TEKA. COMMISSION OF MOTORIZATION AND ENERGETICS IN AGRICULTURE – 2012, Vol. 12, No.4, 101-107

Method of a choice of parameters of the air cooling machine of a cascade pressure exchange

Alexander Krajniuk, Yuri Starcheous, Alexander Danileychenko, Maksim Bryantsev

Volodymyr Dahl East-Ukrainian National University, Lugansk, Ukraine

Summary. The algorithm of calculation of the basic dimensional parameters of compound units of the system realizing running cycle of the air refrigerating machine of the cascade pressure exchange of set cooling capacity and object cooling depth is resulted. The technique of an estimation of expenses of thermal and mechanical energy on realization of a cycle of the air refrigerating machine is presented.
Key words: cascade pressure exchanger, mass-exchange, air cooling machine, recuperation, refrigerating capacity.

INTRODUCTION

Along with search of ways of decrease heat expenditure cold production more and more the attention is given to ecological safety of a refrigerating machinery. A serious environmental problem of planetary scale is the exhaustion of an ozone layer of an aerosphere, substantially in an influence kind galogen hydrocarbons (freons), widely used as a working body in a refrigerating machinery of compressor type.

According to opinion of scientists at conservation of dynamics of technogenic influence on an atmosphere predicted rates thining an ozone layer will make 7 % in 60 years [Rowland 1994, Anderson 2002].

It is necessary to notice, that operation of a refrigerating machinery on transport is interfaced to the raised probability of leaks of a coolant in view of high vibrating loadings and the limited possibility of the timely control of tightness of system in vehicle movement.

OBJECTS AND PROBLEMS

Now interest to air refrigerating machines (ARM), having a more potential of low temperature coolings without use ozone destroy coolants renews. The refrigerating factor of air installations at rather small relations of ambient temperature and cooled object concedes to indicator steam compression installations, however in the field of deep cooling the running cycle air ARM with regeneration realises higher power efficiency.

At the same time, approved ARM on base blade units of compression and expansion are expensive in manufacturing and in view of high frequency of rotation of rotors turbine-compressor unit have the limited resource, demand high level of maintenance service. Insufficiently high power efficiency of turbine-compressor ARM is caused by the limited possibility of the further efficiency increase of working processes of blade machines, reached by considerable complication of a design of the last.

Possibility of reduction in price ARM contacts use as the detander-compressor of a wave pressure exchanger (WPE) similar on a design to the unit of air supply of system of pressurization ICE “Comprex” [Ersmambetov 2000]. In a rotor of WPE in the course of a direct exchange of energy between compressing and compressed environment along with cooling of an air stream compression to 25 … 30 % of a coolant is carried out. The coolant most part is compressed in a separate, basic step of the compressor resulted from an external source of mechanical energy. Simplicity and reliability of a design of the device concerns advantages of the wave detander-compressor, and also lower concerning a turbine unit frequency of rotation of a rotor (6500 … 10000 minutes^{-1}).

However, wave character of an exchange of energy predetermines high sensitivity of working process of WPE to a picture of interacting of primary waves with forward edges of gas-distribution ports. The deviation of an operating mode of WPE from settlement conditions on frequency of rotation of a rotor, pressure and temperature of working environments is accompanied by sharp deterioration of indicator of its work as owing to a mismatch of phases of movement of primary waves, and owing to incompleteness of replacement of compressed air from rotor cells. The increase in a share of the compressed air which has remained in a cell at the moment of its dissociation with port of a high pressure, caused almost proportional decrease of efficiency, to similarly negative effect of so-called "dead" volume in the piston compressor. Besides, inevitable dissipation the phenomena in processes of formation and interaction of strong shock waves limit of efficiency of the best samples of wave pressure exchangers on settlement modes values 0,55 … 0,56 [Krajniuk 2000].

Considerably much reserve of perfection of operational indicator of ARM is connected with use in quality detander, and also as the basic compressor of essentially new version heat exchange devices - a cascade pressure exchanger (CPE) of prof. A.I. Krajniuk [Krajniuk 2004]. On fig. 1 the modified scheme of air refrigerator is shown.

In given article attempt to reflect the basic laws of work ARM CPE with probably more simple engineering dependences within the limits of a statement of algorithm of a choice of dimensional parameters and definition of effective indicators of work of system is undertaken. Remarks to a choice of initial parameters of system of the settlement scheme.

The demanded temperature mode in the chamber of cooled object and corresponding cooling capacity of ARM CPE can be realized with various combination of temperature T_3 and consumption G_3 of an air coolant submitted to the chamber. In turn decrease in temperature forced ARM CPE of an air coolant can be reached increase in throughput (dimensions) of the CPE unit of a compensatory step (CPEII) or increase of the maximum temperature of heating of air T_5 in a contour of a high pressure a source of a supply of warmth. The combination of noted regime parameters influences a parity of expenses of the thermal and mechanical (electric) energy spent for realization of a refrigerating cycle. For example,

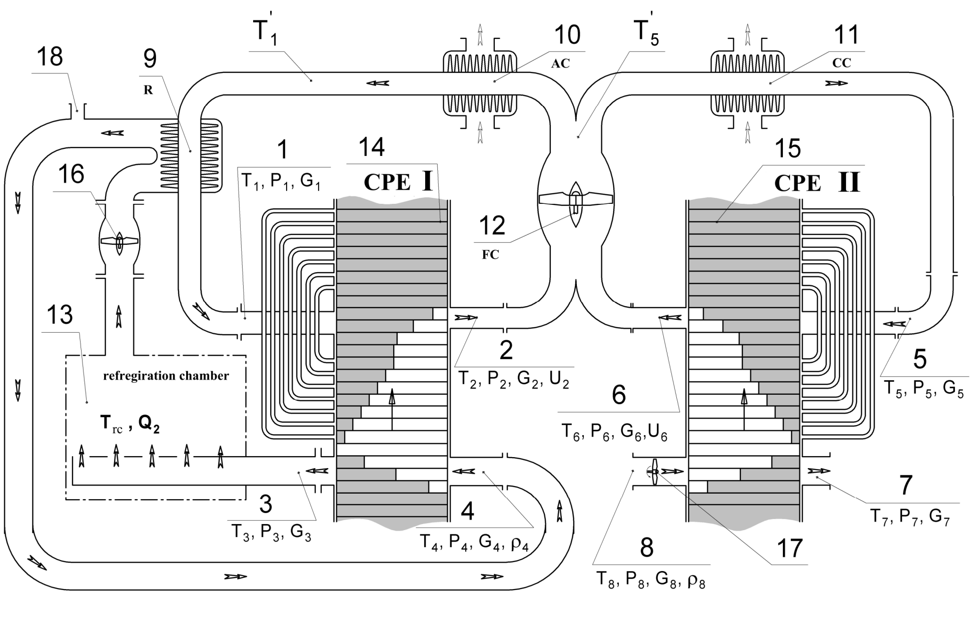

Fig. 1. The settlement scheme of an air refrigerator with a cascade exchanger of pressure:
1, 2, 5, 6 - ports of supply and tap high pressure (HPS and HPT) accordingly CPEI and CPEII; 3, 4, 7, 8 - ports of low tap and supply pressure (LPT and LPS) accordingly CPEI and CPEII; 9 - regenerative heat exchanger; 10 - air cooler; 11 - warmth supply source; 12 - circulating fan; 13 - refrigerating chamber; 14, 15 - cascade pressure exchangers CPE1 and CPE2; 16, 17 - blowing-off fans, 18 - branch pipe.

decrease of temperature T_3 (increase of expansion degree π_p in CPEI) at the expense of increase T_5 or increase of CPEII dimensions is accompanied by increase in expenses of thermal energy in a source of a supply of warmth at simultaneous reduction of expenses of mechanical energy by a drive of the circulating fan. Thus, the choice of rational parameters ARM CPE depends from parity of the developed prices on electric and thermal energy and, finally, is reduced to search of the regime conditions providing the minimum sum of cost equivalents thermal and mechanical energies, spent for realization of running cycle ARM CPE.

The algorithm of calculation of the basic dimensional parameters of compound units of the system realizing running cycle ARM CPE with following set initial data is more low resulted: 1 – cooling capacity Q_2, 2 - temperature of cooled object T_{rc}, 3 - the minimum temperature of cycle T_3 (temperature of air arriving in the refrigerating chamber), 4 - the maximum temperature of cycle T_5.

As the basic assumptions of calculation one-dimensionality of a current of a working body in flowing elements of system, absence of leaks of a working body through backlashes in mobile interfaces and heat exchange with walls, absence of a zone of hashing of compressing and compressed environments, and also full replacement of the compressed environment from cells of rotors and a full purge of cells a fresh charge is accepted.

Completeness of replacement and purge is estimated by corresponding factors. The purge factor in CPEI and CPEII ($\psi_{\text{АПI}}$, $\psi_{\text{АПII}}$) represents relation of consumptions of air through ports of a supply of low pressure to mass throughput of a rotor

$$\psi_{LPI} = \frac{G_4}{g_I},\ \psi_{LPII} = \frac{G_8}{g_{II}}. \qquad (1)$$

Mass throughput accordingly rotors CPEI and CPEII is defined as

$$g_I = \frac{V_{RI} \cdot n_{RI} \cdot \rho_4}{60},\ g_{II} = \frac{V_{RII} \cdot n_{RII} \cdot \rho_8}{60}, \qquad (2)$$

where: V_{RI}, V_{RII}, n_{RI}, n_{RII} - volumes of a flowing part and frequency of rotation of rotors accordingly CPEI and CPEII, ρ_4, ρ_8 - density of a working body in ports of a supply of low pressure accordingly CPEI and CPEII.

Similarly, the replacement factor accordingly in CPEI and CPEII (ψ_{HPI}, ψ_{HPII}) represents itself the consumption relation through ports of tap of a high pressure to mass throughput of a rotor

$$\psi_{HPI} = \frac{G_2}{g_I},\ \psi_{HPII} = \frac{G_6}{g_{II}}. \qquad (3)$$

The dimensions of rotor CPEI of the detander-compressor are from a condition of conformity of mass throughput of a rotor to the consumption of coolant in the chamber cooled object.

The coolant consumption, providing set cooling capacity of ARM, is calculated by the formula

$$G_3 = \frac{Q_2}{Cp_a \cdot (T_{rc} - T_3)}, \qquad (4)$$

where: Cp_a - a mass heat capacity of air.

Accepted above an assumption of completeness of replacement and a blowing ($\psi_{HPI}=1$, $\psi_{LPI}=1$) in essence suggest, that, on the one hand, all compressed air arriving in cells of a rotor through port 1, after expansion is taken away in the refrigerating chamber 13 through port 3 and, thus $G_1 = G_3$, on the other hand, all air charge arriving in cells of a rotor through port 4, after compression is taken away through port 2 $G_2 = G_4$.

In a kind noted easily also to show, that the consumption ratio in high pressure port can be expressed through a ratio of densities of air in a line of low pressure

$$\frac{G_1}{G_2} = \frac{\rho_{res1}}{\rho_4}. \qquad (5)$$

Here ρ_4 - air density in port 4 CPEI; ρ_{res1} - density of the rotor which has extended in cells of air, during the moment previous their connecting with port of low pressure.

$$\rho_4 = \frac{P_4}{R \cdot T_4},\ \rho_{res1} = \frac{P_{res1}}{R \cdot T_{res1}}, \qquad (6)$$

where: P_4 - the pressure created by the blowing fan 16, T_4 - air temperature in port 4, P_{res1}, T_{res1} - pressure and temperature extended in a cell rotor CPEI of air, R - a gas constant of air.

In turn air temperature in port 4 with ratio of water equivalents of flows in the regenerative heat exchanger 9 is calculated with formula:

$$T_4 = T_{rc} + \varepsilon(T_1^{'} - T_{rc}), \qquad (7)$$

where: ε - regenerator efficiency, for plate-fin recuperators reaches values 0,8 … 0,85; $T_1^{'}$ - air temperature in a line of a high pressure to a regenerator 9.

Then, air temperature in port 1 is defined from the equation of heat balance of a regenerator 9

$$T_1 = T_1^{'} - \frac{G_3 C_p (T_4 - T_{rc})}{G_1 C_p}. \qquad (8)$$

Residual pressure P_{res} in a cell of rotor CPEI in moment previous its connection to port of low pressure, can be found from resulted below the analytical dependence received by authors as a result is settlement-experimental researches of row of CPE models [2, 3]:

$$P_{res} = P_4 + \frac{P_2 - P_4}{z_к - 1}, \qquad (9)$$

where: $z_{к1}$ - number of mass-exchange channels in CPEI.

Pressure in port 2 tap of high pressure P_2 depends on difference of pressure between ports HPS and HPT ΔP_{1-2}, created with circulating fan 12.

$$P_2 = P_1 - \Delta P_{1-2}. \qquad (10)$$

The choice of value ΔP_{1-2} is corresponded with compromise search between overall dimensions of exchanger and expenses of power for a drive of the fan 12 which are the basic consumer of mechanical energy ARM CPE.

Pressure in port 1 is determined on the equation polytropic

$$P_1 = P_3 \left(\frac{T_1}{T_3}\right)^{\frac{n_{e1}}{n_{e1} - 1}}, \qquad (11)$$

where: n_{e1} - an indicator of a polytropical of expansion of air in CPEI.

Temperature residual coolant T_{res} in a rotor cell in moment previous its connecting with a port of low pressure

$$T_{res} = T_1 \left(\frac{P_{res}}{P_1}\right)^{\frac{n_{e1} - 1}{n_{e1}}}. \qquad (12)$$

The parameters of working environments found thus allow according to the equations (4), (5) and (6) to define air consumption in a port 2 CPEI

$$G_2 = G_3 \frac{P_1}{P_{res}}. \qquad (13)$$

The area of section of port through passage 3 HPT

$$F_2 = \frac{G_2}{\rho_4 \cdot \left(\frac{P_2}{P_4}\right)^{\frac{1}{n_{c1}}} K_{\rho 2} \cdot U_2 \cdot (1 - \delta_{c1})}, \qquad (14)$$

where: n_{c1} - an indicator of a polytropical of compression in CPEI; U_2 – mean speed of flow in port 2; δ_{c1} - thickness of crosspieces of rotor CPEI; $к_{p2}$ = 0,96-0,97 - factor unoredimentions, considering change of speed of a flow on section of port 2 owing to throwing compressing gas.

In works [2, 4] various ways of calculation of mean speed of flow U_2 in section of port 2 are shown. However, the analysis of experimental researches of CPE shows, that at a stage of a preliminary choice of parameters for rather small relations of pressure in ports 1 and 2 (P_1/P_2=1,01 … 1,03) possibility of equation Bernoulli with empirical factor of decrease speed ($к_{u2}$ = 0,3 … 0,4), considering various kinds of losses (the distributed friction along rotor cells, tear-off sections in boundary conditions in the cells, flooding of jet unsteady flow, etc.) is possible.

$$U_2 = K_{U2} \cdot \sqrt{2 \cdot R \cdot T_1 \cdot \frac{к}{к - 1} \cdot \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{к-1}{к}}\right]}. \qquad (15)$$

At cell connection to high pressure port (HPS and HPT) to process of replacement of air in port HPT precedes compressing up air in the considered cell, preliminary compressed as a result of cascade mas-exchange. Therefore the area of section of a port through passage 1 (F_1) must exceed value F_2 on 10-15 %.

The area of a face surface of a rotor

$$F_{\Sigma R1} = F_2 \cdot \frac{360}{\varphi_2}, \qquad (16)$$

where: φ_2 - a corner of disclosing of port 2.

The length and frequency of rotation of rotor CPE1 are corresponded by the parity as a matter of fact representing a condition of full replacement of a fresh charge from a cell of a rotor:

$$L_1 \cdot n_1 = \frac{\left(\frac{P_2}{P_4}\right)^{\frac{1}{n_{c1}}} \cdot U_2 \cdot \varphi_2}{6}. \quad (17)$$

Pressure of circulating fan should be sufficient for overcoming of hydraulic resistance of elements of a contour of high pressure CPEI (air cooler - ΔP_{AC}, a regenerator - ΔP_R, connecting pipelines - ΔP_{CPI}) and creations of nominal difference of pressure between ports 1 and 2.

$$\Delta P_{CF} = \Delta P_{AC} + \Delta P_R + \Delta P_{CPI} + \Delta P_{1-2}. \quad (18)$$

Passing to definition of design data CPEII, we will notice, that taking into account hydraulic resistance of elements of a contour of a high pressure of a compensatory step difference of pressure between port 5 and 6 CPEII is as

$$\Delta P_{5-6} = \Delta P_{CF} - \Delta P_Q - \Delta P_{CPII}. \quad (19)$$

Where: ΔP_Q - hydraulic resistance of a source of a supply of warmth, ΔP_{CPII} - hydraulic resistance of connecting pipelines.

Condition of coordinated work CPEI and CPEII is completion by a compensatory step of deficiency of the consumption in a contour of high pressure CPEI:

$$G_1 - G_2 = G_6 - G_5. \quad (20)$$

By analogy to calculation CPEI for CPEII it is possible to write down

$$\frac{G_5}{G_6} = \frac{\rho_{res2}}{\rho_8}, \quad (21)$$

where: $\rho_8 = \rho_0$ - air density in port 8 CPEII, ρ_{res2} - density of air which has extended in rotor CPEII in the moment previous connection of a cell with port of low pressure.

$$\rho_{res2} = \frac{P_{res2}}{R \cdot T_{res2}}. \quad (22)$$

The temperature of residual gases in a cell of rotor CPEII T_{res2} is defined on the polytropic equation:

$$T_{res2} = T_5 \cdot \left(\frac{P_{res2}}{P_5}\right)^{\frac{n_{e2}-1}{n_{e2}}}, \quad (23)$$

where: n_{e2} - an indicator of a polytrack of expansion in CPEII, $P_5 = P_6 - \Delta P_{5-6}$ - pressure in port 5 CPEII.

Residual pressure in cell P_{res2} is defined under the empirical formula:

$$P_{res2} = P_8 + \frac{P_6 - P_8}{Z_{K2} + 2}, \quad (24)$$

where: $P_6 = P_2$ - pressure in port 6 CPEII, P_8 - pressure 17 air forced by the blowing fan, $Z_{\kappa 2}$ - quantity mass-exchange channels in CPEII.

According to the equations (20) and (21) expense G_6 and G_5 are defined as air expenses in port of high pressure CPEII:

$$G_6 = \frac{(G_1 - G_2) \cdot \rho_8}{\rho_8 - \rho_{res2}}, \; G_5 = G_6 - G_1 + G_2. \quad (25)$$

Basic constructive and regime parameters CPEII, it is similar CPEI, are under following formulas:

- the area of section of a window through passage 6 HPT

$$F_6 = \frac{G_6}{\rho_8 \cdot \left(\frac{P_6}{P_8}\right)^{\frac{1}{n_{c2}}} \cdot K_{\rho 6} \cdot U_6 \cdot (1 - \delta_{c2})}; \quad (26)$$

$$U_6 = K_{G2} \cdot \sqrt{2 \cdot R \cdot T_5 \cdot \frac{\kappa}{\kappa - 1} \cdot \left[1 - \left(\frac{P_6}{P_5}\right)^{\frac{\kappa-1}{\kappa}}\right]}, \quad (27)$$

where: $K_{\rho 6}$ - factor un-one-dimensions, considering change of speed of flow on section of port 6 owing to throwing compressing gas;

- the area of a face surface of rotor CPEII

$$F_{\Sigma R2} = F_6 \cdot \frac{360}{\varphi_6}, \quad (28)$$

where: φ_6 - the corner of disclosing of port of tap of a high pressure 6, gets out according to resulted above recommendations.

The length of rotor CPEII and frequency of its rotation are connected by the equation

$$L_2 \cdot n_2 = \frac{\left(\frac{P_6}{P_8}\right)^{\frac{1}{n_{c2}}} \cdot U_6 \cdot \varphi_6}{6}. \quad (29)$$

Expenses of thermal energy for realization of working process ARM are defined by air heating in a source of a supply of warmth

$$Q_3 = G_5 \cdot Cp \cdot (T_5 - T_5'), \qquad (30)$$

where: T_5' - temperature of air mix behind the fan 12. In a running cycle of the elementary scheme ARM CPE without recycling the residual temperature extended in CPEII hot air are calculated off on the equation:

$$T_5' = \frac{T_6 \cdot G_6 + T_2 \cdot G_2}{G_6 + G_2} \cdot \left(\frac{P_6 + \Delta P_{CF}}{P_6} \right)^{\frac{\kappa - 1}{\kappa}}, \qquad (31)$$

where: T_6 and T_2 - temperature accordingly in port 6 CPEII and port 2 CPEI:

$$T_6 = T_8 \left(\frac{P_6}{P_8} \right)^{\frac{n_{c2} - 1}{n_{c2}}}, \quad T_2 = T_4 \left(\frac{P_2}{P_4} \right)^{\frac{n_{c1} - 1}{n_{c1}}}, \qquad (32)$$

here: n_{c2} - an indicator of a polytropic of compression in CPEII.

Quantity of the warmth which is taken away from a coolant in a cooler 10:

$$Q_1 = G_1 \cdot Cp \cdot (T_5' - T_1'). \qquad (33)$$

Expenses of mechanical energy for a drive of each of fans of ARM CPE consist of expenses for a drive of each of fans 12, 16 and 17, defined by adiabatic work of compression of corresponding flows of air with the fan efficiency

$$N_{fi} = G_i \cdot Cp_{ai} \cdot T_{beg}i \cdot \left[\left(\frac{P_{begi} + \Delta P_i}{P_{begi}} \right)^{\frac{k-1}{k}} - 1 \right] \cdot \frac{1}{\eta_{fi}}, \qquad (34)$$

where: G_i - the mass consumption of air through the corresponding fan, T_{begi} - temperature of a stream arriving in the fan, P_{begi} - pressure in front of the fan, ΔP_i - a pressure of the corresponding fan, η_{fi} - efficiency of the corresponding fan, k - an indicator of an adiabatic curve of compression.

CONCLUSIONS

The resulted algorithm allows to define the dimensions of each of cascade pressure exchangers as a part of the air refrigerator realizing demanded cooling capacity for accepted various conditions of the organization of a running cycle (T_5, T_3), and also estimations of expenses of thermal and mechanical energy on realization of cycle ARM CPE. The divergence of results of calculation with experimental researches of skilled installation with cooling capacity 10 kW at T_{rc} =-40°C on expenses of electric energy does not exceed 10 %, on thermal energy expenses - no more than 15 %.

Engineering availability and comprehensible accuracy of a considered technique predetermines possibility of its use as the tool of expansion of search of the regime conditions providing peak efficiency of work ARM.

REFERENCES

1. **Anderson, S.O., Sarma, K.M., 2002.:** Protection the Ozone Layer, Earthscan Publications Ltd, US.
2. **Ersmambetov V.S., Beleski A., 2000.:** US Patent №6138471, F25D 9/00, 31.10.2000. Device for air conditioning.
3. **Krajniuk A.I., Starcheous J.V., 2000.:** Systems of gas-dynamic supercharging. – Publ. EUSU. Lugansk., 224 p.
4. **Krajniuk A.I., Krajniuk A.A., 2004.:** The cascade pressure exchanger. Patent UA №77261A, International patent classification 7 F 02 B 33/00/ ,- №20040806986; it is declared 21.08.2004; it is published 15.11.2006, bulletin №11.
5. **Krajniuk A. 2010.:** Thermal compressor of cascade exchange by pressure. Problemy transportu. Miedzynarodowe czasopismo naukowe, Tom 5 zeszyt 1, Wydawnictwo politechniki slaskiej Gliwice, pp.15-22.
6. **Krajniuk A.I., Krajniuk A.A, Bryantsev M.A, 2008.:** Use of the principles of a cascade exchange by pressure in a running cycle of gas refrigerators / Internal combustion engine, NTU "HPI" Scientific and technical journal.-. Kharkov. - No. 1. - page-57-61.
7. **Krajniuk A.I., Krajniuk A.A, Alexeev S.V., Bryantsev M.A, 2007.:** Calculation and choice of key parameters of a cascade exchanger of pressure / Internal combustion engine, NTU "HPI" Scientific and technical journal.-. Kharkov. - No.1. - page-57-62.
8. **Krajniuk A.I., Alexeev S.V., Bryantsev M.A, 2006.:** Features of working process of a cascade exchanger of pressure / Internal combustion engine, NTU "HPI" Scientific and technical journal.-. Kharkov. - No. 2. - page-102-105.

9. **Krajniuk A.I., Storcheous J.V., Gogylya A.M., Danileychenko A.A., Bryantsev M.A, 2001.:** Calculation of processes of a heatmass exchange in blowing-off volumes of heatpower plants. / Politechnika szczecinska. Wydzial mechaniczny. Katedra eksploatacji pojazdow samochodowych. Towarzystwo wiedzy powszechnej oddzial regionalny w szczecinie. Eksploatacja silnikow spalinowych. Zeszyt Nr.3.
10. **Krajniuk A.I., 2010.:** New schemes and principles of the organization of working processes of heat power cars. / The messenger of the East Ukrainian national university of V. Dahl, - Lugansk: Publ. EUSU , - № 7(125). P.21.- page.-197-200.
11. **Krajniuk A.I., Krajniuk A.A., 2005.:** Features of the organization of working process of units of cascade compression and some directions of their application / The messenger of the East Ukrainian national university of V. Dahl, - Lugansk: Publ. EUSU , - № 8(90). - page.-169-173.
12. **Krajniuk A.I., 2009.:** Thermal compressor of cascade pressure exchange/ Silesian university of technology publication faculty of transport. I International Scientific Conference. Transport problems, Katowice-Kroczyce. №17-19.- p.186-191.
13. **Krajniuk A.I., 2010.:** Development of supercharging systems of internal combustion engines with the cascade pressure exchanger /TEKA Komisji Motoryzacji i Energetyki Rolnictwa. – OL PAN, Lublin , №10A, p.303-310
14. **Krajniuk A.I., 2010.:** The Krajniuk's cascade exchanger and new principles of the organization of working process of the gas-turbine engine/ TEKA Komisji Motoryzacji i Energetyki Rolnictwa. – OL PAN, Lublin, №10C, p.151-162
15. **Krajniuk A.I., 2010.:** The organization of working process and sampling of parameters of Krajniuk cascade-recuperative pressure exchanger / TEKA Komisji Motoryzacji i Energetyki Rolnictwa. – OL PAN, Lublin, №10C, p.140-150
16. **Krajniuk A.I., Alexeev S.V., Krajniuk A. A., 2009.:** The ICE supercharge system with boosting air deep cooling // Internal combustion engine, NTU "HPI" Scientific and technical journal.-. Kharkov. - No. 1. - page-57-61.
17. **Krajniuk A.I., Alexeev S.V., Kovtoon A.S., 2011.:** The cascade pressure exchange supercharge system with boosting air deep cooling test results / Aerospace equipment and technology: collection of works: Thermal engines and power installations. Kharkov. №10(87), - page 168-172.
18. **Krajniuk A.I., Bryantsev M.A., 2010.:** Krajniuk's air refrigeration unit / Alternative kilowatt. Russia, Rybinsk, Publishing house “Media grandee ”. -№5. - page -40-45.
19. **Krajniuk A.I., Bryantsev M.A., Krajniuk A.A., 2010.:** The new principle of transport refrigeration installation working process organization on Krajniuk's cascade pressure exchanger base. The messenger of the East Ukrainian national university of V. Dahl, - Lugansk: Publ. EUSU , - № 5(147). P.1.- page.-168-177.
20. **Krajniuk A.I., Bryantsev M.A., 2010.:** The Krajniuk's cascade-recuperative pressure exchange refrigeration unit. / Carload park. International information scientific and technical magazine, – Harkov: Publishing house Rolling stock of Tekhnostandart corporation, - № 11.- page.-25-30.
21. **Krajniuk A.I., 2011.:** The new schemes and principles of heat-power machines working processes organization. / The messenger of the East Ukrainian national university of V. Dahl, - Lugansk: Publ. EUSU, № 12(166). P.1.- page.-94-106.
22. **Krajniuk A.I., Krajniuk A.A., 2004.:** Cascade exchanger of pressure. Patent UA №77261A, International patent classification F 02 B 33/00, request №20040806986; it is declared 21.08.2004; it is published 15.11.2006, bulletin №11.
23. **Krajniuk A.I., Pushov V.V, Klyus O.V., Krajniuk A.A., 2010.:** Pressure exchanger. Patent RF №2382240, International patent classification F 02 B 33/00, request № 2008143154/06; it is declared 30.10.2008; it is published 20.02.2010, bulletin №5.
24. **Krajniuk A.I., 2009.:** Krajniuk's air refrigeration unit. Patent UA №44210, International patent classification F 25 B 9/00, request № U200903450; it is declared 10.04.2009; it is published 25.09.2009, bulletin №18.
25. **Rowland, F.S., Molina, M.J., 1994.:** “Ozone Depletion: 20 Years the Alarm”, Chemical and Engineering News, Vol. 72, pp: 8-13.

МЕТОД ВЫБОРА ПАРАМЕТРОВ ВОЗДУШНОЙ ХОЛОДИЛЬНОЙ МАШИНЫ КАСКАДНОГО ОБМЕНА ДАВЛЕНИЕМ

Александр Крайнюк, Юрий Сторчеус, Александр Данилейченко, Максим Брянцев

Аннотация. Приведен алгоритм расчета основных размерных параметров составных агрегатов системы, реализующей рабочий цикл воздушной холодильной машины каскадного обмена давлением заданной холодопроизводительности и глубины охлаждения объекта. Дана методика оценки затрат тепловой и механической энергии на осуществление цикла воздушной холодильной машины.

Ключевые слова: каскадный обменник давления, массообмен, воздушная холодильная машина, рекуперация, холодопроизводительность.