = 35.43 \ \text{kJ/kg};$$

– enthalpy of liquefied supercooled refrigerant (point 5)

$$h_5 = h_4 - q_{\text{superheat}} = 254.67 - 35.43 = 219.24 \ \text{kJ/kg};$$

– specific mass cooling capacity

$$q_0 = h_7 - h_5 = 323.03 - 219.24 = 103.79 \ \text{kJ/kg};$$

– specific work of the polytropic process of steam compression by a compressor

$$l_{\text{k}} = T_{\text{aver}}(s_2 - s_1) - (h_2 - h_1) = 313.15(1.5160 - 1.6181) - (359.27 - 362.35) = -28.89 \ \text{kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon = \frac{q_0}{|l_{\text{k}}|} = \frac{103.79}{28.89} = 3.5971.$$

With the same cooling capacity of the installation, the following is calculated:

– consumption of refrigerant circulating in the system

$$M_{\text{R125}} = \frac{Q_0}{q_0} = \frac{200}{103.79} = 1.9270 \ \text{kg/sec};$$

– theoretical power of the compressor

$$N_{\text{T}} = M_{\text{R125}} \cdot l_{\text{k}} = 1.9270 \cdot 28.89 = 55.67 \ \text{kW};$$

– indicator power of the compressor

$$N_{\text{i}} = \frac{N_{\text{T}}}{\eta_{\text{i}}} = \frac{55.67}{0.95} = 58.60 \ \text{kW};$$

where the indicator efficiency of the compressor is equal to

$$\eta_{\text{i}} = \lambda_{\text{w}} + \text{b} \cdot \text{t} = 1 + 0.0025 \cdot (-20) = 0.95;$$

where the heating factor $\lambda_{\text{w}} = 1$ (there is no heating of the refrigerant vapor in the compressor cylinder)

– feed rate

$$\lambda_{\text{V}} = \lambda_{\text{c}} \cdot \lambda_{\text{throttling}} \cdot \lambda_{\text{w}} \cdot \lambda_{\text{d}} = 0.8514 \cdot 0.98 \cdot 1.0 \cdot 0.98 = 0.8177;$$

– the actual hourly volume of steam that must be pumped through the system by the compressor

$$V_{\text{act}} = \frac{M_{\text{R125}}}{\rho_1} = \frac{1.9270}{21.331} = 0.09034 \ \text{m}^3/\text{sec};$$

– hourly volume of the compressor (provided that the so-called compressor supply ratio remains the same as in the previous version of the cycle)

$$V_{\text{h}} = \frac{V_{\text{act}}}{\lambda_{\text{V}}} = \frac{0.09034}{0.7768} = 0.1163 \ \text{m}^3/\text{sec};$$

– the power spent by the compressor on internal friction

$$N_{\text{friction}} = V_{\text{h}} \cdot p_{\text{friction}} = 0.1163 \cdot 50 = 5.815 \ \text{kW};$$

– the actual (effective) capacity of the compressor

$$N_{\text{e}} = N_{\text{i}} + N_{\text{friction}} = 58.60 + 5.815 = 64.415 \ \text{kW};$$

– the effective (actual) cooling coefficient

$$\varepsilon_{\text{E}} = \frac{Q_0}{N_{\text{e}}} = \frac{200}{64.415} = 3.1049;$$

– heat load of the condenser (heat pollution of the atmosphere)

$$Q_{\text{k}} = Q_0 + N_{\text{i}} = 200 + 58.60 = 258.60 \ \text{kW}.$$

For the possibility of a comparative analysis of the efficiency of the thermodynamic cycles of a single-stage refrigerating machine investigated in the work, we summarize them in a general table 5, the most important indicators of their effectiveness are calculated.

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Table 5

The most important characteristics of the efficiency of the thermodynamic cycles of refrigeration units investigated in the work

No.	Performance indicator	Thermodynamic cycle			
		base	with limited isobaric superheating of steam	with the maximum isobaric superheat of the steam	with maximum isochoric steam superheat
1	2	3	4	5	6
1.	Cooling capacity of the installation, kW	200			
2.	Specific mass cooling capacity, kJ/kg	68.36	76,28	108.93	103.79
3.	Specific work of vapor compression, kJ/kg	28.12	30.04	33.45	28.89
4.	Theoretical power of the installation, kW	82.27	78.76	61.41	55.67
5.	Theoretical cooling coefficient	2.431	2.5393	3.2565	3.5971
6.	Indicator power of the installation, kW	123.38	118.12	64.64	58.60
7.	Supply ratio	0.5861	0.5861	0.8177	0.8177
8.	Compressor hourly volume, m^3/s	0.2341	0.2214	0.1329	0.1163
9.	Effective (actual) power of the installation, kW	135.09	129.19	71.29	64.42
10.	Effective refrigeration coefficient	1.4805	1.5481	2,810	3.1049
11.	Thermal load of the condenser (thermal pollution of the atmosphere)	335.09	329.19	264.64	258.60
12.	Relative decrease in the hourly volume of the compressor, %		5.7*	66.6**	90.4**
13.	Relative reduction of the effective power of the installation, %		4.6	81.2	in 2 times
14.	Relative increase in the effective cooling coefficient,%		4.6	81.5	in 2 times
15.	Relative reduction of the heat load of the condenser (heat pollution of the atmosphere),%		1.2	24.4	27.3

* relative to the basic cycle: without subcooling of the liquefied refrigerant and overheating of the steam

**relative to a cycle with limited regenerative isobaric heat exchange between liquid refrigerant and steam

Conclusions

Analyzing the tabular data, it is possible to unequivocally state:

– the most effective improved thermodynamic cycle of single-stage refrigerating machines is a cycle with the maximum (maximum possible) isochoric regenerative overheating of steam and the polytropic process of its compression;

– the reduction of the effective compressor capacity and the increase of the effective cooling coefficient more than twice exceed the corresponding indicators of the cycle with limited regenerative heat exchange, which means that the amount of electricity consumed or fuel for its production decreases accordingly;

– the decrease in the hourly volume of the compressor in the cycle with isochoric regenerative steam overheating relative to the corresponding indicator of the cycle with limited regenerative heat exchange is 90.4 %;

– a similar cycle with isobaric regenerative superheating of steam is less effective in terms of this efficiency indicator, but structurally simpler: the decrease in the hourly volume of the compressor of such a cycle, relative to the corresponding indicator of the base cycle, is 66.6 %, which means that the compressors mass-size indicators of the refrigerating units implemented according to the corresponding thermodynamic cycles will be smaller;

– the decrease in the heat load of the condensers of the two latest modifications of the thermodynamic cycle of the single-stage refrigerating unit shows how much their mass and size indicators decrease, and, most importantly, it shows how much the thermal pollution of the atmosphere decreases.

Therefore, if one-stage refrigerating units are manufactured according to the proposed improvement of thermodynamic cycles with isochoric or isobaric processes of the maximum possible regenerative overheating of steam and polytropic process of its compression, it is possible to significantly reduce the amount of electricity consumed by compressors, which means that thermal pollution of the atmosphere by power plants will decrease. In addition, the refrigeration units themselves will pollute the atmosphere less, which is extremely relevant now in solving the issue of increasing the temperature of the Earth's atmosphere.

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