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COMPLEX APPROACH TO THE CONVERSION OF EXISTING REFRIGERATION SYSTEMS TO A2L GROUP REFRIGERANTS

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Modern requirements for refrigeration equipment include the cessation of the use of systems with refrigerants that destroy the ozone layer, as well as a gradual reduction in the use of refrigerants with a high impact on global warming. The current task is to replace an environmentally unacceptable refrigerant with a neutral refrigerant for ozone and with a low global warming potential. The purpose of this paper is to develop and demonstrate a multivariate approach to the analysis of the specified problem – replacing HCFC and HFC refrigerants with refrigerants of the A2L group with a global warming potential below 500. Special attention is paid to the potential for increasing the productivity and energy efficiency of the refrigeration system. The following research tasks are solved in the paper: the impact of refrigerant replacement on the operation of the main elements of the system is determined; means and methods for increasing the cooling capacity of the refrigeration system when replacing the refrigerant are proposed; methods for increasing the energy efficiency of the refrigeration system are developed. The main changes in the operation of a refrigeration machine when replacing with a refrigerant of group A2L are identified in the paper. Namely, it is determined that the compressor performance changes, the lubricant needs to be replaced, it is necessary to take into account the influence of temperature glide, as well as changes in the operation of the condenser and evaporator. To increase the cooling capacity of a refrigeration machine, the following means and methods are proposed: selection of a refrigerant that can provide the required cooling capacity; increasing the compressor capacity either by frequency regulation or by installing an additional compressor; minimizing pressure losses in the hot steam and suction pipelines; reducing the temperature gradient on the condenser and evaporator; reducing the air temperature at the condenser inlet by adiabatic cooling; additional subcooling of the liquid refrigerant; optimizing the operating modes of the unit. The most effective method is determined – reducing the temperature difference between the condensation and boiling temperatures in the largest number of hours of the annual cycle.

Keywords: *refrigerant, temperature glide, cooling capacity, global warming potential, adiabatic cooling.*

Introduction

Modern requirements for refrigeration equipment include the cessation of the use of systems with refrigerants that destroy the ozone layer, as well as a gradual reduction in the use of refrigerants with a high impact on global warming. Any leak of the first group of refrigerants creates conditions for the complete cessation of operation of this system on the specified refrigerant due to the impossibility of refilling with a prohibited substance. There are two ways to solve this problem for owners of such systems: the first one is to completely replace the refrigeration system, taking into account the replacement in the long term of synthetic refrigerants with natural ones, namely carbon dioxide, propane or ammonia. In addition to increasing financial costs, the transition to one of the three natural refrigerants will require additional increased attention from the operating personnel to the safety of the refrigeration system. Also, the reason for increased safety measures is the increased pressures during the operation of CO_2 (R744) systems, especially during periods of high ambient temperatures, the high flammability of propane (R290) and the extreme toxicity and explosive flammability of ammonia NH_3 (R717). Apart from that, such a replacement will be significantly more expensive than previously purchased systems, but the most undesirable thing for existing production is the need to stop production during the dismantling of the old system and the installation and commissioning of the new one.

A possible alternative (second way), which fits into the current legislation, is less dangerous, and not so labor- and capital-intensive, and most importantly, operational, is the replacement of an environmentally unacceptable refrigerant with an ozone-neutral refrigerant with a low global warming potential (GWP). The proposed option is also not an absolutely ideal solution, since the proposed alternative refrigerants are dangerous in terms of flammability, as they are flammable. It is worth adding that, on the one hand, they are not as flammable as propane, because they ignite at a much higher concentration, but on the other hand, they have different degrees of GWP. Some are quite close to natural refrigerants in terms of performance, while others have higher performance. However, there is a third important fact, which is a certain loss of performance of the refrigerants being replaced. The latter option can be critically important for consumers, since it is directly related to the quality of refrigeration processing of products, and the loss of power by the refrigeration system is a violation of the established technological process.

As is known, the refrigeration system has significant potential to increase the established cooling performance. With the right approach to replacing refrigerants, it is possible not only significantly increase productivity, but also, thanks to the use of various technical solutions, reduce the energy consumption of the system, which, in turn, will reduce the negative impact on the environment and reduce operating costs.

Literature review

The issue of transition to a new generation of refrigerants, safe in terms of environmental performance and having high energy efficiency at the same time, has been the subject of study by many authors [1–7]. In [1], the problem of replacing (retrofitting) prohibited refrigerants with alternative working substances that are not inferior in their thermodynamic and operational characteristics to the replaced refrigerants was considered. A methodology, which was implemented in the form of a package of application programs for working bodies, such as refrigerants 134a and R410A, was proposed.

The refrigerant retrofit procedure is relevant for refrigeration equipment that is in operation. Taking the above into account, in [6], split air conditioners were considered as the most common systems in society that are actively operated. It was proposed to replace HFC refrigerants, for example, R410a, which is a zeotropic refrigerant, with R32 one. Based on the results of the experimental study and the analysis, it was concluded that in the same system, R32 refrigerant is suitable as a refrigerant to replace R410a refrigerant with a reduction in the cooling coefficient of performance (COP) by 4%, approximately.

In the study [7], the performance of a domestic air conditioner with the use of R32 to replace R410A is evaluated. One of the biggest challenges during retrofitting is determining the ideal refrigerant mass that will ultimately lead to optimal system performance. The experiments were conducted using a 2.5 kW split air conditioner. The results showed that replacement of R410A with R32 reduces the COP. Therefore, the authors of [7] do not recommend replacing R410A with R32 in air conditioners designed for R410A.

In [8], an experimental study of the heat transfer characteristics of the evaporation of a zeotropic mixture of R407C and a quasi-zeotropic mixture of R410A through a smooth tube of small diameter was presented. The results showed that R410A has a better heat transfer coefficient than R407C.

To evaluate the effect of cooling load on the performance of a domestic split air conditioner, an experimental study was conducted in [9] by replacing the refrigerant R22 with a hydrocarbon refrigerant (HCR22) during the retrofit. According to the results, the COP of R22 increases with increasing cooling load (0 W, 1000 W, 2000 W and 3000 W) by 16.10%, 12.66%, 16.56% and 19.99%, respectively. In addition, experimental data prove that HCR22 has better performance compared to R22, which indicates that HCR22 can be used to upgrade the existing RSAC with R22.

As can be seen from the literature review, the papers, mainly devoted to experimental studies, are local in nature with respect to low-capacity split systems, and therefore cannot be extended to other systems. In addition, the papers, except for [10], do not consider refrigerants of the A2L group. Thus, the development of an approach to replacing existing refrigeration systems of various capacities in operation with ozone-safe refrigerants remains the acute problem.

Purpose and objectives of the study

The purpose of this paper is to develop and demonstrate a multivariate approach to the analysis of the outlined problem – replacing HCFC and HFC refrigerants with A2L refrigerants with a GWP below 500. Special attention is paid to the potential for increasing the performance and energy efficiency of the refrigeration system.

Research objectives:

– to determine the impact of refrigerant replacement on the operation of the main elements of the system;

– to propose means and methods for increasing the cooling capacity of the refrigeration system when replacing the refrigerant;

– to develop methods for increasing the energy efficiency of the refrigeration system.

Indicators to consider when replacing refrigerants

When converting a refrigeration system to a new refrigerant, various indicators should be taken into account, such as: ozone depletion potential; GWP; temperature hysteresis – glide, compatibility with a certain type of lubricant; toxicity and safety group, lower flammability limit. Each of the above indicators is a serious condition for the use of a particular refrigerant.

It is worth noting that currently the use of systems in which a refrigerant with an ozone depletion potential other than 0 is considered prohibited.

The GWP indicator denotes the degree of influence of various gases on global warming. The unit is taken to be the influence of 1 t of CO_2. The effect of the emission of 1 t of a particular gas into the Earth's atmosphere is equivalent to the emission of 1 t of CO_2 multiplied by the GWP of this gas. Due to the spread of refrigerants with low GWP, more and more products that are classified as flammable appear. All currently available refrigerants with a GWP <500, with the exception of CO_2 (R744), are flammable, and some of them, such as ammonia (R717), are also toxic. There is a plan for a phase-out of high-GWP refrigerants, according to which refrigerants with an index of less than 500 should be considered after 2030 (Fig. 1).

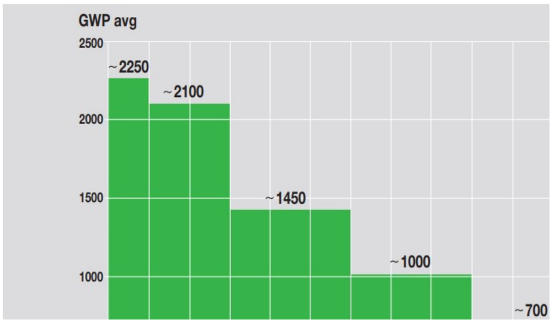

Fig. 1. Diagram. Theoretical average GWP due to gradual phase-out from 2015 to 2030 [11]

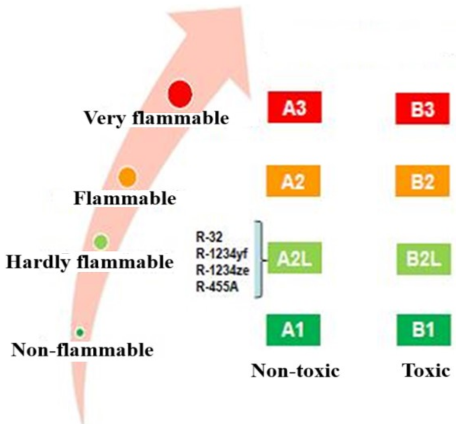

Fig. 2. Scheme. Toxicity and flammability [12]

The presence of temperature hysteresis (glide) means a discrepancy at the same pressure of the temperatures of the refrigerant components of the mixtures. This, in turn, affects the operation of heat exchange equipment and creates certain difficulties in adjusting and servicing the system. Changing the type of lubricant in the refrigeration circuit creates high requirements for cleaning the system when replacing the refrigerant.

The safety category shows the degree of toxicity and flammability of the refrigerant, which may require special attention to the volumes of refrigerant charging, replacement of certain elements and additional safety devices.

The classification of refrigerants from a safety point of view is carried out according to the ASHRAE 34 standard on the basis of their toxicity and flammability indicators obtained during standardized tests. It defines two categories of toxicity of refrigerants – A and B. Category A includes non-toxic refrigerants, category B – toxic ones. Four groups are distinguished for the classification of flammability of refrigerants: 1, 2L, 2, 3. The higher the number, the easier the refrigerant ignites. Category A2L includes non-toxic, hardly flammable refrigerants, for example, R-1234yf, R-454C, R-455A. They are characterized by lower flammability and a much lower burning rate than the flammable refrigerants A2 and highly flammable A3 (R290). Taking into account various criteria allows to draw up the final classification, presented in the scheme (Fig. 2).

Changes in the refrigeration machine when replacing the refrigerant

To simulate the system operation, a program from one of the leading manufacturers of Bitzer compressors is used [13]. The six-cylinder semi-hermetic compressor 6FE-44Y was taken as a model for the study. R22 is considered as the basic refrigerant to be replaced, and R454C is considered as an alternative one (Table 1).

This refrigerant has a low GWP <500 (148), is non-toxic and belongs to the highly flammable group (A2L). According to the manufacturer, the 6FE-44Y compressor can work with this refrigerant and have the best performance characteristics compared to other refrigerants of the A2L group [14].

Before making a replacement, it is advisable to make sure that the specific refrigerant is compatible with this compressor model. Special attention should be paid to measures to increase cooling capacity, improve energy efficiency and safety when working with A2L group refrigerants. Safety measures apply to all electrical equipment (heating elements of air coolers, connection of fan and compressor motors, electrical panels) and response systems for possible leaks of flammable refrigerant.

Table 1. R22 and alternative refrigerants [13]

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	R22	R407C	R422D	R438A	R290	R127
Group	HCFC	HFC	HFC	HFC	HC	HC
Components		R32/ 125/ 134a	R125/ 134a/ 600a	R32/ 125/ 134a/ 600/ 601a		
Application max (°C)	12	12	0	0	12	12
Application min (°C)	-45	-25	-40	-40	-40	-40
Appl. 2-stage max (°C)	-20				-20	-20
Appl. 2-stage min (°C)	-50				-55	-55
Oil 1	MO	POE	POE	POE	PAO	PAO
Oil 2	AB	PVE	PVE	PVE	PAG	PAG
Oil 3					POE	POE
Normal boiling point (°C)	-40,7	-43,8	-43,2	-42,3	-42,1	-47,6
Normal dew point (°C)	-40,7	-36,7	-38,3	-35,7	-42,1	-47,6
Temperature glide (K)	0	7,1	4,9	6,6	0	0
Crit. temp. (°C)	96	86	78	83	97	91

Next, we will consider step by step the impact on the main elements and the operation of the refrigeration machine as a whole when switching to A2L group refrigerants.

Lubricant selection when switching to A2L refrigerants

The characteristics of R22 and its alternative refrigerants are shown in Table 1. When replacing, attention should be paid to the type of lubricant – mineral or semi-synthetic, which differs from the synthetic lubricants used by the A2L group. The compatibility of lubricants is strictly regulated (Table 2). The maximum permissible content of mineral lubricant in synthetic lubricants should not exceed 1% [15]. Therefore, after removing old mineral lubricant from the system, the system must be thoroughly flushed with special fluids to remove its residues and other contaminants. Hoses and measuring instruments connected to the system must be clean and free of mineral lubricant residues.

Due to the high hygroscopicity of synthetic lubricants, the vacuuming operation must be performed especially carefully.

It is necessary to carry out high-quality welding work when adding additional elements to the system, as well as use fine filters on the compressor suction. Filter driers and indicator sight glasses must be replaced, taking into account the new refrigerant and lubricant.

Table 2. Lubricants for compressors [14]

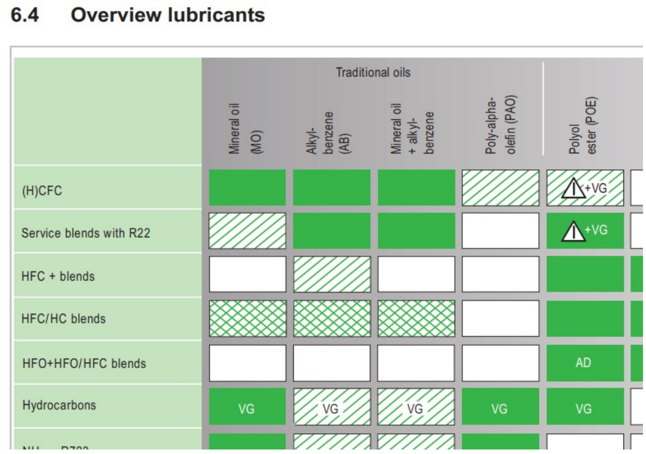


Table 3. Performance of the 6FE-44Y compressor with R22 and R454C at the same modes

Indicators	Refrigerant	
	R22	R454C
Boiling point t_{ev}, °C	-5	-5
Condensation temperature t_{cond}, °C	45	45
Liquid subcooling t_{sc}, K	1	1
Total superheat Σt_{sh}, K	8	8
Evaporator superheat t_{sh}, K	6	6
Rotation frequency *freq*, Hz	50	50
Cooling capacity Q, kW	93.8	78.1
Compressor consumption N, kW	33.4	30.5
Condenser load QN, kW	128	110
COP	2.80	2.56

Changes in compressor performance when switching to A2L refrigerants

When replacing different refrigerants, they will provide excellent cooling capacity, which the compressor will provide at the same boiling and condensation temperature conditions. For the 6FE-44Y compressor, when replacing R22 with R454C, the cooling capacity will decrease by approximately 20% at the conditions specified in Table 3. Under the same conditions, the compressor COP will decrease by

9.4%. However, changes that will also affect the heat exchange equipment will change this ratio. The main reason for this will be the temperature glide of R454C.

Effect of temperature glide

Some refrigerants, which are mixtures of several chemicals, may have temperature hysteresis (glide) – a difference (slip) in the temperatures of the compounds at the same pressure during phase transitions. Table 1 shows refrigerants with a glide of 7.1 K (R407C), 4.9 K (R422D), 6.6 K (R438) and 7.81 K (R454C). These are the maximum possible differences in phase transition temperatures. Depending on the specific pressures, the values may be slightly smaller. Therefore, the system should be charged and refilled only in the liquid phase [15]. When working with such refrigerants, it is not possible to determine the temperature of the refrigerant in the evaporator or condenser by reading the pressure gauge. Reference literature and calculation programs give two different temperatures for a given pressure: the dew point temperature (final temperature of the vapor at boiling or the beginning of condensation) and the boiling point (initial boiling temperature or final temperature of condensation). This affects: the performance of the heat exchange equipment (evaporators and condensers), the setting of the thermostatic valve (TSV), the analysis of the operation and performance of the entire system.

Features of condenser operation when replacing refrigerant

The difference in condenser operation at temperature glide is as follows (Table 4). For refrigerant R454C at a pressure of 16.54 bar, condensation will begin at a temperature of 45 °C. For refrigerant with zero glide, condensation occurs at the same temperature, for example, at 45 °C. Instead, a decrease in the temperature in the condenser below 45 °C indicates subcooling of the liquid. For R454C, vapor condensation will end at 38.3 °C, and only after this temperature, with its further decrease, will subcooling of the liquid refrigerant begin. In this case, the condenser power will decrease due to a decrease in the average logarithmic temperature. And with the same temperature difference between the environment and the beginning of condensation, instead of 128 kW for refrigerant R22, it will provide only 84 kW for refrigerant R454C. Even taking into account the total loss of compressor performance, the condensation pressure will increase slightly.

Table 4. Compressor performance (6FE-44Y) on R22 and R454C taking into account the change in Δt on the condenser and evaporator

Indicators	Refrigerant	
	R22	R454C
Boiling point t_{ev}, °C	-5	-5
Condensation temperature t_{cond}, °C	45.0	47.1
Liquid subcooling t_{sc}, K	1	1
Total superheat Σt_{sh}, K	8	8
Evaporator superheat t_{sh}, K	6	6
Rotation frequency *freq*, Hz	50	50
Cooling capacity Q, kW	93.8	82.0
Compressor consumption N, kW	33.4	32.1
Condenser load QN, kW	128	115
COP	2.80	2.5

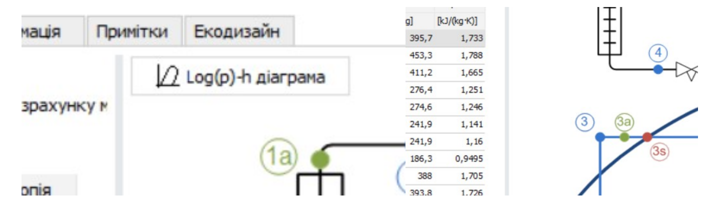
Changes in evaporator operation when replacing refrigerant

More serious changes occur in the evaporator. If we neglect pressure losses in heat exchange equipment, pipelines and fittings, the picture will be as follows. For refrigerant R454C at a pressure in the evaporator of 3.821 bar, the temperature of the beginning of boiling of the refrigerant on the boiling line will be -12.8 °C, the temperature at the outlet of the expansion valve – -10.3 °C, the dew point on the boiling line – -5 °C, the temperature at the outlet of the evaporator will be 1 °C at a superheat of 6 K, the temperature at the inlet of the compressor valve on the suction side taking into account full superheating – 8 K, of the compressor – 3 °C.

Thus, if with the refrigerant R22 the boiling point in the evaporator was -5 °C at a constant pressure of 4.22 bar (abs.) and did not change until the entire liquid fraction boiled off, then in the case of R454C at a constant pressure of 3.957 bar, which corresponds to a dew point on the boiling line of -4 °C, the temperature in the evaporator during the boiling process changes from -10.3 °C to -5 °C. The potential performance of the evaporator in this mode increases from 93.8 kW for R22 to 118 kW for R4545C. However, given that the compressor performance is less than the specified value, a decrease in the dew point temperature difference on the evaporator from -5 °C to -3 °C should be expected. Thus, the operation with the change of refrigerant affects the characteristics of the heat exchange equipment, which, in turn, affects the performance of the refrigeration machine as a whole (Table 5).

The boiling and condensation temperatures of the R454C refrigerant on the dew point line will increase from the expected values 1, taken in a similar operating mode of the equipment on R22, to the established values taking into account the new temperature differences on the condenser and evaporator 2 (Table 5).

Table 5 shows the data for the points shown in the refrigeration machine diagram and the refrigeration cycle diagram (Fig. 3), under the following conditions: subcooling in the condenser 1 K; additional subcooling 0 K; superheating in the evaporator 6 K; superheating after the evaporator 2 K; pressure loss on the suction line is equivalent to 0 K; pressure loss on the discharge line is equivalent to 0 K.



a b

Fig. 3. Refrigerant state points on the refrigeration machine schematic (a) and cycle diagram (b) [16]

Table 5. R454C refrigeration system performance

Point	Processes in system elements	Temperature, °C		Pressure (abs.), bar	
		1	2	1	2
1	Compressor suction	3	5	3.821	4.096
2	Compressor discharge	73.7	75.2	16.54	17.44
2s	Dew point at the condensation line	45.0	47.1	16.54	17.44
3s	End of condensation at the condensation line	38.3	40.5	16.54	17.44
3a	Condenser outlet	37.3	39.5	16.54	17.44
3	Expansion valve inlet (including additional subcooling)	37.3	39.5	16.54	17.44
4	Expansion valve outlet	-10.3	-8.2	3.821	4.096
4s	Minimum possible boiling point	-12.8	-10.8	3.821	4.096
1s	Dew point at the boiling line	-5	-3	3.821	4.096
1a	Evaporator outlet	1	3	3.821	4.096

Methods for increasing the cooling capacity of a refrigeration system

Although increasing the evaporator capacity allows the system to operate at a higher capacity than expected, the system with R454C refrigerant will still have 13% less than the design cooling capacity.

There are at least seven ways to increase the cooling capacity of a refrigeration system:

1. Increasing compressor performance:
– by frequency control;
– by additional compressor.
2. Selection of a refrigerant that can provide the required cooling capacity.
3. Minimization of pressure losses (ΔP) in hot steam and suction pipelines:
– by reducing linear losses;
– by reducing local losses.
4. Reducing the temperature gradient (ΔT) on the condenser and evaporator:
– by increasing the heat exchange area;
– by increasing the heat transfer coefficient.
5. Reducing the air temperature at the condenser inlet by adiabatic cooling:
– single-stage cooling;
– two-stage cooling.
6. Additional subcooling of the liquid refrigerant:
– by using heat exchangers;
– by using refrigeration machines.
7. Process optimization:
– by increasing the coefficient of operation of the refrigeration machine;
– by reducing the thermal loads on the refrigeration system.

Increasing the cooling capacity of the compressor

One of the fairly affordable methods of increasing the cooling capacity of a compressor is the use of a frequency converter (inverter), which, by changing the motor speed, can change the volumetric capacity. Installing an external frequency converter is possible under three conditions. First: when increasing the speed, the engine should not consume a current higher than that specified by the compressor manufacturer. Second: the lubrication system in the compressor should be designed for the use of an inverter, especially when reducing the speed, when the quality of lubrication of friction elements may deteriorate. Third: reducing the speed can worsen the quality of cooling of the motor windings by the sucked-in steam. In general, manufacturers indicate the possibility of using a specific compressor with a frequency converter, the range of performance adjustment, and the maximum operating current in the equipment characteristics. Calculation programs make it possible to assess the effect that the frequency converter provides under specific conditions. It should be taken into account that this method has a number of advantages and disadvantages. Among the advantages, in addition to increasing the cooling capacity, is the possibility of smooth power regulation depending on the thermal load on the compressor, a significant reduction in starting currents. The disadvantage of this method is the additional consumption of electrical energy by the frequency device itself (approximately 5% of the compressor consumption) and a decrease in the total COP due to the corresponding increase in Δt on the heat exchangers. The potential is to increase the performance of the 6FE-44Y compressor thanks to the inverter by up to 40%. It should be noted that increasing the compressor performance is not the same as increasing the performance of the refrigeration machine as a whole, because such changes affect the operation of the heat exchange equipment (increase in temperature gradients) and increase in pressure losses in pipelines and other elements of the hydraulic circuit. From Table 6, it can be seen that increasing the speed by 20%, from 50 to 60 Hz, gives the effect of increasing the overall performance of the refrigeration machine by 13% compared to the previous level.

Table 6. Compressor (6FE-44Y) performance taking into account the use of a frequency converter

Indicators	Refrigerant + device		
	R22	R454C	R454C + FC
Boiling point t_{ev}, °C	-5	-3	-3,3
Condensation temperature t_{cond}, °C	45.0	47.1	48.6
Liquid subcooling t_{sc}, К	1	1	1
Total superheat Σt_{sh}, К	8	8	8
Evaporator superheat t_{sh}, К	6	6	6
Rotation frequency *freq*, Hz	50	50	60
Cooling capacity *Q*, kW	93.8	82.0	93.9
Compressor consumption *N*, kW	33.4	32.1	39.8
Condenser load *QN*, kW	128	115	135
COP	2.80	2.55	2.36

If the installation of a frequency converter (FC) is not possible, an additional compressor of appropriate capacity can be integrated into the system.

Using a different refrigerant

Among the refrigerants available for use with a specific compressor model, unfortunately, there are none that are able to increase the cooling capacity to the required (initial) level.

Minimization of costs in hot steam and suction pipelines and automation elements

Systems with sufficiently large distances between the heat exchange equipment and the compressor usually have linear pressure losses in the pipelines. Besides, additional pressure losses may be on any elements of refrigeration automation located on the steam pipelines. When designing, it is assumed that the total losses can be up to 2 K (equivalent). A decrease in the condensation temperature by 1 K increases the compressor cooling capacity by 1.6–2.8% (Table 7). An increase in the evaporation temperature by 1 K increases the compressor cooling capacity by 4.03–6.31% (Table 7), i.e., if there are maximum permissible pressure losses, the system has the potential for improvement. First of all, this is either the use of pipelines with a larger diameter, or the installation of additional parallel pipelines. However, in each case, it is necessary to compare the total costs of such methods with the effect that they can provide compared to other methods. In addition, it is necessary to take into account the fact that increasing the diameter of the pipelines affects the reduction of the speed of the refrigerant, which, in turn, can lead to a deterioration in the circulation of the refrigeration oil in the system, its poor return to the compressor crankcase.

Table 7. Compressor (6FE-44Y R454C) performance improvement (%) with decreasing and increasing condensing temperature

Change in condensation temperature, К	Boiling point, °C			
	0	-10	-20	-30
-1	1.69	1.91	2.08	2.80
-5	8.20	8.73	9.83	11.86
-10	15.20	16.10	18.10	21.20
+1	4.03	4.49	5.05	6.31
+5	18.60	20.10	22.60	26.80

As for individual automation elements, which may have significant pressure losses, two ways can also be considered. The first one is to replace these elements with new ones with greater throughput capacity. The second one is to install the same elements in parallel along the refrigerant flow.

The potential to increase the cooling capacity of the compressor, due to the reduction of pressure losses, is approximately 8 to 18%, depending on the operating modes of the system. It is also important to understand that the potential to reduce pressure losses on the discharge and suction lines is not the same as the potential to increase the cooling capacity of the refrigeration machine as a whole.

In the considered example, the system did not have additional pressure losses. However, if the dew point temperatures were closer by 2 K on the condensation side and by 2 K on the suction side, the compressor capacity would increase by ≈11%.

Reducing the temperature gradient Δ*t* on the condenser and evaporator

The increase in cooling capacity is possible by reducing the temperature gradient on the main heat exchangers of the refrigeration machine: for reducing the condensation temperature and/or increasing the evaporation temperature, this is a more effective means than the previous one. Firstly, this is possible by increasing the heat exchange area either by replacing the old one with a new, more powerful heat exchanger, or by installing an additional one in parallel. Secondly, the heat transfer coefficient can be increased by either increasing the fan motor speed or installing a more powerful fan. Of course, in both cases, if possible, the first method can lead to a 50% reduction in the temperature gradient, but provided that the heat exchanger filling control system, for example, such as an expansion valve, allows this. The improvement potential of the second method is approximately up to 20%. The final efficiency depends on how large the gradient was initially. Typically, for an evaporator with a DX connection scheme at Δt=10 K, we can make an improvement of 3.4 K, for a condenser with Δt=10 K – up to 5 K. More effective options may also exist, but they will be too complex and expensive. It should be understood that the potential for improving heat transfer is not the same as the potential for increasing the cooling capacity of the refrigeration machine as a whole, because just like in the previous methods, it directly affects the compressor performance.

Reducing the air temperature at the condenser inlet by adiabatic cooling of water

This method allows to increase the cooling capacity not only of machines in which such a need arose due to the transfer to another refrigerant, but also when in the warm season due to a significant increase in the condensation temperature the cooling capacity of all refrigeration machines with air-cooled condensers decreases. A more effective way to improve this situation than the previous one is to use the effect of reducing the air temperature due to adiabatic cooling during water evaporation. The higher the air temperature, the lower its relative humidity, the higher the temperature gradient of cooling. The use of modern adiabatic surfaces allows to cool the air very close to the so-called wet bulb temperature. For example, at a dry bulb temperature of 35 °C and a relative humidity of 25%, the wet bulb temperature will be 20.3 °C, and the temperature of the air passing through the humidification chamber or adiabatic surface, depending on the saturation coefficient, can be 21–23 °C. Two-stage adiabatic cooling can be used when the air temperature is between the dew point (12 °C) and wet bulb (20.3 °C) temperatures. Two-stage evaporative cooling is carried out as follows. The ambient air at the inlet to the direct evaporative cooling heat exchanger is pre-cooled as it passes through an air-to-water heat exchanger. The water for this heat exchanger is pre-cooled as it passes through a separate multi-layer direct evaporative cooling heat exchanger [17].

The calculations must take into account the fact that installing additional adiabatic surfaces on the condensers reduces the air flow through the heat exchange surface, i.e., on the one hand, we reduce the temperature of the air entering the condenser, and on the other hand, we increase Δt.

In our case, reducing the condensation temperature in the system from R454C to 38.6 °C due to single-stage adiabatic cooling will allow us to fully meet the design capacity conditions (Table 8). It should be noted that changing the compressor operating conditions increases the temperature gradients on both the evaporator and condenser. Therefore,

Table 8. Compressor (6FE-44Y) performance taking into account the use of adiabatic cooling of air at the condenser inlet (AC)

Indicators	Refrigerant + device		
	R22	R454C	R454C + AC
Boiling point t_{ev}, °C	-5.0	-3.0	-3.2
Condensation temperature t_{cond}, °C	45.0	47.1	38.6
Liquid subcooling t_{sc}, K	1	1	1
Total superheat Σt_{sh}, K	8	8	8
Evaporator superheat t_{sh}, K	6	6	6
Rotation frequency *freq*, Hz	50	50	50
Cooling capacity *Q*, kW	93.8	82.0	93.9
Compressor consumption *N*, kW	33.4	32.1	29.3
Condenser load *QN*, kW	128	115	124
COP	2.80	2.55	3.21

an improvement in the condensing temperature that could only affect the compressor by about 17%, resulted in an improvement of 13% on the system as a whole due to the increase in the temperature gradient on the evaporator.

Additional subcooling of liquid refrigerant

The most effective method of increasing the cooling capacity is additional subcooling of the liquid refrigerant before the TSV (Table 9). An increase in the subcooling of the refrigerant by 1 °C gives a 1.5–2% increase in the cooling capacity, i.e. the same mass of refrigerant can receive more heat under appropriate conditions in the evaporator if it has a lower temperature at the inlet to the TSV.

Table 9. Compressor (6FE-44Y R454C) performance improvement with additional liquid subcooling

Changing settings	Boiling point, °C											
	0			-10			-20			-30		
Change of subcooling, K	1	10	20	1	10	20	1	10	20	1	10	20
Power growth, %	1.06	10.30	18.40	1.12	10.70	19.10	1.30	11.10	19.80	1.42	11.90	20.90

Technically, the liquid refrigerant is subcooled by internal and external subcoolers. These can be heat exchangers or refrigeration machines. Internal units additionally load the condenser of the refrigeration machine, which requires improvement. External units have their own heat exchange surface to transfer heat to the external environment. Heat exchangers consume many times less electrical energy than refrigeration machines. However, the amount of subcooling of external heat exchangers depends on the external environmental conditions, and refrigeration machines can be designed for any subcooling. The greater the amount of subcooling of the liquid at the inlet to the TSV, the more heat the evaporator can take from the object being cooled. Usually there are some limitations. TSV manufacturers set the recommended value of the vapor content at throttling as x>0.2 (20%) regardless of the used refrigerant (Fig. 4). A lower vapor content during throttling can lead to rapid frictional wear of the valve. Approximate minimum subcooling temperatures for R454C is given in Table 10.

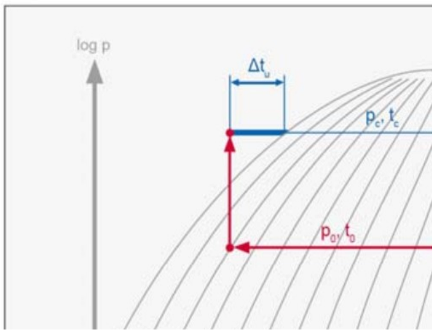


Fig. 4. Diagram for determiningthe maximum possible subcooling for a control valve (vapor content in the valve is 20%) [18]

As mentioned above, heat exchangers-subcoolers depend on the external conditions. The temperature glide of R454C 7.8 K in the system under consideration and having a temperature gradient of 12.1 K between the air temperature at the inlet to the condenser and the dew point temperature at the condensation line, practically makes it impossible to use an additional air subcooler. In the system under consideration, at an air temperature at the inlet to the condenser of 35 °C, the temperature of the refrigerant leaving the condenser is39.5 °C, i.e. a difference of 4.5 K has the potential to be up to 4%, which in this case is not a significant improvement in the cooling capacity by subcooling the liquid. Substantial subcooling, which can provide 13% of the cooling capacity that

Table 10. Minimum liquid subcooling temperature for R454C

"Dew point" temperature on the boiling line	Pressure (abs), bar	Minimum subcooling temperature, °C
0	4.54	23.4
-5	3.82	19.7
-10	3.20	15.9
-15	2.65	12.0
-20	2.18	8.1
-25	1.78	4.6
-30	1.44	5.9
-35	1.15	5.2
-40	0.91	4.6

Table 11. Compressor (6FE-44Y) performance using external subcooler (ES)

Indicators	Refrigerant + device		
	R22	R454C	R454C + ES
Boiling point t_{ev}, °C	-5.0	-3.0	-3.2
Condensation temperature t_{cond}, °C	45.0	47.1	47.0
Liquid subcooling t_{sc}, K	1	1	14
Total superheat Σt_{sh}, K	8	8	8
Evaporator superheat t_{sh}, K	6	6	6
Rotation frequency *freq*, Hz	50	50	50
Cooling capacity *Q*, kW	93.8	82.0	93.9
Compressor consumption *N*, kW	33.4	32.1	32.0*
Condenser load *QN*, kW	128	115	114
COP	2.8	2.55	2.94

*excluding subcooler electrical consumption the system lost when switching from R22 to R454C, will be at a liquid temperature at the inlet to the expansion valve of 26.4 °C (Table 11). This level of subcooling can be achieved by air heat exchangers-subcoolers with multilayer adiabatic panels with one or two-stage cooling. Effective evaporation of water occurs in irrigated heat exchangers consisting of heat exchange elements containing a layered heat-conducting material with a hydrophilic and hygroscopic coating [19]. In other cases, this can be achieved with higher energy costs by subcoolers based on separate refrigeration machines.

Optimization of refrigeration system processes

Any technical solution, even the best one, can have the potential for improvement. This applies to both an individual refrigeration machine and the entire refrigeration system in which it operates. And this potential for efficient use of the available refrigeration capacity lies in increasing the operating time factor of the refrigeration machine and reducing the thermal loads on the refrigeration system. Such methods can be: accelerating the defrosting process of air coolers, reducing heat losses during defrosting, reducing excessive icing of heat exchange surfaces, and accumulating cold.

A refrigeration machine can produce more "cold" by increasing the operating time factor, thanks to optimizing the processes associated with defrosting air coolers. Defrosting time takes up from 10 to 20% of the working time in storage rooms per day. Reducing defrosting time gives the refrigeration machine more time to produce "cold". Such measures may include: accelerating the defrosting time when replacing air defrosting with electric or hot steam, activating defrosting not by time, but by demand, organizing sequential defrosting on air coolers.

Measures that can lead to a reduction in heat losses during defrosting will generally reduce the calculated load on the refrigeration equipment. In addition to accelerating the cycle, this can be done by automatically closing the free openings of air coolers during defrosting, or by additional thermal insulation of the housings.

Defrosting time can also be reduced by limiting the impact of excessive icing. Installing curtains and air curtains, organizing vestibules, high-speed automatic doors and devices for tight adjacency of vehicles to cargo openings, optimizing cargo logistics in the chamber are the main measures that reduce air infiltration through open openings to the chamber.

Increasing the heat exchange surface and improving heat transfer reduce temperature deltas on evaporators, which significantly affects moisture condensation [15].

Relative air humidity, in turn, is a function of the heat inflow into the chamber, which determines the duration of operation of cooling devices, and therefore their drying effect. Studies have shown that "the maximums of drying coincide in time with the maximums of heat inflows" [20].

Effective measures to reduce the heat load can be: additional thermal insulation, reducing the consumption of lighting by electric fans of air coolers, etc.

Cold accumulation is an additional tool for increasing capacity. Daily fluctuations in outdoor air temperature and uneven operational loads create conditions under which the average load on refrigeration equipment is significantly less than the calculated peak. At times of load reduction, opportunities arise for the accumulation of "cold".

In cold storage rooms and "in large production facilities with 2–3-shift operation, the heat load schedule is more uniform (for example, as in Fig. 5). With such fluctuations in heat load, the use of systems with cold accumulation becomes impractical. Recommendations developed and published in [23] advise accumulating cold when the duration of peak heat loads is no more than 4 hours per day and when it exceeds the average daily heat load by more than 40%" [21]. This applies more to traditional solutions using liquid cooling systems. However, accumulation is possible even for these conditions. Both the product and heavy internal building structures can be used as an accumulator.

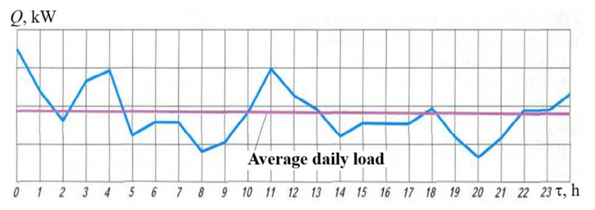


Fig. 5. Daily heat load schedule

Manufacturers of refrigeration equipment also suggest placing special containers with a eutectic medium in the cold air flow, which effectively accumulates "cold" during the operation of the refrigerator and releases it when the equipment stops due to temperature or for defrosting.

Methods for increasing the energy efficiency of a refrigeration system

The energy consumption of a refrigerator is an equally important factor influencing the volume of CO_2 emissions, because thermal power plants that produce this energy constantly emit CO_2 into the atmosphere. The average global carbon intensity of electricity in 2018 was about 479 g CO_2/(kW·h) [22].

Therefore, it is necessary to carry out modernization in such a way that, at a minimum, it does not worsen the previous level of consumption, and, at a maximum, it makes the operation of the refrigerator more efficient.

In the example under consideration, replacing R22 with R454C will lead to a change in the main indicator of the refrigerator – COP at maximum loads and ambient temperatures:

– under the same conditions of boiling and condensation (Table 2): -8.6%;
– at the established mode (Table 4): -8.9%;
– when using a variable speed drive or additional compressor (Table 6): -15.7%;
– when using adiabatic air cooling (Table 8): +14.6%;
– when using an adiabatic subcooler (Table 11): +5%.

Therefore, from the point of view of the need to improve the COP, the use of frequency control or the use of an additional compressor will only worsen the energy efficiency situation at high air temperatures. However, the ability to reduce excess cooling capacity by maintaining a stable refrigerant boiling pressure for most of the year will provide a significant improvement in the annual COP indicator.

The method of using adiabatic technologies provides high COP indicators only at high ambient temperatures, i.e. during the year in a fairly short period of time. Provided that the minimum condensation temperature is maintained in the system for most of the year, due to low air temperatures, this method does not give a special advantage when used in industrial refrigeration systems. However, it can be highly effective when used in air conditioning systems in the summer.

Effective improvement of COP is facilitated by methods that provide a minimum delta (Δt) between the condensation and boiling indicators for the largest number of hours of the annual cycle. And the fundamental difference in the efficiency of the systems is almost not in the choice of refrigerant. So, in part, "the choice of the working fluid directly affects the COP. In practice, however, the COP remains almost constant for a wide range of refrigerants with significantly different pressures and densities, if the evaporation and condensation temperatures are the same. It should be noted that the COP remains constant within ... +/- 10%" [23].

The COP values for two refrigerants at the same operating conditions are shown in Table 12.

It is enough to reduce the temperature difference Δt to 4–5 K to increase the COP of R454C to the level of R22. And this can be done either by improving the operation of the condenser or evaporator, or by reducing pressure losses in the discharge and suction lines.

Table 12. Average COP for R22 and R4545C depending on temperatures

Condensation temperature (dew point at the condensation line), °C	Average value of COP at boiling point -5 °C (dew point at the boiling line)	
	R22	R454C
50	2.48	2.23
45	2.80	2.56
40	3.19	2.94
35	3.64	3.36
30	4.20	3.86
25	4.89	4.45
20	5.79	5.17

Fig. 6. COP values of R454C at different boiling points and different Δt differences between dew point temperatures on the condensation and boiling lines

The change in the COP value of refrigerants directly depends on the difference in boiling and condensation temperatures (Fig. 6). If we take the average COP value, then at different boiling temperatures in our example for R454C the values differ by no more than 6%.

In each specific case, there may be one or another way to increase cooling capacity and energy efficiency, or a combination of several.

Conclusions

The main goal of the provided material is to demonstrate various methods of transition without deterioration of cooling technology conditions using the example of replacing R22 refrigerants in a vapor compression refrigeration machine with A2L group refrigerant (R454C). This transition should be not only environmentally friendly and safe, but also powerful and energy efficient.

The main changes in the operation of a refrigeration machine when replacing it with A2L group refrigerant, namely changing the compressor performance, replacing the lubricant, taking into account the influence of temperature glide, and changing the operation of the condenser and evaporator, are identified in the paper.

Means and methods to increase the refrigeration capacity of the refrigeration machine, as well as methods for increasing the energy efficiency of the refrigeration system are proposed, and the most effective method is determined – reducing the temperature difference Δt between the condensation and boiling temperatures in the largest number of hours of the annual cycle.

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Комплексний підхід до переведення існуючих холодильних систем на холодоагенти групи A2L

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Сучасні вимоги до холодильної техніки передбачають припинення використання системі з холодоагентами, що руйнують озоновий шар, а також поступове зменшення застосування холодоагентів із високим показником впливу на глобальне потепління. З огляду на сказане актуальним завданням сьогодення є заміна екологічно неприйнятного холодоагенту на нейтральний холодоагент до озону і з низьким потенціалом глобального потепління. Мета даної роботи – розроблення й демонстрація багатоваріантного підходу при аналізі зазначеної проблеми – заміни холодоагентів HCFC та HFC, на холодоагенти групи A2L із потенціалом глобального потепління нижче 500. Особлива увага приділена потенціалу підвищення продуктивності й енергоефективності холодильної системи. У статті розв'язано наступні задачі дослідження: визначено вплив заміни холодоагенту на роботу основних елементів системи; запропоновано засоби і методи підвищення холодопродуктивності холодильної системи при заміні холодоагенту; розроблено методи підвищення енергетичної ефективності холодильної системи. Визначено основні змінення у роботі холодильної машини при заміні на холодоагент групи A2L, а саме встановлено, що при цьому змінюються показники компресора, потребується заміна мастила, необхідно враховувати вплив температурного глайду, а також змінення у роботі конденсатора та випарнику. Для підвищення холодопродуктивності холодильної машини запропоновано наступні засоби і методи: підбір холодоагенту, що може забезпечити необхідну холодопродуктивність; підвищення продуктивності компресора або частотним регулюванням, або встановленням додаткового компресора; мінімізація витрат тиску в трубопроводах гарячої пари і всмоктування; зменшення градієнту температур на конденсаторі й випарнику; зниження температури повітря на вході в конденсатор адіабатним охолодженням; додаткове переохолодження рідкого холодоагенту; оптимізація режимів роботи установки. Встановлено найбільш ефективний метод – зменшення перепаду температур між температурами конденсації та кипіння у найбільшій кількості годин річного циклу.

***Ключові слова**: холодоагент, температурний глайд, холодопродуктивність, потенціалом глобального потепління, адіабатне охолодження.*

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