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Prospects for the creation of compact refrigeratingma chines based on low-speedreciprocating machines

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Annotation
Currently, refrigeratingmachineswith two-stagecompression are most oftenusedtoobtaintemperaturesbelowminus400C.Suchcompressordesigns are cumbersomeandtechnologicallycomplex. Atthemoment, there is an alternative to multistagecompression– the use of low-speedcompressorscapable of compressinggastohighpressuresinonestageatacceptabledischargetemperatures. The use of a schemewith a single-stage low-speedcompressor makes it possible to increase the refrigerationcoefficientby12÷20%.At the same time, there isnoneed to install a heat exchanger- capacitor,whichreduces the weightanddimensions of the entireinstallationon20%.As can be seenfrom the resultsobtained, it is possible to obtain the liquidphase of the refrigerantin the workingchamber of the compressor.For low-speedcompressors, the presence of liquid is notscaryand does notcausehydraulicshocks.However, the study of the condensationprocess of the workingfluidin the compressorrequiresadditionalresearchandmustbeconfirmedexperimentally.

Keywords: *refrigeratingmachine, low-speedcompressor,workingfluidR744,reduction of weight and size parameters,mathematicalmodel,condenser,experimentalstudies*

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I. Introduction

Reciprocatingcompressorsarecurrentlythemostapplicabletypes of compressorsinsmallrefrigeratingmachines[1, 2].Inrefrigerationtechnology, it is oftennecessary to obtaintemperaturesin the rangeminus 40÷60 °C.Suchtemperaturescannotbeobtainedinrefrigeratingmachineswith a single-stagereciprocatingcompressor[1-3]. The reason for thisis the increasedratio of the condensationpressure to the boiling pressure of the refrigerant(P_c/P_e),which is traditionallylimited to a value of 8forrefrigeratingmachines[4].

The use of reciprocatingmachineswith an increasedratio of dischargepressure to suction pressure in the theory of reciprocatingcompressorshasits own rangesbasedongenerally accepted factorsleadingto the transitiontomultistagecompression. The mostsignificantofthesefactors(causes)is an increase in temperaturewith an increase in pressure. The value of the maximumtemperature is limited by safetyrequirements,ensuring that there is nooilignitionanddeformation of the workingchamberparts. The second most importantis the decrease in productivityassociatedwith the presence of harmfulspacein the reciprocatingcompressor. The thirdfactoris the increase in pistonforces,whichleadsto an over-dimensionality of the movementmechanism. The fourthfactoris the possibility of reducingindicatorperformancewhenimplementing two-stagecompression.And the fifthof the mostsignificantfactorsis the decrease in indicatorefficiency.Alltheseissues are solvedbyimplementingsequentialcompressioninonestage,cooling the compressedgasandcompressionto the finalpressurein the secondstage, the diameter of which is smaller than the diameter of the firststage,whichleadstoequality of pistonforcesin the firstandsecondrows[5,6].

The problems that arisewhencompressingrefrigerantsatP_c/P_e>8can be solvedusing low-speed long-strokecompressors[7-9]. It should be notedthatthedesignminimizes the effect of deadvolumedue to the elongatedshape of the cylinderandincreasesthecycle time toseveralseconds,thisallows you to obtain a degree of pressureincreaseexceeding100÷120withacceptableperformanceindicators of the workflow.Figure1shows a model of such a compressor.

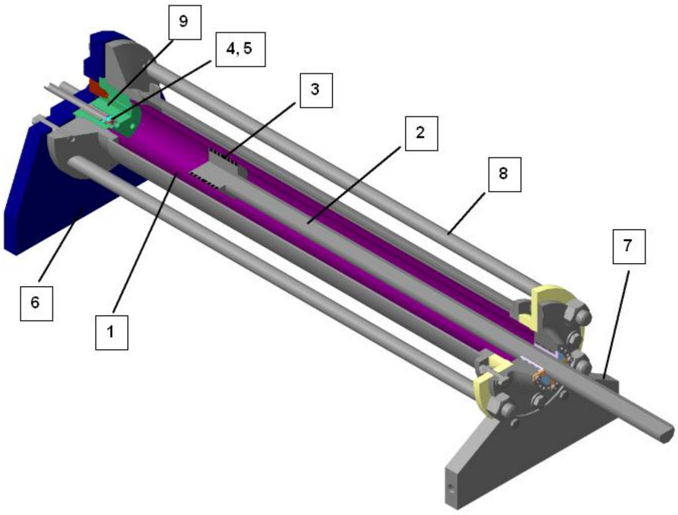

Figure1– Low-speedcompressormodel

Toanalyze the possiblereplacement of a refrigeratingmachinewith a two-stagecompressor with amachinewith a low-speedcompressor, we take as abasis the well-knownscheme of a two-stagerefrigeratingmachinewithdoublethrottlingFigure2.The use of a low-speedcompressor will allow, underthesameconditions, to implement a single-stagecompressionschemeFigure2.

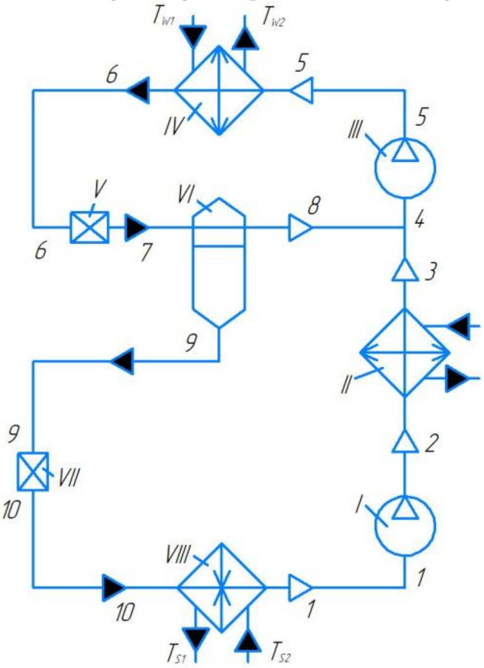

Figure2–Diagram of a two-stagerefrigerationunitwithdoublethrottling:
1-2Compressionin the firststage;2-3Intermediatecooling;4-5Compressionin the secondstage;5-6Condensation;6-7Throttling;9-10Throttling;10-1Evaporation

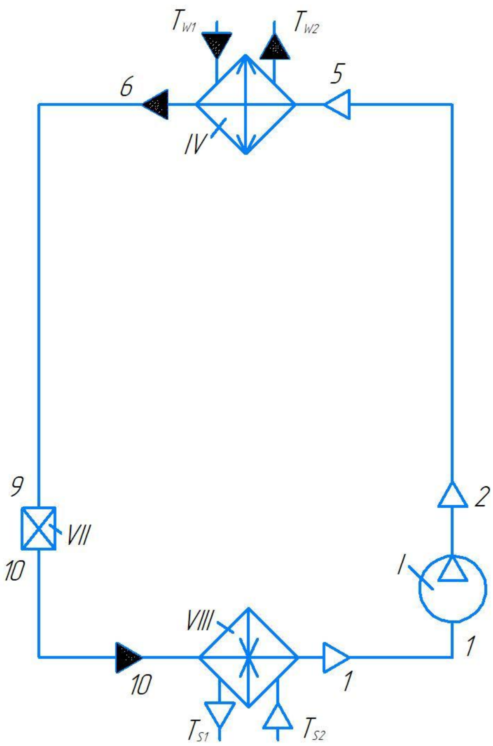

Figure3–Diagram of a refrigerationunitwith a low-speedcompressor: 1-2Compression;5-6Condensation;9-10Throttling;10-1Evaporation

A preliminaryanalysis of the installationsshowninFigures2.3suggestsasignificantsimplification of the circuitusing a low-speedcompressorwith the possibility of significantlyreducing the capacitororexcluding it from the circuit.
Inmodernsmallrefrigeratingmachines, for example,insuchlow-temperature machines based onCOPELANDcompressors(Germany), condenserunitsaccount for 15-25% of theweight and size parameters of the entireinstallation(Table1).

Table1–Characteristics of smallrefrigeratingmachines

ModelCondenser/Compressors	Cooling capacity (W)att_e=minus 70°C,t_c=plus32°C	PowerconsumptionH.M.(W)	Refrigerant
MDE114-4/AK-D4SL-150X/D4SL-1500	6050	17160	
MDE114-4/AK-D4SL-200X/D4ST-2000	6690	18160	
MDE123-4/AK-D6ST-320X/D4ST-3200	10100	29640	R22, R23

The presenteddataindicatetheurgency of the problem of reducing the loadon the capacitor.Basedon the experimentalstudiespresentedbelow, a mathematicalmodel of a low-speedrefrigerationcompressor will becreated.

The object of the study
Carbondioxide(R744) is widelyusedin the foodindustryforfreezingproducts[13], is environmentallysafeandhas a lowcriticaltemperature, is compatiblewithalmostallstructuralmaterials,butcannotcondenseattemperaturesaboveplus31 °C.The low-speedcompressor in question ishermeticallysealed and grease-free. A workingchamberwith a diameter of 0,05mand a pistonstroke of 0,5m,theworkingprocess time is 2-4seconds.
The stagehasexternalwatercooling.Initialconditions:suctiontemperature293K;suctionpressure0,5MPa, the degree of pressureincreaseis100. The refrigerantisR744.Basedon the experimentalstudiespresentedbelow, a mathematicalmodel of a low-speedrefrigerationcompressor will becreated.

Experimentalresearchmethodology
An experimentalstudy of the workingprocesses of thecompressorstage under consideration involvesmeasuringitsactualperformanceandindicatorpower, the averagedischargetemperature,aswell as the

instantaneousparameters of the compressibleworkingfluid.Theseresults will allow us to determinetheefficiency of the workingprocessand the heat transfer coefficienton the innersurface of the workingchamber.

An experimentalstandwith a linear(hydraulic)drive has been developedforconductingexperimentalstudies. The generalview of the experimentalstandfrom the studiedstage of the reciprocatingcompressor is also represented by asuctioncylinderFig.3. The linearactuatorinthisschemeis a hydraulicdrive. The measuringcircuit is showninFig.4. The operation of the stand is carried out as follows: the piston1isdriventhrough the rod2from the rod of the hydraulic cylinder,whichinturn is drivenfrom the accumulatorstation.Since the developedstageis a stagewithoutlubrication, the sealingcuffs3mountedon the piston1 are madeof self-lubricatingmaterialbased on PTFE.Carbondioxide is suppliedfrom the cylinder10, the pressure of the suppliedgas is regulated by a reducer9.

Datafrom the temperaturesensor and pressure sensor are transmitted to thedigitaloscilloscope7through an amplifier6. The gasflow rate at the outlet is measured by a flow meter8, the signalfromwhich is alsotransmitted to thedigitaloscilloscope7through an amplifier6.

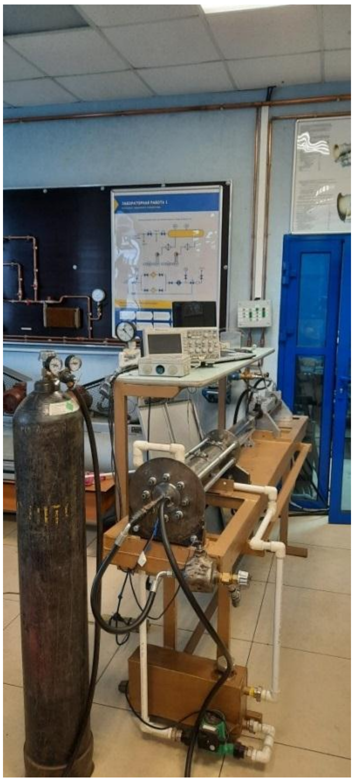

Figure4– An experimentalstandwith a slow-moving long-strokestage

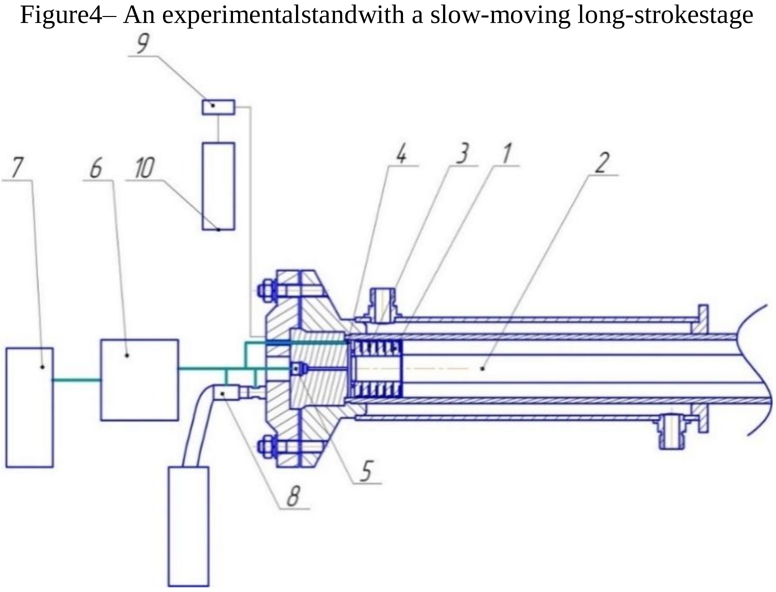

Figure5–Measuringcircuit of the stand

A Honeywell AWM720P1 sensor was used to measure the flow rate. The advantage of this sensor is the digital output signal, which allows it to be connected to modern PCs and oscilloscopes, as well as the factory output characteristic for various gases. The error of this sensor is 2%.Sensorsbased on a thermistortypeST1-18 Awere usedtomeasure the temperature of compressedair[6,7]. The datafrom the temperaturesensor is transmitted to thedigitaloscilloscopevia an amplifier.

Waveforms of instantaneousparameterssuchastemperatureandpressurewereobtained. An example of waveformsis shown inFigure6. The instrumenterror of pressuremeasurementis shown below. Let's determine the generalerror of the temperaturesensorbased on a bead thermistor[8-10].

$$\delta_T = \sqrt{\delta_{osc}{}^2 + \delta_t{}^2 + \delta_V{}^2 + \delta_F{}^2}\,, \tag{1}$$

where δ_{osc}–the relativeerror of the oscilloscope,0,05%;

δ_t– the error of the thermometer,determined by the error of the device,0,1%;

δ_V– the error of the voltmeter,determined by the error of the device,0,3%;

δ_F– the calculationerroraccording to the interpolatedformulaobtained,considering the nonlineardependence of voltageontemperature,1,5%.

Thus, the error in measuring the instantaneousairtemperaturein the workingchamber of the experimentalstage will be:

$$\delta_T = \sqrt{0{,}05^2 + 0{,}1^2 + 0{,}3^2 + 1{,}5^2} = 1{,}53\ \%.$$

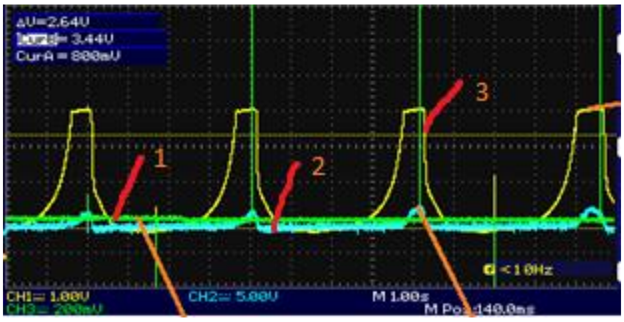

Figure6–Parameters of a low-speedcompressoratP$_{suc}$=0,5MPa;P$_{disc}$=6,0MPa:
1–flow rate,2–temperature,3–pressure

Siliconpressuresensors of the D16type were usedtomeasuregaspressure[11]. The temperaturemeasurementwasperformed by 4thermistorswith a negativetemperaturecoefficient of resistance.The actualgasflow rate at the injectionstage is determined by the AWM720P1 typeflowsensor8.Datafrom the temperaturesensor, pressuresensor and flow sensor are transmitted to thedigitaloscilloscope7via an amplifier6. The instrumenterror of measuringtemperature,pressureandflowis shown below.

Let's determine the instrumenterrorwhencalibrating the pressuresensor,determinedby the formula[9]:

$$\delta_{ps} = \sqrt{\delta_P{}^2 + \delta_{MPG}{}^2 + \delta_0{}^2}\,, \tag{2}$$

Where δ_P – relativeerror of the pressuresensor, 1,4 %;

δ_{MPG} – relativeerror of the model pressure gauge, 1,5 %;

δ_0 – the relativeerror of the oscilloscope,3 %.

$$\delta_{ps} = \sqrt{3^2 + 1{,}5^2 + 1{,}4^2} = 3{,}63\ \%.$$

Determination of integralcharacteristics

Conductingexperimentalstudies of the workingprocesses of pistonunitsimplies the determination of integralcharacteristicsthatcannot be determineddirectlybyanydevice.Theseparametersinclude the indicatorpower of the pistonstage, the feedratioand the isothermalindicatorefficiency.

Themethod of graphicaluse of the indicatordiagram has become the mostwidespreadindetermining the indicatorpower[12]. The experimentallyobtainedexpandedindicatordiagramusing a pressuresensor is subject to foldingtoobtain a collapseddiagramrepresenting a closedcycle. The area of the resultingdiagramcharacterizes the experimentalindicatorworkspent,andknowing the cycletime, the indicatorpower is obtained.

The isothermalindicatorefficiency is determined by the ratio of the value of the indicatorworkobtainedaccording to the experimentalgraph of the dependence of the instantaneouspressureonthevolume of the workingchamberto the operation of an idealisothermalcompressor[13]:

$$\eta = \frac{P_{suc} \cdot \overline{V}_h \cdot \ln(\frac{P_{disc}}{P_{suc}})}{L_{ind.}}. \tag{3}$$

The actualperformance of the stagecanbemeasured by the selectedflowsensorandrecalculatedforsuctionconditions – V_e.Thus, the experimentalfeedcoefficient is definedas the ratio of the actualperformanceto the performance of an ideal low-speedstageunderidenticalsuctionconditions,operatinganddesignparameters[13]:

$$\lambda = \frac{V_e}{V_h}. \tag{4}$$

The conductedexperimentalstudies made it possible to create a refinedmethodology for the numericalcalculation of a low-speedcompressorwhenoperatingon carbon dioxide.

Numericalcalculationmethod

The mainequation that implements the relationshipbetween the mainprocessesoccurringin the workingchamberis the equation of the firstlaw of thermodynamics. Let's determine the change in the internalenergy of the system[14,15]:

$$dU = dA \pm dQ \pm (dm \cdot i), \tag{5}$$

wheredA– the workperformed by the refrigerantor the workperformedon the refrigerant,J;
dQ– the heatwithdrawnfrom the gasortransferred to itfrom the walls of the workingchamber,J;
complex$dm \cdot i$– we characterize the energyenteringorremovingfrom the system by gasflows(the product of the mass of the gasanditsspecificenthalpy),J.
It should be notedthathereandfurther the equations are writtenfor a certainsmallperiod of time,where the quantities includedin the equationshave a constantvalueforthisperiod of time.

$$dA = P_g \cdot dV, \tag{6}$$

whereP_g– the gaspressure,Pa;
dV– the change in volume(the product of the area of the pistonbyitsspeed of movement),m^3.
The refrigerantpressure is determinedfrom the equation of state:

$$P_\Gamma = \frac{z(P) \cdot m \cdot R \cdot T_g}{V_g}, \tag{7}$$

where$m$– the currentmass of the workingfluidin the system,kg;
$z(P)$– a function of the change in the compressibilitycoefficient of theworkingfluid in question;
R– the gasconstant,J/K;
V_g–volume of gas,m^3;
T_g– the temperature, there is a function of the energy of the system-U,K.

$$T_\Gamma = \frac{U}{m \cdot C_V}, \tag{8}$$

whereC_V– the specificmassheat capacity in the isochoricprocess,J/(kg·K).

$$dQ = \boldsymbol{\alpha}_{avg} \cdot (T_g - T_w) \cdot f, \tag{9}$$

where $\boldsymbol{\alpha}_{avg}$ – the heat transfer coefficient,determinedexperimentally,$W/m^2 \cdot K$;
T_w–walltemperature,K;
f– the heat exchange area,m^2.
Heat transfer coefficient[169]:

$$\alpha = \lambda \cdot (\rho/\mu)^x W^x D^{1-x}_{egu}, \tag{10}$$

whereλ,μ,$D_{egu}$and$W$– the currentvalues, respectively, of the coefficient of thermal conductivity,dynamicviscosity,equivalentcylinderdiameterandconditionalgasvelocityin the workingchamber;
x– the empiricalcoefficient,

$$dm = \alpha \cdot \varepsilon \cdot A \cdot \sqrt{2 \cdot \rho \cdot \Delta P}, \tag{11}$$

whereα– the consumptioncoefficient;

A–expirationarea,m^2;
ΔP–pressure drop,Pa;
p– the density,kg/m^3.
Equation(11) is usedtodeterminemassflowsboththroughvalves(openorpartiallyopen), thenthevalue of areaAincludes the variableliftingheight of the valveplate(h)andtodeterminemassflowsthroughleaks.In the case of determining the massflowsthrough the valvegaps, the values of the conditionalgapsobtainedexperimentally are used.Whendeterminingleaksthrough a cylinder pistonseal, the areaAis the product of the perimeter of the cylinder pistonseal by thevalue of the conditionalgapin the cylinder pistonseal.
Todetermine the coordinate of the locking device (h), the followingequation is solved:

$$m_{pl} \cdot \ddot{h} = \vec{F}_g + \vec{F}_{sf} + \vec{F}_{ff} + \vec{G} + \vec{F}_{ef}, \qquad (12)$$

where $\vec{F}_g$ – the totalforceactingon the platefrom the gasside,N;

$\vec{F}_{sf}$ – the spring's elasticforce,N;

$\vec{F}_{ff}$ – the frictionforce of the gas,N;

$\vec{F}_{ef}$ – the elasticforce of an elastomericelement;

$\vec{G}$ – the weight of the locking device.
The simplifyingassumptionsadoptedtocreatethiscalculationmethodcorrespond to those generally acceptedforthisclass of mathematicalmodels[16].
The results of numericalandfieldexperimentswerecompared. The results of the comparisonare shown inFig.7-10.

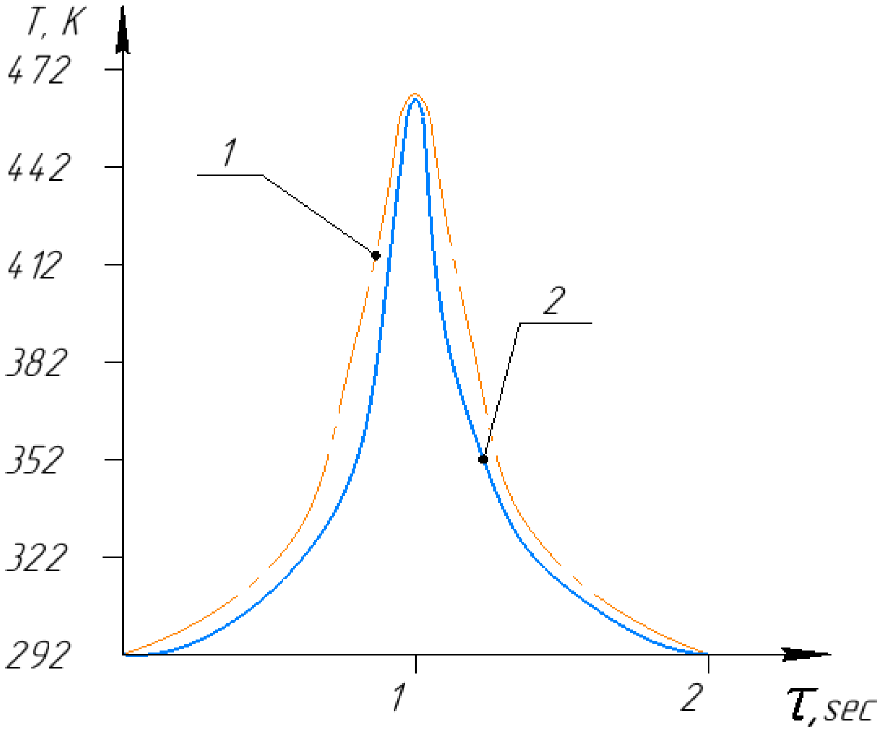

Figure7–Graph of the instantaneoustemperaturechangeduringthecycleat a pressure of 10MPa

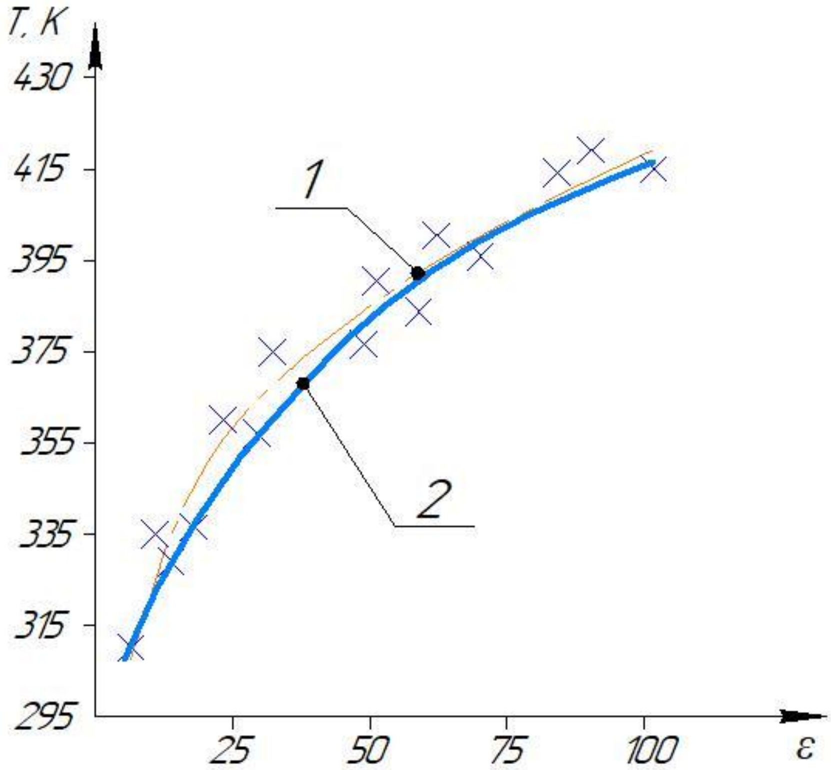

Figure8–Graph of the averagetemperaturechangeduringthecycleat a pressure of 10MPa

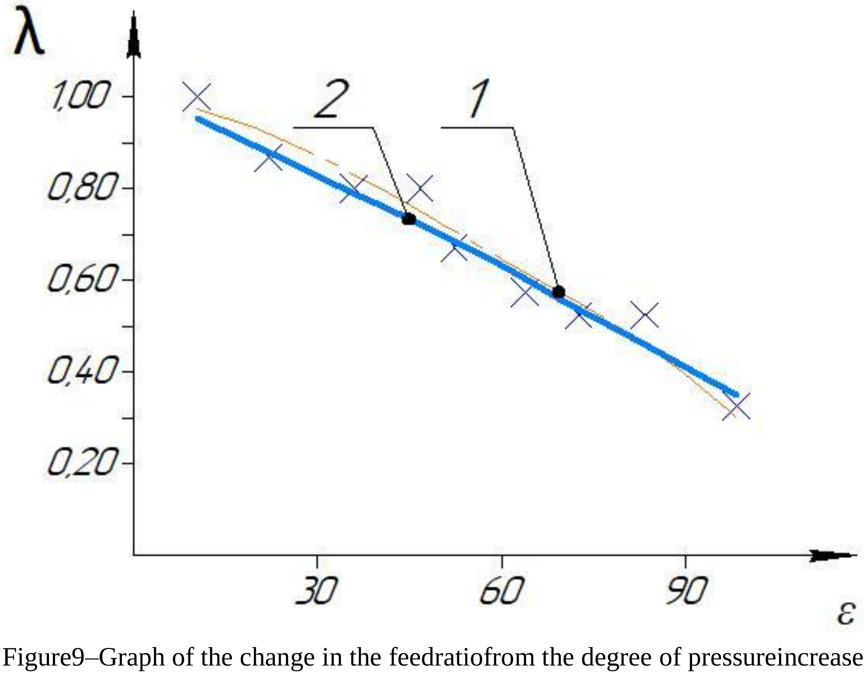

Figure9–Graph of the change in the feedratiofrom the degree of pressureincrease

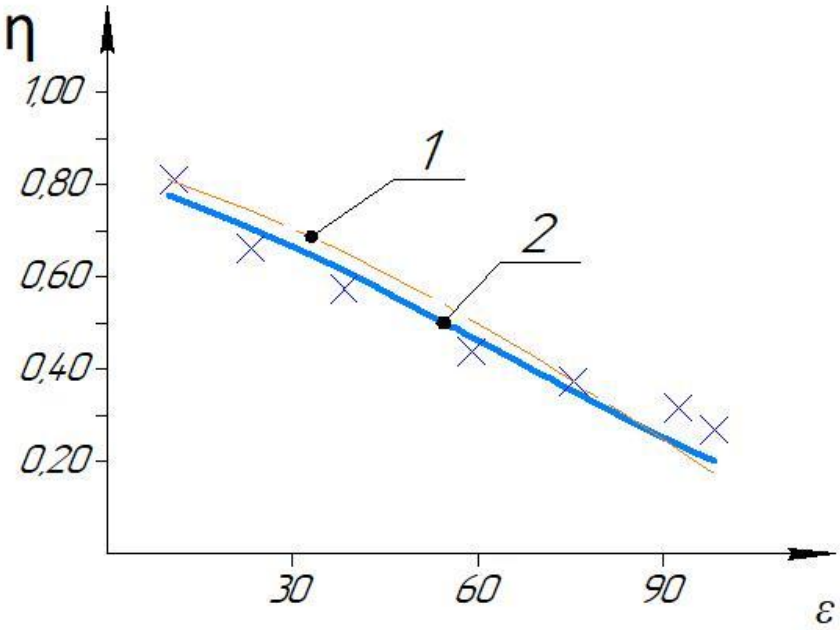

Figure10–Graph of the change in the indicatorisothermalefficiencyfrom the degree of pressureincrease

Basedontheexperimentaldata, equation(10)takes the form:

$$\alpha = \lambda_g \cdot (\rho/\mu)^{0,8} W^{0,8} D^{0,2}{}_{equ} \quad (13)$$

Research results

Let's consider the cost of indicatorworkoncompressionwhenusing a two-stagecircuitand a single-stagecircuitwith a low-speedcompressor.Notethat, according to the well-knownmethod, the compressionpolytropecoefficientfor a low-speedcompressor is determinedaccording to the dataof[17].Figures11,12showindicatordiagrams of two-stagecompressionand single-stage compression using a low-speedcompressor.

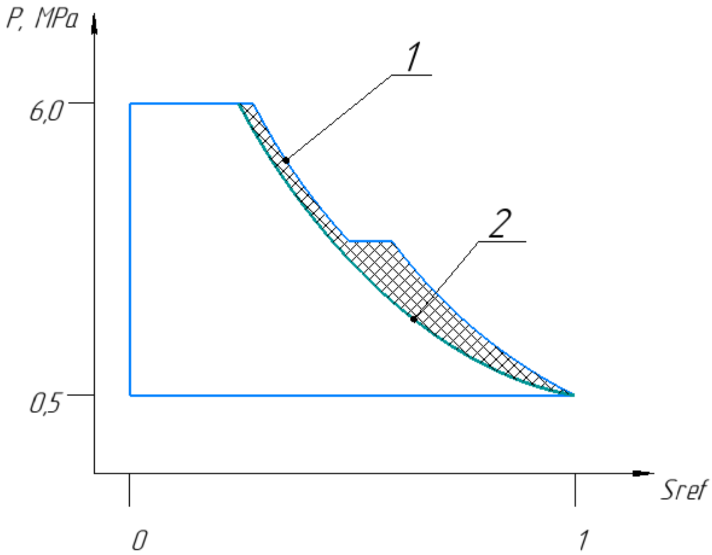

Figure11–Indicatordiagrams of two-stagecompressionand single-stage compression using a low-speedcompressor(cycletime2s)

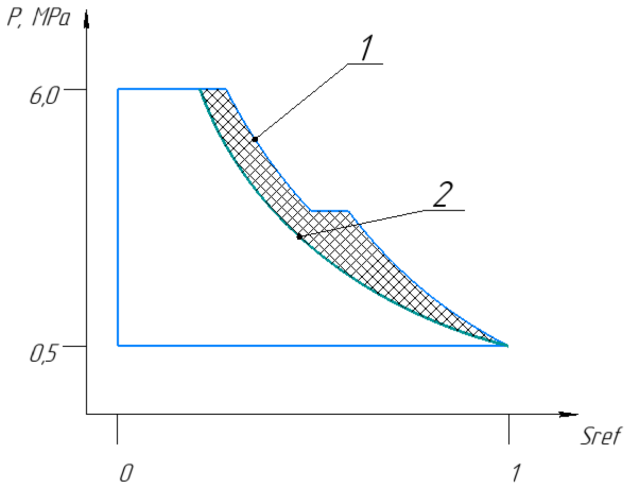

Figure12–Indicatordiagrams of two-stagecompressionand single-stage compression using a low-speedcompressor(cycletime4s)

The analysis of the presentedgraphssuggestsadecrease in the indicatoroperation of compressionwhenusing a low-speedcompressor.Line1corresponds to compressionin a two-stagemachine,line2 corresponds to compressionin a low-speed single-stage machine.Inthis case, the shadedarea is proportional to the decrease in compressionwork. It can be seenthat the compressionline of a low-speedcompressor is located to the leftrelativeto the compressionline of a two-stagecompressor,thatis, compressionoccursin a low-speedcompressorin a quasi-isothermalmode. The equivalentpolytropyindicatorisapproximately1.1.For a modewith a cycletime of 4 seconds, the powerconsumption is reduced by about15%relativeto the two-stagecompression,andfor a cycletime of 2 seconds –about10%.
Thus, the exclusionof the condenser from the circuitallows, alongwith an increase in the efficiency of the refrigeratingmachine(the refrigeratingcoefficientincreasesby10÷15%), to reduce the weight and dimensions of the refrigeratingmachineby20%.

II. Conclusions

Theoreticalstudies have shown the possibility of replacing two-stagerefrigeratingmachines with single-stage ones using low-speedcompressors. The lowrate of compressionpolytropein low-speedmachines makes it possible to realizecompressionclosetoisothermal.Due to this, the temperatureat the end of compression is significantlylower,whichallows the use of single-stagemachinesup to compression ratios of 100andabove. The use of such a schemeallows to increase the refrigerationcoefficientby12÷20%.At the same time, there isnoneed to install a condenser heat exchanger,whichreduces the weightandoverall dimensions of the entireinstallationby20%.As can be seenfrom the resultsobtained, it is possible to obtain the liquidphase of the refrigerantin the workingchamber of the compressor.For low-speedcompressors, the presence of liquid is notscaryand does notcausehydraulicshocks.However, the study of the condensationprocess of the workingfluidin the compressorrequiresadditionalresearchandmustbeconfirmedexperimentally.Theauthors of thisarticle are dealing with this issue.

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