Journal of Chemistry and Technologies, 2025, *33*(2), 548-557

Journal of Chemistry and Technologies

pISSN 2663-2934 (Print), ISSN 2663-2942 (Online).

journal homepage: http://chemistry.dnu.dp.ua
editorial e-mail: chem.dnu@gmail.com

UDC 536.7

IMPROVED SINGLE-STAGE REFRIGERATION UNITS TO REPLACE TWO-STAGE UNITS

Alexey G. Slynko[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

Received 24 March 2025; accepted 28 April 2025; available online 15 July 2025

Abstract

The ratio between the condensation pressure and the evaporation pressure of the working fluid in vapor-compressor refrigeration machines increases, the losses during throttling of the liquefied refrigerant and the work of adiabatic vapor compression increase. When this ratio $\pi = \rho_{cond} / \rho_{evap}$ reaches a value of 8, to reduce the specified losses, they switch to a cycle with two-stage compression and intermediate vapor cooling and with double throttling of the liquefied refrigerant. However, such an improvement of vapor-compressor units complicates their operation. Therefore, in practice, single-stage refrigeration units are sometimes used even when π slightly exceeds the specified π value, and this leads to significant losses in the efficiency of the refrigeration cycle. The paper proposes to use improved single-stage refrigeration units as an alternative to two-stage refrigeration units. To verify the feasibility and effectiveness of this idea, a comparison of the efficiency indicators of two possible modifications of single-stage refrigeration units was performed to replace a two-stage one with a compression ratio π equal to 9. The calculations showed that the most effective substitute for two-stage refrigeration units is an improved single-stage unit with isochoric limit (maximum possible) regenerative superheating of steam and a polytropic process of its compression. Less effective, but more structurally simple, is a plant with isobaric superheating of steam. The reduction in the effective power of the compressors in these units relative to the indicator of a two-stage unit with limited regenerative heat exchange is 15 and 5 %, respectively, which means that the costs of electricity and fuel used for its generation are reduced. In addition, the thermal loads on the condensers of such plants are reduced, which means their weight and dimensions are reduced, and, most importantly, thermal pollution of the atmosphere is reduced (by 3 and 1 %, respectively).

Keywords: throttling; compression work; compressor volume and energy losses; refrigeration

УДОСКОНАЛЕНІ ОДНОСТУПЕНЕВІ ХОЛОДИЛЬНІ УСТАНОВКИ ДЛЯ ЗАМІНИ ДВОСТУПЕНЕВИХ

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, 65029, Україна

[2]ТОВ «Інститут низькотемпературних енерготехнологій», а/с 285, Одеса, 65023, Україна

Анотація

Збільшення співвідношення між тиском конденсації і випаровування робочого тіла в парокомпресорних холодильних машинах підвищує втрати під час дроселювання зрідженого холодоагенту і роботу адіабатного стиснення пари. Коли це співвідношення $\pi = \rho_{конд}/\rho_{вип}$ досягає 8, для зменшення вказаних втрат переходять до циклу з двоступеневим стисненням і проміжним охолодженням пари та з двократним дроселюванням зрідженого холодоагенту. Таке вдосконалення парокомпресорних установок ускладнює їх експлуатацію. Тому на практиці іноді використовують одноступеневі холодильні установки навіть тоді, коли π дещо перевищує 8, а це призводить до значних втрат ефективності холодильного циклу. В роботі пропонується використовувати удосконалені одноступеневі холодильні установки як альтернативу двоступеневим. Для перевірки дієздатності й ефективності цієї ідеї зіставлені показники ефективності двох можливих модифікацій одноступеневих холодильних установок для заміни двоступеневої зі ступенем стиснення π = 9. Виконані розрахунки показали, що найбільш ефективним замінником двоступеневих холодильних установок є удосконалена одноступенева установка з ізохорним межовим (максимально можливим) регенеративним перегрівом пари та політропним процесом її стиснення. Менш ефективною, але більш конструктивно простою є установка з ізобарним перегрівом пари. Зменшення ефективної потужності компресорів у цих установках відносно такого показника двоступеневої установки із обмеженим регенеративним теплообміном складає 15 та 5 %, відповідно, що означає зменшення витрат електроенергії і палива на її вироблення. Крім того, зменшуються теплові навантаження на конденсатори таких установок, а значить, зменшуються їх масогабаритні показники, а головне, зменшується теплове забруднення атмосфери (на 3 та 1 %, відповідно).

Ключові слова: дроселювання; робота стиснення; об'ємні і енергетичні втрати компресору; холодильний коефіцієнт; теплове забруднення атмосфери.

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

 doi: 10.15421/jchemtech.v33i2.325344

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Introduction

Two-stage refrigeration units with full intercooling and double throttling are used in cases where the ratio between the condensation and evaporation pressure of steam ($\pi = \rho_{cond}/\rho_{evap}$) reaches a value of 8. The use of such units, instead of single-stage ones, is introduced to reduce the work of vapor compression and increase the specific mass refrigeration capacity, and hence the refrigeration coefficient of the cycle [1–3]. However, such improvements to two-stage units cause a number of disadvantages associated, primarily, with their operation. Therefore, in practice, to simplify operation, single-stage vapor compressor units are sometimes used even with a ratio of $\pi > 8$. The paper investigates two types of improved single-stage refrigeration units, which are proposed as alternatives to replace two-stage ones.

For this purpose, the most important thermal energy characteristics of such single-stage thermodynamic cycles of refrigeration units, which are proposed to replace similar two-stage refrigeration units, were calculated and compared.

The performed comparative calculations showed that the main efficiency indicators of a classic single-stage installation with limited regenerative heat exchange (mentioned above, as sometimes used in practice to replace a two-stage one) are $\pi > 8$) are 7...56 % worse than the corresponding indicators of a similar in π, but two-stage cycle (basic): (without any subcooling of the liquefied refrigerant and superheating of saturated vapor). At the same time, the excess of such indicators of improved single-stage installations with isobaric maximum (limit) regenerative superheating of vapor and polytropic compression process over the indicators of the basic two-stage installation is 15...1 %, and the excess of the indicators of a similar single-stage installation, but with isochoric regenerative superheating of vapor is 18...3 %.

Calculations and discussions

Calculations of equalized cycles were performed for the same working fluid – R134a and with the same input data: refrigerant evaporation temperature –*minus* 20 °C, the total vapor compression ratio $\pi = 9$, and cooling capacity of the units – $Q_0 = 200$ kW.

Thermodynamic cycle of a basic two-stage refrigeration machine

Fig. 1 shows the T,s-diagram of the thermodynamic cycle of a basic two-stage refrigeration machine with two-stage isentropic vapor compression, complete intercooling, and double throttling. This improvement of single-stage refrigeration units, which was proposed long ago (at the beginning of the development of thermodynamic cycles of refrigeration machines [4–10]), reduces the throttling losses during throttling of the saturated liquefied refrigerant (process 8–9), the work of vapor compression, and its maximum temperature in the cycle.

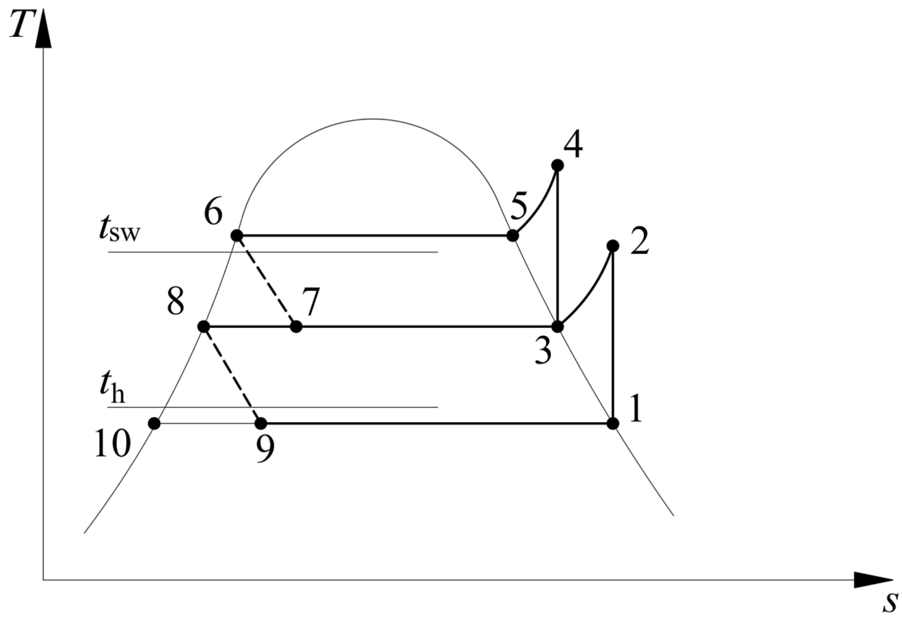

Fig. 1. Thermodynamic cycle of a basic two-stage refrigeration unit with full intercooling and double throttling

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In total, all this leads to an increase in the thermodynamic efficiency of such a cycle, but complicates the operation of such two-stage plants. Since this cycle is the basic one in our study, with which the improvements proposed in the work will be compared, its efficiency indicators were calculated. For this, using the REFPROP program [11], the thermodynamic properties were first determined R134a at points 1, 2, 3, 4, 6, 8 and 9. Using the REFPROP program [11], the thermodynamic properties of the working fluid at the above points were calculated and the results are summarized in Table 1.

Table 1

Thermodynamic properties of R134a at points required to determine the efficiency of the 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	1.3273	6.7845	0.147390	366.99	386.55	1.7413
2	14.488	3.9819	18.864	0.053010	387.85	408.95	1.7413
3	8.7960	3.9819	19.442	0.051434	383.16	403.64	1.7227
4	49.908	11.9457	57.819	0.0017295	405.78	426.44	1.7227
6	46.138	11.9457	1120.0	0.89288	264.61	265.68	1.2193
8	8.7960	3.9819	1265.1	0.00079044	383.16	211.93	1.0427
9	-20.0	1.3273	36.880	0.027115	208.33	211.93	1.0515

Using tabular data, the following was calculated:

– specific mass cooling capacity

$$q_0 = h_1 - h_9 =$$
$$= 386.55 - 211.93 = 174.62 \text{ kJ/kg};$$

– specific work of vapor compression by a low-pressure compressor (LPC)

$$\left|l_{lpc}\right| = |h_1 - h_2| =$$
$$= |386.55 - 408.95| = 22.4 \text{ kJ/kg};$$

– specific work of vapor compression by a high-pressure compressor (HPC)

$$\left|l_{hpc}\right| = |h_3 - h_4| =$$
$$= |403.64 - 426.44| = 22.8 \text{ kJ/kg};$$

– total work of steam compression of LPC and HPC

$$l_к = \left(l_{lpc} + l_{hpc}\right) =$$
$$= 22.4 + 22.8 = 45.2 \text{ kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon_т = \frac{q_0}{l_к} = \frac{174{,}62}{45{,}2} = 3.863;$$

– flow rate of refrigerant R134a, which is sucked and compressed by the LPC

$$M_{lpc} = \frac{Q_0}{q_0} = \frac{200}{174.62} = 1.145 \text{ kg/sec};$$

– consumption of refrigerant R134a, which is sucked and compressed by HPC

$$M_{hpc} = M_{lpc}\frac{(h_2 - h_8)}{(h_3 - h_6)} =$$
$$= 1.145\frac{408.95 - 211.93}{403.64 - 211.93} = 1.177 \text{ kg/sec};$$

– theoretical power of LPC

$$N_{T,lpc} = M_{lpc} \cdot l_{lpc} = 1.145 \cdot 22.4 = 25.65 \text{ kW};$$

– theoretical power HPC

$$N_{T,hpc} = M_{hpc} \cdot l_{hpc} = 1.177 \cdot 22.8 = 26.84 \text{ kW};$$

– the heating coefficient of the refrigerant when compressing by LPC

$$\lambda_{lpc,W} = \frac{T_0}{T_{3-8} + 40} = \frac{253.15}{281.95 + 40} = 0.7863;$$

– the heating coefficient of the refrigerant when compressed by HPC

$$\lambda_{hpc,W} = \frac{T_{3-8}}{T_{5-6} + 40} = \frac{281.95}{319.29 + 40} = 0.7847;$$

– indicator efficiency of LPC

$$\eta_{lpc,i} = \lambda_{lpc,W} + b \cdot t_0 =$$
$$= 0.7863 + 0.0025 \cdot (-20) = 0.7363;$$

– indicative efficiency of HPC

$$\eta_{hpc,i} = \lambda_{hpc,W} + b \cdot t_{3-8} =$$
$$= 0.7847 + 0.0025 \cdot 8.8 = 0.8067;$$

– indicator power of LPC

$$N_{lpc,i} = \frac{N_{lpc,T}}{\eta_{lpc,i}} = \frac{25.65}{0.7363} = 34.84 \text{ kW};$$

– indicated power of HPC

$$N_{hpc,i} = \frac{N_{hpc,T}}{\eta_{hpc,i}} = \frac{26.84}{0.8067} = 33.27 \text{ kW};$$

– indicated power of the installation

$$N_{i,inst} = N_{i,lpc} + N_{i,hpc} =$$
$$= 34.84 + 33.27 = 68.11 \text{ kW};$$

– LPC feed rate

$$\lambda_{V,lpc} = \lambda_{C,lpc} \cdot \lambda_{throt} \cdot \lambda_{lpc,W} \cdot \lambda_{seal} =$$
$$= 0.9400 \cdot 0.98 \cdot 0.7863 \cdot 0.98 = 0.7099,$$

where is the coefficient of influence of the harmful volume of the LPC on its steam supply

$$\lambda_{C,lpc} = 1 - \text{c}\left(\frac{p_{3-8}}{p_{evap}} - 1\right) =$$
$$= 1 - 0.03\left(\frac{3.9819}{1.3273} - 1\right) = 0.9400;$$

– HPC feed rate

$$\lambda_{V,hpc} = \lambda_{C,hpc} \cdot \lambda_{throt} \cdot \lambda_{hpc,W} \cdot \lambda_{harm} =$$
$$= 0.9400 \cdot 0.98 \cdot 0.7847 \cdot 0.98 = 0.7084,$$

where is the coefficient of influence of the harmful volume of the HPC on its steam supply

$$\lambda_{C,hpc} = 1 - \text{c}\left(\frac{p_{5-6}}{p_{3-8}} - 1\right) =$$
$$= 1 - 0.03\left(\frac{11.9457}{3.9819} - 1\right) = 0.9400;$$

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– theoretical hourly volume of steam that is sucked and compressed by the LPC

$$V_{throt,lpc} = \frac{M_{lpc}}{\rho_1} = \frac{1.145}{6.7845} = 0.1688 \ \text{m}^3/\text{sec};$$

– actual hourly volume of LPC

$$V_{h,lpc} = \frac{V_{throt,lpc}}{\lambda_{V,lpc}} = \frac{0.1688}{0.7099} = 0.2377 \ \text{m}^3/\text{sec};$$

– theoretical hourly volume of steam that is sucked and compressed by the HPC

$$V_{throt,hpc} = \frac{M_{hpc}}{\rho_3} = \frac{1.177}{19.447} = 0.0605 \ \text{m}^3/\text{sec};$$

– actual, hourly volume of HPC

$$V_{h,hpc} = \frac{V_{throt,hpc}}{\lambda_{V,hpc}} = \frac{0.0605}{0.7084} = 0.08544 \ \text{m}^3/\text{sec};$$

– power consumed in the LPC for friction in kinematic pairs

$$N_{friction,lpc} = V_{h,lpc} \cdot p_{friction} = = 0.2377 \cdot 50 = 11.89 \text{ kW};$$

– effective (actual) power of the LPC

$$N_{e,lpc} = N_{i,lpc} + N_{friction,lpc} = = 34.84 + 11.89 = 46.73 \text{ kW};$$

– power consumed in HPC for friction

$$N_{friction,hpc} = V_{h,hpc} \cdot p_{friction} = = 0.08544 \cdot 50 = 4.27 \text{ kW};$$

– effecive (actual) power kW

$$N_{e,hpc} = N_{i,hpc} + N_{\text{friction,hpc}} = = 33.27 + 4.27 = 37.54 \text{ kW};$$

–effective power of a two-stage installation

$$N_{e,inst} = N_{e,lpc} + N_{e,hpc} = = 46.73 + 37.54 = 84.27 \text{ kW};$$

– effective (actual) cooling coefficient

$$\varepsilon_e = \frac{Q_0}{N_{e,inst}} = \frac{200}{84.27} = 2.373;$$

– condenser heat load (thermal pollution of the atmosphere)

$$Q_k = Q_0 + N_{i,inst} = = 200 + 68.11 = 268.11 \text{ kW}.$$

The most important indicators of the effectiveness of this *basic* two-stage refrigeration unit is summarized in Table 5, column 3.

Study of the suitability and efficiency of a classical single-stage refrigeration unit with limited regenerative heat exchange, sometimes used in practice, as an alternative to a two-stage installation with π *= 9.*

As mentioned above, to simplify the operation of a two-stage refrigeration unit, a single-stage unit with limited regenerative heat exchange is sometimes used in practice. This is due to the fact that the design and operation of two-stage refrigeration units are much more complicated than single-stage ones [12–20]. Such a thermodynamic cycle of a single-stage refrigeration unit with limited regenerative heat exchange is shown in Fig. 2.

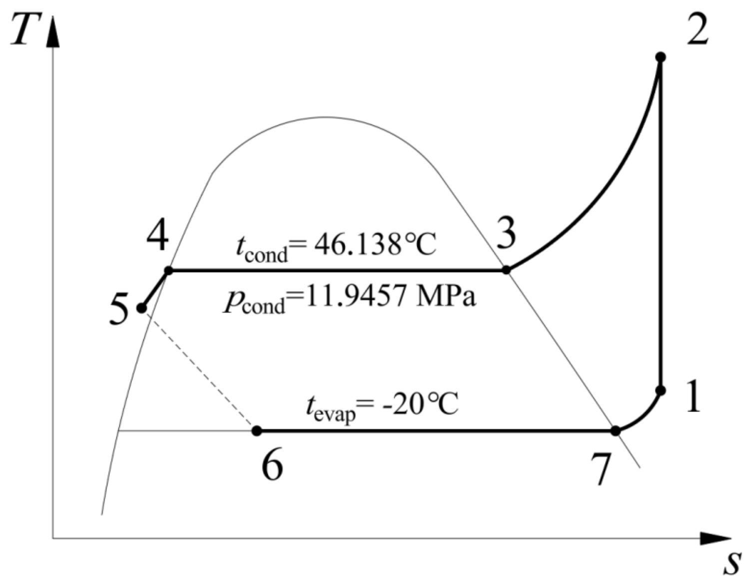

Fig. 2. Thermodynamic cycle of a single-stage refrigeration unit with limited regenerative heat exchange, as an alternative to a two-stage unit with π = 9

When constructing such a thermodynamic cycle, the superheat temperature of the vapor is usually set in the range of 5...10°C. Therefore, when determining the thermodynamic properties of R134a at point 1, it is accepted

$$t_1 = -10 \text{ °C},$$

$$p_1 = p_s = = f(-20 \text{ °C}) = 1.3273 \text{ bar}.$$

The enthalpy of the regeneratively supercooled liquefied refrigerant (point 5) was determined from the heat balance of the regenerative heat exchanger $(h_1 - h_7) = (h_4 - h_5)$.

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Table 2

Thermodynamic properties of R134a at characteristic points of the thermodynamic cycle of a single-stage refrigeration unit with limited regenerative heat exchange, as an alternative to a two-stage unit with π = 9

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
1	–10	1.3273	6.4739	0.154470	374.21	394.71	1.7729
2	64.645	11.9457	52.542	0.09032	420.28	443.02	1.7729
3	46.138	11.9457	59.521	0.016801	401.90	421.97	1.7088
4	46.138	11.9457	1120.0	0.00089288	264.61	265.68	1.2193
5	40.765	11.9457	1144.8	0.0008749	256.48	257.52	1.1937
6	–20	1.3273	17.089	0.058576	249.75	257.52	1.2316
7	–20	1.3273	6.7845	0.14739	366.99	386.55	1.7413

Using tabular data, the following was calculated:

– heat of superheating of steam (enthalpy difference at points 1 and 7)

$$\Delta h = h_1 - h_7 =$$
$$= 394.71 - 386.55 = 8.16 \text{ kJ/kg};$$

– enthalpy of liquefied supercooled refrigerant (point 5)

$$h_5 = h_4 - \Delta h =$$
$$= 265.68 - 8.16 = 265.68 \text{ kJ/kg};$$

– specific mass cooling capacity

$$q_0 = h_7 - h_5 =$$
$$= 386.55 - 265.68 = 120.87 \text{ kJ/kg};$$

– specific work in the isentropic vapor compression process

$$l_{\text{т}} = h_2 - h_1 =$$
$$= 443.02 - 394.71 = 48.31 \text{ kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon_{\text{т}} = \frac{q_0}{l_{\text{т}}} = \frac{120.87}{48.31} = 2.502;$$

– flow rate of R134a circulating in the system

$$M_{R134\text{a}} = \frac{Q_0}{q_0} = \frac{200}{120.87} = 1.6547 \text{ kg/sec};$$

– theoretical capacity of the refrigeration machine

$$N_T = M_{R134\text{a}} \cdot l = 1.6547 \cdot 48.31 = 79.94 \text{ kW};$$

– the heating coefficient of the refrigerant during its compression

$$\lambda_W = \frac{T_{6-7}}{T_{3-4} + 40} = \frac{253.15}{319.29 + 40} = 0.7929;$$

– compressor indicator efficiency

$$\eta_i = \lambda_W + bt_0 =$$
$$= 0.7929 + 0.0025 \cdot (-20) = 0.7429;$$

– compressor indicator power

$$N_i = \frac{N_T}{\eta_i} = \frac{79.94}{0.7429} = 106.74 \text{ kW};$$

– theoretical hourly volume of steam pumped through the system by the compressor

$$V_{throt} = \frac{M_{R134\text{a}}}{\rho_1} = \frac{1.6547}{6.4739} = 0.2556 \text{ m}^3/\text{sec};$$

– compressor delivery coefficient

$$\lambda_V = \lambda_C \cdot \lambda_{throt} \cdot \lambda_W \cdot \lambda_{seal} =$$
$$= 0.7600 \cdot 0.98 \cdot 0.7929 \cdot 0.98 = 0.5787,$$

where the coefficient of influence of the harmful volume in the compressor on the steam supply

$$\lambda_C = 1 - \text{c}\left(\frac{p_{3-4}}{p_{6-7}} - 1\right) =$$
$$= 1 - 0{,}03\left(\frac{11.9457}{1.3273} - 1\right) = 0.7600;$$

– effective hourly capacity of the compressor

$$V_h = \frac{V_{throt}}{\lambda_V} = \frac{0.2556}{0.5787} = 0.4416 \text{ m}^3/\text{sec};$$

– power consumed by the compressor on internal friction

$$N_{friction} = V_h \cdot p_{friction} =$$
$$= 0.4416 \cdot 50 = 22.08 \text{ kW};$$

– effective (actual) compressor power

$$N_e = N_i + N_{friction} =$$
$$= 106.74 + 22.08 = 128.82 \text{ kW};$$

– effective (actual) cooling coefficient

$$\varepsilon_{\text{E}} = \frac{Q_0}{N_e} = \frac{200}{128.82} = 1.5525;$$

– condenser heat load (thermal pollution of the atmosphere)

$$Q_k = Q_0 + N_i =$$
$$= 200 + 106.74 = 306.74 \text{ kW}.$$

The most important indicators of the effectiveness of this cycle for their further comparison with the indicators of other comparable cycles are also summarized in Table 5, column 4.

Study of the suitability and efficiency of a single-stage plant with a finite regenerative ***isobaric*** *steam superheating and the polytropic process of its compression, as an alternative to a two-stage installation with π = 9.*

The indicated thermodynamic cycle is proposed as an alternative to the two cycles considered above and is depicted in Fig. 3.

The limiting regenerative superheat of steam is depicted by line 7–1. In this case, firstly, superheating of steam to ambient temperature is ensured (dashed horizontal line), and secondly, the subcooling of the liquefied refrigerant (process 4–6) is significantly increased.

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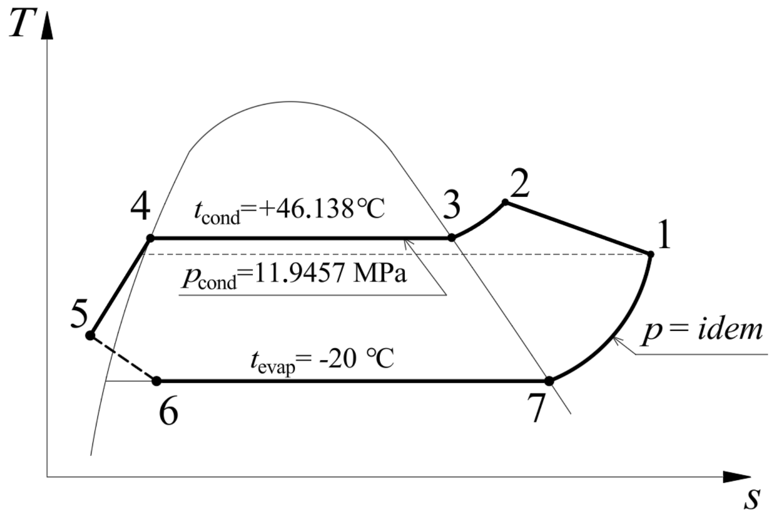

Fig. 3. Thermodynamic cycle of a single-stage refrigeration unit with a limiting isobaric regenerative steam superheating and polytropic compression process, as an alternative to a two-stage installation with π = 9

This reduces throttling losses during throttling of the liquefied refrigerant (process 5–6), which leads to an increase in the specific mass cooling capacity. Replacing the isentropic process of steam compression with a polytropic one reduces the specific work of steam compression and its temperature at the end of compression. To prevent hydraulic impact in the compressor, the temperature of the steam at the end of the polytropic compression process (pv^{n2}=idem, point 2) is taken 5 ℃ above the condensation temperature (point 3). The compressor and condenser are cooled by a refrigerant from the environment (water or air).

When calculating the thermodynamic properties of the supercooled liquefied refrigerant at point 5, the specific heat of limiting isobaric regenerative superheat of steam (process 7–1)

$$\Delta q_{superheat} = \Delta h = h_1 - h_7 =$$
$$= 433.58 - 386.55 = 47.03 \text{ kJ/kg},$$

and then the enthalpy of the liquefied supercooled refrigerant R134a at this point

$$h_5 = h_4 - \Delta h =$$
$$= 265.68 - 47.03 = 218.65 \text{ kJ/kg}.$$

Thermodynamic properties of R134a determined at characteristic points of this cycle are summarized in Table 3 in the order of their determination.

Table 3

Thermodynamic properties of R134a at characteristic points of the cycle of a single-stage refrigeration unit with limiting isobaric regenerative superheating of steam and a polytropic process of its compression, as an alternative to a two-stage unit with π = 9

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
7	–20	1.3273	6.7845	0.147390	366.99	386.55	1.7413
4	46.138	11.9457	1120.0	$0.89288 \cdot 10^3$	264.61	265.68	1.2193
2	51.138	11.9457	57.303	0.017451	407.02	427.87	1.7271
1	36.138	1.3273	5.3994	0.18541	408.97	433.58	1.9089
5	13.604	11.9457	1251.9	0.7988·103	217.70	218.65	1.0641

Using tabular data, the following was calculated:

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

$$|l_k| = T_{av}(s_2 - s_1) - (h_2 - h_1) =$$
$$= 316.7881(1.7271 - 1.9089) -$$
$$-(427.87 - 433.58) =$$
$$= 51.88 \text{ kJ/kg};$$

– specific mass cooling capacity

$$q_0 = h_7 - h_5 =$$
$$= 386.55 - 218.65 = 167.90 \text{ kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon_T = \frac{q_0}{|l_k|} = \frac{167.90}{51.88} = 3.236;$$

– the flow rate of refrigerant circulating in the system

$$M_{R134a} = \frac{Q_0}{q_0} = \frac{200}{167.90} = 1.1912 \text{ kg/sec};$$

– theoretical compressor power

$$N_T = M_{R134a} \cdot l_k = 1.1912 \cdot 51.88 = 61.80 \text{ kW};$$

– compressor power indicator

$$N_i = \frac{N_T}{\eta_i} = \frac{61.80}{0.95} = 65.05 \text{ kW},$$

where the indicated compressor efficiency is equal

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

heating coefficient λ_w=1 (heating of refrigerant vapor during the polytropic process of its compression is neglected);

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– coefficient of influence of harmful volume in the compressor on steam supply

$$\lambda_C = 1 - c\left(\frac{p_{5-6}}{p_{6-7}} - 1\right) =$$
$$= 1 - 0{,}03\left(\frac{11.9457}{1.3273} - 1\right) = 0.7600;$$

– full feed rate

$$\lambda_V = \lambda_c \cdot \lambda_{throt} \cdot \lambda_w \cdot \lambda_{seal} =$$
$$= 0.7600 \cdot 0.98 \cdot 1.0 \cdot 0.98 = 0.7299;$$

– theoretical hourly volume of steam pumped through the system by the compressor

$$V_{throt} = \frac{M_{R134a}}{\rho_1} = \frac{1.1912}{5.3994} = 0.2206\ \mathrm{m^3/sec};$$

– effective hourly capacity of the compressor

$$V_h = \frac{V_{throt}}{\lambda_V} = \frac{0.2206}{0.7299} = 0.3023\ \mathrm{m^3/sec};$$

– power consumed in the compressor for internal friction

$$N_{friction} = V_h \cdot p_{friction} =$$
$$= 0.3023 \cdot 50 = 15.11\ \text{kW};$$

– effective (actual) compressor power

$$N_e = N_i + N_{friction} = 65.05 + 15.11 = 80.16\ \text{kW};$$

– effective (actual) cooling coefficient

$$\varepsilon_e = \frac{Q_0}{N_e} = \frac{200}{80.16} = 2.495;$$

– condenser heat load (thermal pollution of the atmosphere)

$$Q_k = Q_0 + N_i = 200 + 65.05 = 265.05\ \text{kW}.$$

The most important indicators of the effectiveness of this cycle for their further comparison with the indicators of other comparable cycles are also summarized in Table 5, column 5.

Study of the suitability and efficiency of a single-stage refrigeration unit with a limiting regenerative ***isochoric*** *steam superheating and polytropic process of its compression, as an alternative to a two-stage installation with* π *= 9.*

The thermodynamic cycle of such a refrigeration unit is shown in Fig. 4.

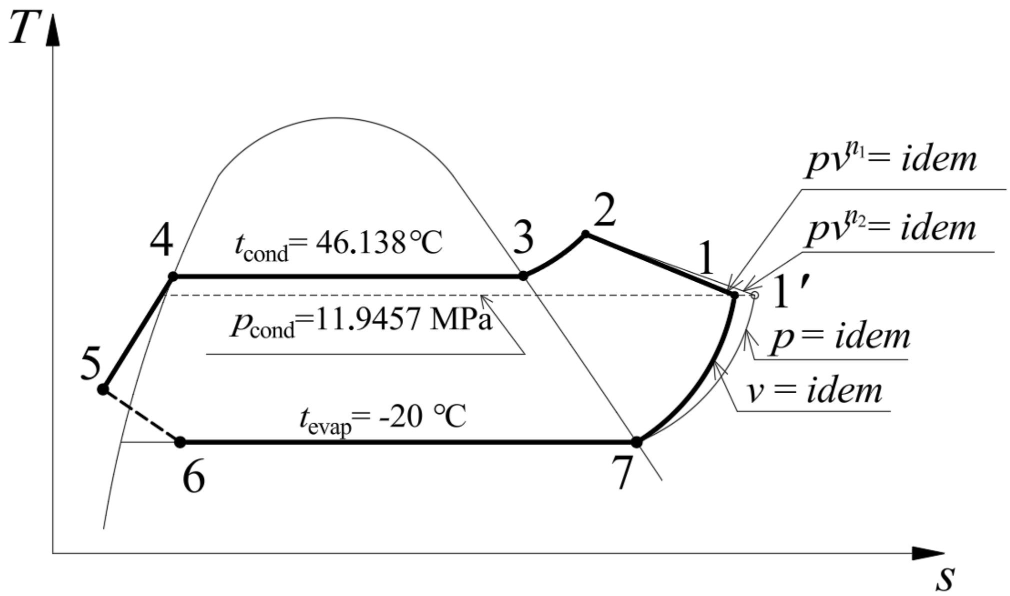

Fig. 4. Thermodynamic cycle of a single-stage refrigeration unit with a limiting isochoric regenerative steam superheating and polytropic process of its compression, as an alternative to a two-stage installation with π = 9

In this figure, the maximum regenerative (maximum possible) superheating of steam is carried out at *isochoric process v=idem* and is depicted by curve 7–1. The use of an isochoric process of superheating steam reduces the amount of heat for its superheating. From the point of view of regenerative subcooling of the liquefied refrigerant, this is bad, but very effective in terms of the work expended in the polytropic process of vapor compression by a compressor (*pv^{n1}=idem*).

To calculate the thermodynamic properties of the supercooled liquefied refrigerant at point 5, first calculate heat of the limiting isochoric regenerative superheat of steam (process 7-1)

$$q_{superheat} = u_1 - u_7 =$$
$$= 408.56 - 366.99 = 41.57\ \text{kJ/kg},$$

and then the enthalpy of the liquefied supercooled refrigerant (at point 5)

$$h_5 = h_4 - q_{superheat} =$$
$$= 265.68 - 41.57 = 224.11\ \text{kJ/kg}.$$

The determined thermodynamic properties of R134a at characteristic points of this cycle are summarized in Table 4 in the order of their determination.

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Table 4

Thermodynamic properties of R134a at characteristic points of thermodynamic cycle of a single-stage refrigeration unit with limiting isochoric regenerative superheating of steam and a polytropic compression process, as an alternative to a two-stage unit with π = 9

points	*t*, °C	*p*, bar	ρ, kg/m³	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)
7	–20	1.3273	6.7845	0.147390	366.99	386.55	1.7413
4	46.138	11.9457	1120.0	$0.89288 \cdot 10^{-3}$	264.61	265.68	1.2193
2	51.138	11.9457	57.303	0.017451	407.02	427.87	1.7271
1	36.138	1.6599	6.7845	0.147390	408.56	433.02	1.8894
5	13.604	11.9457	1251.9	$0.7988 \cdot 10^{-3}$	217.70	218.65	1.0641

Using tabular data, the following was calculated:

– specific mass cooling capacity

$$q_0 = h_7 - h_5 =$$
$$= 386.55 - 224.11 = 162.44 \ \text{kJ/kg};$$

– specific work consumed in the polytropic vapor compression process

$$|l_k| = T_{av}(s_2 - s_z) - (h_2 - h_1) =$$
$$= 316.788 \cdot (1.7271 - 1.8894) -$$
$$-(427.02 - 433.02) = 45.41 \ \text{kJ/kg};$$

– theoretical cooling coefficient

$$\varepsilon_\text{т} = \frac{q_0}{|l_k|} = \frac{162.44}{45.41} = 3.577;$$

– the flow rate of refrigerant circulating in the system

$$M_{R134a} = \frac{Q_0}{q_0} = \frac{200}{162.44} = 1.2312 \ \text{kg/sec};$$

– theoretical compressor power

$$N_T = M_{R134a} \cdot |l_k| =$$
$$= 1.2312 \cdot 45.41 = 55.91 \text{ kW};$$

– compressor indicator power

$$N_i = \frac{N_T}{\eta_i} = \frac{55.91}{0.95} = 58.85 \text{ kW};$$

where the indicated compressor efficiency is equal

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

where the steam heating coefficient λ_w=1 (heating of the refrigerant steam is neglected),

– theoretical volume of steam pumped through the system by the compressor per unit time

$$V_{throt} = \frac{M_{R134a}}{\rho_1} = \frac{1.2312}{6.7845} = 0.1815 \, \text{m}^3/\text{sec};$$

– the total feed rate is the same as in the previous version of the refrigeration unit –$\lambda_V = 0.7299$;

– effective hourly capacity of the compressor

$$V_h = \frac{V_{throt}}{\lambda_V} = \frac{0.1815}{0.7299} = 0.2487 \ \text{m}^3/\text{sec};$$

– power consumed in the compressor for internal friction

$$N_{friction} = V_h \cdot p_{friction} =$$
$$= 0.2487 \cdot 50 = 12.43 \text{ kW};$$

– effective (actual) compressor power

$$N_e = N_i + N_{friction} =$$
$$= 58.85 + 12.43 = 71.28 \text{ kW};$$

– effective (actual) cooling coefficient

$$\varepsilon_\text{e} = \frac{Q_0}{N_e} = \frac{200}{71.28} = 2.8058;$$

– condenser heat load (thermal pollution of the atmosphere)

$$Q_k = Q_0 + N_i =$$
$$= 200 + 58.85 = 258.85 \text{ kW}.$$

The most important indicators of its efficiency calculated in this cycle, as well as the indicators of the cycles calculated above, are summarized in Table 5, column 6, for the convenience/possibility of their comparative analysis.

Table 5

Comparison of the most important efficiency characteristics of the thermodynamic cycles of refrigeration units studied in the work

No./ No.	Performance indicator	Two-stage *basic* installation	Thermodynamic cycle of a single-stage plant		
			With *limited isobaric* steam overheating	with maximum *isobaric* steam overheating	with maximum *isochoric* steam overheating
1	2	3	4	5	6
1.	Cooling capacity of the unit, kW	200			
2.	Specific mass cooling capacity, kJ/kg	174.62	120.87	167.90	162.44
3.	Specific work of vapor compression, kJ/kg	45.2	48.31	51.88	45.41
4.	Theoretical installation capacity, kW	52.49	79.94	61.80	55.91
5.	Theoretical cooling coefficient	3.863	2.502	3.236	3.577
6.	Indicative power of the installation, kW	68.11	106.74	65.05	58.85
7.	Feed rate	0.7099 / 0.7084*	0.5787	0.7299	0.7299
8.	Hourly volume of compressor(s), m³/s	0.2377 / 0.08544*	0.4416	0.3023	0.2487

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Continuation of Table 5

1	2	3	4	5	6
9.	Effective (actual) power of the installation, kW	84.27	128.82	80.16	71.28
10.	Efficient refrigeration coefficient	2.373	1.5525	2.495	2.8058
11.	Condenser heat load (thermal pollution of the atmosphere)	268.11	306.74	265.05	258.85
12.	Relative change *effective power* in stallations, %		+52.9**	-4.88**	-14.88**
13.	Relative change of *effective cooling coefficient*, %		-34.6**	+5.14**	+18.23**
14.	Relative change of *thermal load on* condenser (thermal pollution of the atmosphere), %		+14.4**	-1.14**	-3.45**

**low pressure compressor/high pressure compressor performance*
***relative deviations were determined by the formula , i.e., the sign (+) means an increase in the indicator, the sign (-) means a decrease* $\delta y = \frac{(y_{basic} - y_{modified})}{y_{basic}} \cdot 100$

Conclusions

Analyzing the tabular data, we can unequivocally state:

- the most effective substitutes for two-stage refrigeration units (column 3) are single-stage refrigeration units with a limiting (maximum possible) *isochoric* (column 6) and *isobaric* (column 5) by regenerative steam superheating and polytropic compression process;

- the process with isochoric steam superheating is more efficient than with isobaric, but also more complex from the point of view of constructive implementation;

- the hourly volume of the compressor in these cycles is smaller than the sum of the volumes of the compressors of the basic two-stage cycle, which means that the weight and dimensions of such installations will be smaller;

- the reduction in the effective power of the compressors of these cycles relative to the corresponding indicators of the basic two-stage cycle is 4.88 % and 14.88 %, which means that the electricity consumption for the operation of refrigeration units and fuel for its generation are correspondingly reduced;

– almost the same ratio occurs between the values of the effective refrigeration coefficients of the compared cycles (5.14 % and 18.23 %);

– the reduction in the heat load of the condensers of the proposed refrigeration units relative to the same indicator of the basic two-stage unit shows how much the mass and dimensions of their condensers are reduced, and, most importantly, shows how much thermal pollution of the atmosphere is reduced (by 1.14 and 3.45 %, respectively);

– the modern use in practice of a single-stage refrigeration unit with limited regenerative heat exchange (column 4) instead of a two-stage unit is inefficient from many points of view: the effective power of the unit N_e increases, the refrigeration coefficient εe decreases, the overall dimensions of the condenser and, accordingly, thermal pollution of the atmosphere increase Q_k and so on.

Therefore, if instead of two-stage refrigeration units, single-stage refrigeration units are used according to a single-stage thermodynamic cycle with *isochoric* process of regenerative steam superheating and the polytropic process of steam compression, it is possible to significantly reduce the amount of electricity consumed. This means that thermal pollution of the atmosphere by power plants that produce electricity will decrease. In addition, the refrigeration plants themselves will pollute the atmosphere with less heat, which is extremely important in solving the current issue of today - preventing an increase in the temperature of the Earth's atmosphere.

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