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Titlov A.,
Vasyliv O.,
Abdelkader A.,
Morozov O.

ANALYSIS OF ENERGY CHARACTERISTICS OF ABSORPTION WATER-AMMONIA REFRIGERATION MACHINES IN THE WASTE HEAT RECOVERY SYSTEMS OF GAS TURBINE INSTALLATIONS ON GAS MAIN PIPELINES

Одним з перспективних напрямків зниження експлуатаційних втрат в магістральних газопроводах є попереднє охолодження компріміруемого газу за допомогою тепловикорестуючих абсорбційних водоаміачних холодильних машин (АВХМ), які утилізують дармове тепло відхідних продуктів згоряння газоперекачувальних агрегатів. Об'єктом дослідження є енергетичні характеристики АВХМ в широкому діапазоні параметрів експлуатації (температур зовнішнього повітря), які в даний час не вивчені. Дане дослідження присвячено розробці методики моделювання режимів АВХМ, проведенню аналітичних досліджень і аналізу отриманих результатів в широкому діапазоні температур зовнішнього повітря.

Дослідження проводилося за допомогою теоретичного аналізу циклів АВХМ в широкому діапазоні температур зовнішнього повітря і температур об'єкта охолодження. Аналіз отриманих результатів показав, що в діапазоні розрахункових параметрів має місце максимум енергетичної ефективності АВХМ. Найбільш очевидно наявність максимуму для умов роботи при температурах охолоджуючої середовища 20...32 °C і низьких температурах об'єкта охолодження (мінус 25 °C). При зниженні температур об'єкта охолодження максимум енергетичної ефективності зміщується в область високих температур гріючого середовища, а його чисельні значення зменшуються. При температурах, що гріє джерела від 90 °C до 130 °C, електрична потужність циркуляційного насоса має максимальне значення. Надалі із зростанням температури, що гріє джерела спостерігається її асимптотичне зниження і повільне зменшення. Найбільші зміни при цьому відбуваються при підвищених температурах охолоджуючого середовища (32 °C).

Результати моделювання дозволяють визначити найбільш енергетично вигідні режими роботи АВХМ з різними джерелами теплової енергії (температури від 90 до 160 °C) і проводити розробку систем охолодження для широкого діапазону температур (мінус 30...15 °C). Для досягнення таких оптимальних режимів необхідна відповідна комбінація складу робочого тіла і температури, що гріє джерела.

Ключові слова: *абсорбційні водоаміачні холодильні машини, дармове тепло, енергетична ефективність, попереднє охолодження природного газу, магістральні газопроводи.*

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1. Introduction

The gas transportation system of Ukraine is designed to supply gas to both domestic consumers and fuel transit to Western Europe. For the transportation of natural gas through pipelines at compressor stations (CS), gas pumping units (GPU) were installed, the energy carrier for which, in most cases, is transported natural gas. 0.5...1.5 % of the volume of transported gas is consumed (burned) by the GPU drive [1].

The efficiency of most of the GPU fleet currently in operation in Ukraine is in the range of 24...27 [2].

Pumping costs can be reduced as follows:
– replacement of existing units with low efficiency by more economical ones, with an efficiency of 36 % and higher;
– modernization of existing equipment with the use of new approaches to the organization of the processes of compression (compression) at the main compressor stations.

The situation with the replacement of existing equipment with modern equipment is associated with significant investments, on the one hand, and the uncertainty with the transit of Russian natural gas through Ukrainian gas transmission systems in the near future [3, 4].

An analysis of ways to increase the energy efficiency of gas compressor units [5] shows that one of the promising areas could be pre-cooling of the compressed gas using heat-using absorption water-ammonia refrigeration machines (AWRM) that utilize the waste heat of the exhaust products of combustion [6, 7]. It is shown [5] that for the current economic situation (July 2019) in the Ukrainian gas market, the daily decrease in operating costs in typical gas pipelines with a decrease in gas temperature before compression by 20 K in gas compressor units ranges from 1800 to 3360 USD.

Since gas is transported year-round under various climatic (temperature) environmental conditions, it is necessary to evaluate the changes in the energy efficiency of the AWRM cycle.

Of particular interest are also the lowered temperatures of the cooling object, which make it possible to increase the useful effect of pre-cooling natural gas before compression [5].

Any systematic research in this area of refrigeration is currently unknown to the authors.

*Thus, the object of resear*ch is the energy characteristics of the AWRM in a wide range of operating parameters (outdoor temperatures) and temperatures of the cooling object.

The aim of this research is conducting analytical studies of the energy characteristics of the AWRM in the waste heat recovery systems of gas turbine plants on gas pipelines in a wide range of outdoor temperatures and temperatures of the cooling object.

2. Methods of research

One of the features of the AWRM is the interdependence of temperatures in the characteristic processes of the cycle – the temperature of the heating medium t_h, the temperature of the cooling medium t_w, and the temperature of the cooling object t_{ob}. Of the three temperatures, only two can be arbitrarily specified [8].

When modeling AWRM cycles, a subsystem of library functions of the thermodynamic and thermophysical properties of pure ammonia and a water-ammonia solution (WAS) is developed, based on the use of standard approximation functions (linear or spline) of the MathCAD system [9].

Fig. 1 is a diagram of a pumping AWRM with two regenerative heat exchangers – solutions (RHS) and ammonia (RHA) [8].

A low-potential heat is supplied to the AWRM 1 generator, which is filled with liquid WAS, as a result of which a low-boiling component (ammonia) with insignificant particles of water vapor will mainly boil out of the solution. The steam enters the rectifier 2, in which the cooled saturated WAS with RHS 5 and absorber 4 flows towards the steam coming from the generator 1. In this case, less volatile water vapor condenses first, increasing the concentration of ammonia in the stream. Further, the WAS vapors fall into the reflux condenser 3. On its cold tubes, water vapor that remains after the rectifier 2 is the first to condense. The presence of the rectifier 2 and the reflux condenser 3 in the AWRM circuit allows almost completely eliminating water vapor in the ammonia vapor stream that goes to the condenser 7. Next, the ammonia vapor enters the condenser 7, liquefies with the heat of the phase transition, enters the RHA 8, in which the cold ammonia vapor that comes from the evaporator 9 to the absorber 4 is preheated. Due to this, the thermal coefficient of the AWRM cycle increases.

Liquid ammonia is throttled and boils in the evaporator 9, while generating artificial cold. Ammonia vapor comes from the evaporator 9, through the RHA to the absorber 4, where it is absorbed and dissolved in a weak (with a minimum ammonia composition) WAS. A weak WAS through the throttle enters from the generator 1 into the absorber 4 through the RHS 5, in which a strong (saturated) WAS is heated. Saturated WAS using pump 6 enters the rectifier 2 and the cycle repeats again.

The initial data for the calculation are temperatures:

a) cooling medium t_w;

b) cooling object t_{ob};

c) temperature differences on elements that do not explicitly take into account heat transfer conditions and heat under-recovery:

- Δt_h – the temperature difference between the weak WAS and the heating source of heat of the generator;
- Δt_{WK}, Δt_{WA}, Δt_{WD} – temperature head in the condenser, absorber, reflux condenser with a cooling medium;
- Δt_{TO} – temperature head between the flows of weak and strong WAS at the cold end of the RHS;

d) Q_o – cooling capacity of the evaporator.

The variable parameter is the temperature of the heating heat source t_h.

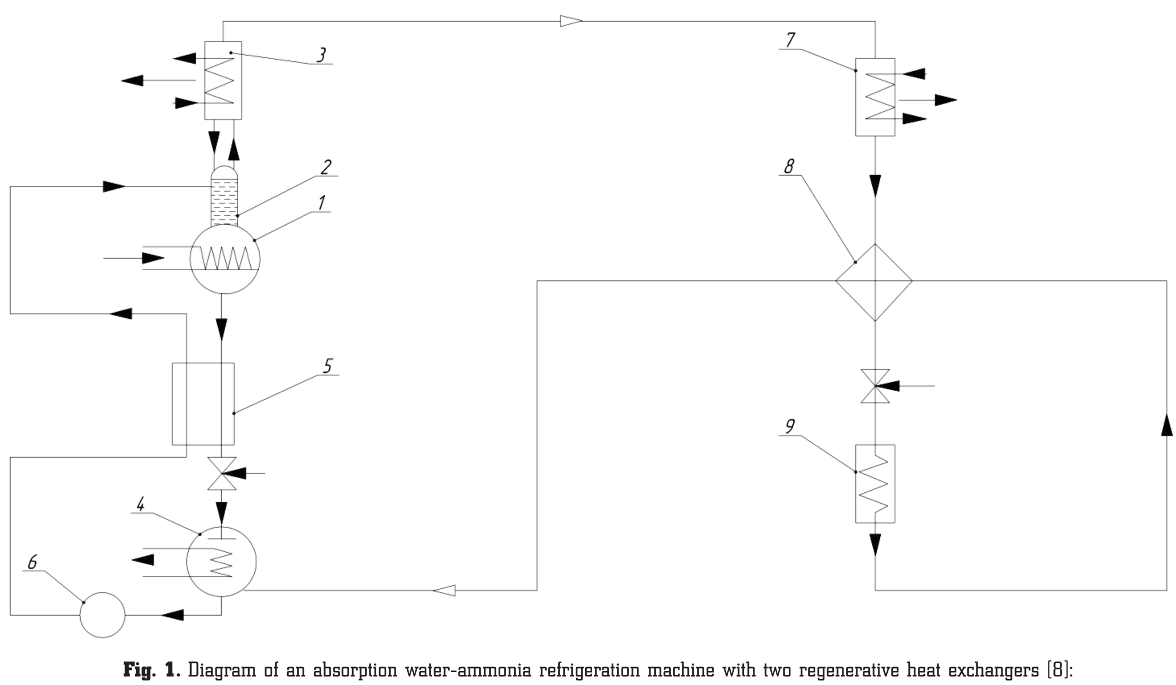

Fig. 1. Diagram of an absorption water-ammonia refrigeration machine with two regenerative heat exchangers [8]:
1 – generator; 2 – rectifier; 3 – reflux condenser; 4 – absorber; 5 – regenerative heat exchanger solutions; 6 – pump; 7 – capacitor; 8 – regenerative heat exchanger of ammonia; 9 – evaporator

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3. Research results and discussion

At the first stage, two pressure levels in the AWRM cycle are determined by the ammonia condensation temperature and the temperature of the cooling object.

Then, thermodynamic parameters (temperature, specific volume and mass fraction) and thermal functions (specific enthalpy) are found at the characteristic points of the cycle (at the input and output of the elements).

The specific thermal loads on the elements are determined:

a) cooling capacity of the evaporator q_o;
b) thermal load of the reflux condenser q_D;
c) thermal load of the generator q_G;
d) thermal load of the capacitor q_K;
e) the thermal load of the absorber q_A;
f) electric power of the circulating pump l_{pump}.

Let's find the heat input:

$$q_1 = q_G + q_o + l_{pump}, \tag{1}$$

and removed heat:

$$q_2 = q_A + q_K + q_D. \tag{2}$$

The mass flow rate of ammonia vapor in the evaporator is:

$$D = \frac{Q_o}{q_o}. \tag{3}$$

Full thermal loads on the AWRM elements are found by relations of the type:

$$Q_i = q_i \cdot D. \tag{4}$$

Thermal cycle coefficient of pumping AWRM:

$$\eta = \frac{Q_o}{Q_G}. \tag{5}$$

Based on the above algorithm, the AWRM cycles are analyzed and their energy characteristics are determined – the heat coefficient and the operation of the circulation pump, depending on the temperature operating conditions.

The temperature of the cooling object in the calculations is: minus 5 °C; minus 15 °C; minus 25 °C, cooling medium temperature: from 10 to 32 °C.

The temperature head Δt_{WK}, Δt_{WA}, Δt_{WD}, Δt_T are taken equal to 5 °C, and Δt_h=10 °C.

The minimum temperature of the heating medium in the analysis is 90 °C, the maximum is 170 °C. The minimum temperature value is chosen at the boundary of the AWRM cycle implementation, and the maximum temperature is taken into account taking into account the onset of active corrosion of the structural material.

When calculating the electric power of the circulation pump, the AWRM cooling capacity of 1 kW is taken.

The results of calculating the AWRM are presented in the form of graphical dependencies in Fig. 2.

Analysis of the calculation results allows to draw the following conclusions.

First, in the range of design parameters, the maximum energy efficiency of the AWRM takes place. The most obvious is the presence of a maximum for operating conditions at cooling medium temperatures of 20...32 °C and low temperatures of the cooling object (minus 25 °C).

As the temperature of the cooling object decreases, the maximum energy efficiency shifts to the region of high temperatures of the heating medium, and its numerical values decrease.

So, for example, when the temperature of the cooling medium is 26 °C and the temperature of the cooling object minus 5 °C, the maximum thermal coefficient of the AWRM cycle (0.53) takes place at the temperature of the cooling object 110 °C. At temperatures of the cooling object minus 15 °C and minus 25 °C, the temperatures and thermal coefficients, respectively, will be: 120 °C and 140 °C; 0.44 and 0.34.

Analysis of the calculation results shows that such a course of calculated dependencies is explained as follows.

In the region of low temperatures of the heating medium (up to the maximum of the thermal coefficient) – high WAS circulation between the generator and the absorber (from 6 to 112), which is due to the narrow degassing area (0.006...0.033).

In the region of high temperatures of the heating medium (after the maximum of the heat coefficient) – an increase in the proportion of water in the steam stream of the ammonia-water mixture leaving the generator. So, for example, at a temperature of the cooling medium of 26 °C and the temperature of the cooling object minus 5 °C, the increase in the proportion of water vapor in the mixture is from 0.036 to 0.408, i. e., more than 10 times.

In the first case, additional heat inflows to the generator with the flow of a strong WAS.

In the second case, despite the decrease in the WAS circulation frequency, the heat load in the generator increases due to additional energy costs for evaporation of the absorbent – water. The increase in the thermal load of the reflux condenser also increases by more than 10 times (at a temperature of the cooling medium of 26 °C and a temperature of the cooling object minus 5 °C – from 0.024 kJ/kg to 2,200 kJ/kg).

The decrease in the thermal coefficient of the AWRM cycle with a decrease in the temperature level of the cooling object is explained by the fact that for such regimes a WAS with an increased absorbent content is required. So, for example, at a cooling medium temperature of 26 °C, a decrease in the temperature of the cooling object from minus 5 °C to minus 25 °C requires a decrease in the proportion of ammonia in a weak WAS from 0.439 to 0.129. At the same time, despite the decrease in the heat load of the generator due to the decrease in the multiplicity of the liquid circulation, the process of additional evaporation of water vapor from the WAS has a leading adverse effect on the energy efficiency of the AWRM cycle.

In the calculation range, in all cases, an increase in the temperature of the heating source leads to a sharp decrease in the power of the circulation pump pumping a strong solution from the absorber to the generator.

As calculations show, at heating source temperatures from 90 °C to 130 °C (depending on the temperature of the cooling medium), the electric power of the circulation pump has a maximum value. Subsequently, with an increase in the temperature of the heating source, its asymptotic decrease and slow decrease are observed. The greatest changes in this case occur at elevated temperatures of the cooling medium (32 °C).

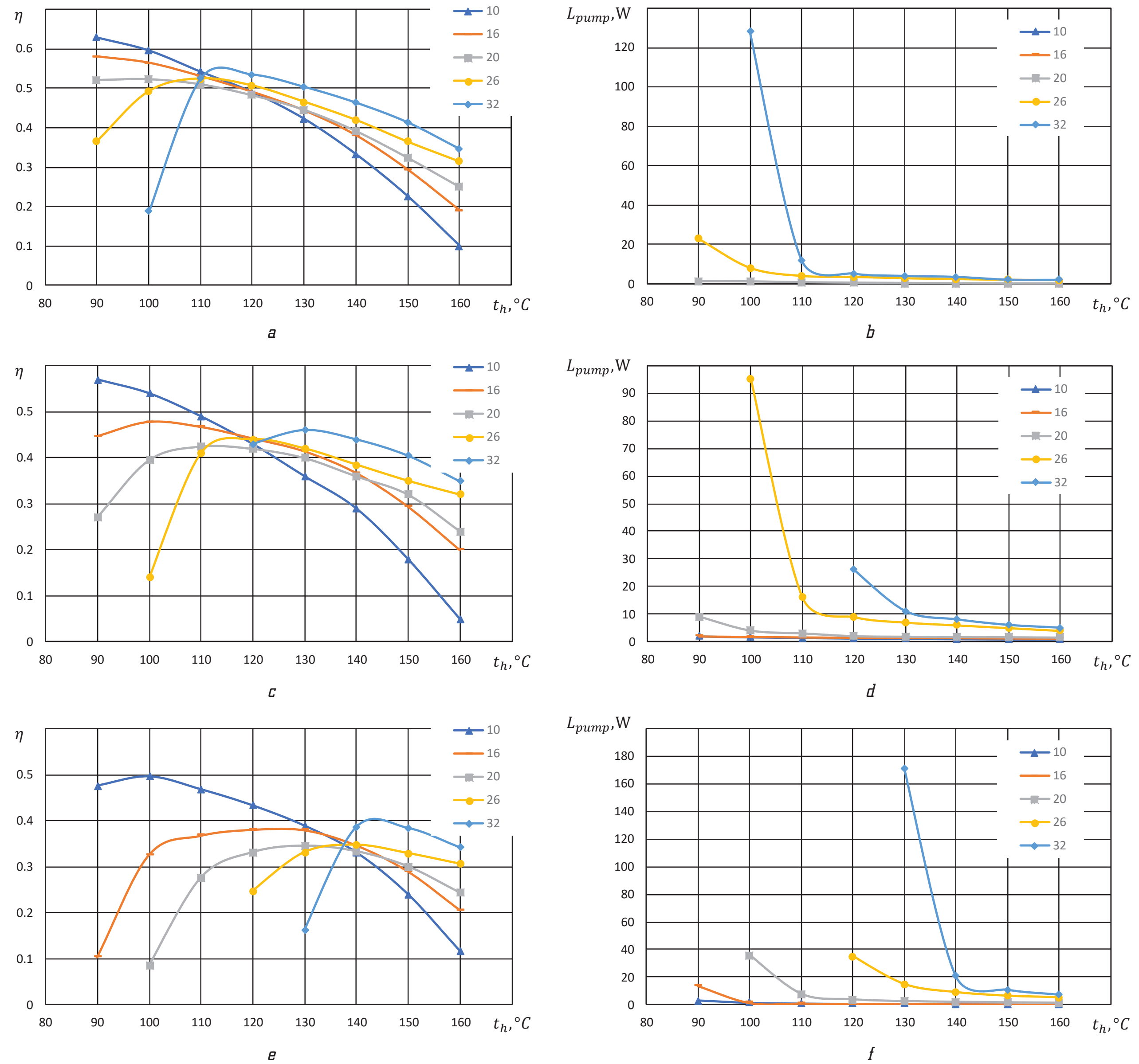

Fig. 2. The results of calculating the thermal coefficient of the cycle (η) of the absorption water-ammonia refrigeration machine and the electric power of the circulation pump (L_{pump}) at various temperatures of the cooling (t_w), heating medium (t_h) and at the temperature of the cooling object (t_{ob}): *a, b* – t_{ob}=minus 5 °C; *c, d* – t_{ob}=minus 15 °C; *e, f* – t_{ob}=minus 25 °C

The nature of the calculated dependences for the circulation pump (Fig. 2, *b*, *d*, *f*) coincides with the results of studies by other authors [10].

The asymptotic increase in the electric power of the pump at lower temperatures in the generation zone is explained by an increase in the degassing zone in the absorber due to a decrease in the mass fraction of ammonia in the weak WAS stream.

4. Conclusions

A technique for modeling AWRM modes is developed and an analysis of the results is carried out in a wide range of working thermodynamic parameters (ambient temperature – 10...32 °C, temperature of the cooling object – minus 30...15 °C). It is shown that during AWRM cycles, there are modes with maximum energy efficiency in the practical temperature ranges of the cooling medium (from 10 to 32 °C) and cooling objects (minus 25...minus 5 °C). To achieve the AWRM operating modes with maximum energy efficiency, an appropriate combination of the working fluid composition and the temperature of the heating source is necessary, which can be achieved using an automatic control system.

The results can be used in the design of energy-efficient systems for transporting natural gas through pipelines and help reduce operating costs during the operation of gas compressor units.

References

1. Serediuk, M. D. (2002). *Proektuvannia ta ekspluatatsiia naftoproduktoprovodiv.* Ivano-Frankivsk: IFNTUNH, 282.
2. Govdiak, R. M. (2012). Povyshenie energeticheskoi i ekologicheskoi effektivnosti raboty magistralnykh gazoprovodov. *Energotekhnologii i resursosberezhenie*, 3, 56–62. Available at: http://nbuv.gov.ua/UJRN/ETRS_2012_3_11.

 ISSN 2664-9969

3. *MSHU «Skolkovo»: ODIN GOD DO CHASA «Ch»: v poiskakh kompromissa po ukrainskomu gazovomu tranzitu – Dekabr 2018* (2018). Khanty-Mansiisk. Available at: https://nangs.org/analytics/mshu-skolkovo-odin-god-do-chasa-ch-v-poiskakh-kompromissa-po-ukrainskomu-gazovomu-tranzitu-dekabr-2018-pdf
4. Dombrovskii, A., Unigovskii, L. (2018). GTS: vremia ne zhdet. *Zerkalo nedeli, 8-9,* 7.
5. Titlov, O., Vasyliv, O., Sahala, T., Bilenko, N. (2019). Evaluation of the prospects for preliminary cooling of natural gas on main pipelines before compression through the discharge of exhaust heat of gas-turbine units. *EUREKA: Physics and Engineering, 5,* 47–55. doi: http://doi.org/10.21303/2461-4262.2019.00978
6. Moroziuk, L. I. (2014). Development and improvement of the heat using refrigerating machines. *Refrigeration Engineering and Technology, 50 (5 (151)),* 23–29. doi: http://doi.org/10.15673/0453-8307.5/2014.28695
7. Titlov, A. S., Sagala, T. A., Artiukh, V. N., Diachenko, T. V. (2017). Analysis of the Prospects for the Use of Steam-Jet and Absorption Refrigeration Units for Technological Gas Cooling and Liquid Hydrocarbon Fuel Producing. *Refrigeration Engineering and Technology, 53 (6),* 11–18. doi: http://doi.org/10.15673/0453-8307.5/2014.28695
8. Galimova, L. V. (1997). *Absorbcionnye kholodilnye mashiny i teplovye nasosy.* Astrakhan: Izd-vo AGTU, 226.
9. Osadchuk, E. A., Titlov, A. S. (2011). Analiticheskie zavisimosti dlia rascheta termodinamicheskikh parametrov i teplofizicheskikh svoistv vodoammiachnogo rastvora. *Naukovi praci ONAKHT, 1 (39),* 178–182.
10. Ischenko, I. N. (2010). Modelirovanie ciklov nasosnykh i beznasosnykh absorbcionnykh kholodilnykh agregatov. *Naukovi praci ONAKHT, 2 (38),* 393–405.

Titlov Alexander, *Doctor of Technical Sciences, Professor, Head of Department of Heat-and-Power Engineering and Oil-and-Gas Transportation and Storing, Odessa National Academy of Food Technologies, Ukraine, ORCID: http://orcid.org/0000-0003-1908-5713, e-mail: titlov1959@gmail.com*

Vasyliv Oleg, *PhD, Associate Professor, Department of Heat-and-Power Engineering and Oil-and-Gas Transportation and Storing, Odessa National Academy of Food Technologies, Ukraine, ORCID: http://orcid.org/0000-0002-0597-8863, e-mail: titlov1959@gmail.com*

Abdelkader Alnamer, *Postgraduate Student, Department of Heat-and-Power Engineering and Oil-and-Gas Transportation and Storing, Odessa National Academy of Food Technologies, Ukraine, ORCID: http://orcid.org/0000-0002-1312-1990, e-mail: ado_namer@yahoo.com*

Morozov Alexey, *Department of Heat-and-Power Engineering and Oil-and-Gas Transportation and Storing, Odessa National Academy of Food Technologies, Ukraine, ORCID: http://orcid.org/0000-0001-5923-2584*