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V International
Scientific-Technical Conference

Book of abstracts

ACTUAL PROBLEMS OF RENEWABLE ENERGY, CONSTRUCTION AND ENVIRONMENTAL ENGINEERING

The time and place of the meeting: 3 – 5 June 2021
Faculty of Environmental, Geomatic and Energy Engineering,
Kielce University of Technology, Poland
al. Tysiąclecia Państwa Polskiego 7, 25-314 Kielce

Conference Chairs:

Anatoliy Pavlenko
prof. doctor of science Department of Building Physics and Renewable Energy, Kielce University of Technology

Aleksander Szkarowski
prof. doctor of science Head of Department of Construction Networks and Systems, Koszalin University of Technology

KIELCE 2021

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KIELCE 2021

The organizers:

- Kielce University of Technology, Faculty of Environmental, Geomatic and Energy Engineering (Poland)
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- University of Žilina, Departament of Power Engineering (Slovakia)
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The general program of the ConferenceTime	Time	Place	Moderator
Registration of participants	3 June 2021 10:00-10:45		The Organizing Committee
This opening session, plenary session	4 June 2021 11:00-12:00		Prof. doctor of science **Anatoliy Pavlenko** Prof. doctor of science **Aleksander Szkarowski** Prof. doctor of science **Engvall Klas** Prof. doctor of science **Yevstakhii Kryzhanivskyi** Prof. doctor of science **Borys Basok** Prof. doctor of science **Ladislav Lazić** Prof. doctor of science **Milan Malcho** Prof. doctor of science **Valerii Deshko**
Actual problems of building physics	4 June 2021 12:30-14:00		Prof. doctor of science **Valerii Deshko**
Actual problems of environmental engineering and ecology	4 June 2021 12:30-14:00		Prof. doctor of science **Hanna Koshlak** Prof. doctor of science **Vera Ulyasheva**
Actual problems of thermal and renewable energy	4 June 2021 12:30-14:00		Prof. doctor of science **Anatoliy Pavlenko**
Actual problems fundamental heat and mass transfer	4 June 2021 12:30-14:00		Prof. doctor of science **Ladislav Lazić** Prof. doctor of science **Łukasz Orman**
Theoretical foundations of thermal machines	4 June 2021 12:30-14:00		Prof. doctor of science **Borys Basok** Prof. doctor of science **Milan Malcho**
Actual problems of quantitative assessment of the low-carbon energy and industry	4 June 2021 12:30-14:00		Prof. doctor of science **Malik Ziganshin** Prof. doctor of science **Aleksander Szkarowski**
Continued work of section	5 June 2021 12:30-14:00		The section moderators
Discussion	5 June 2021 14:15-15:30		The section moderators
Summary and the closing of the conference,	5 June 2021 16:00		The Organizing Committee

Anatoliy Pavlenko - professor of Kielce University of Technology, Department of Building Physics and Renewable Energy. Scientific direction of work - thermophysics of dispersed media. Scientific interest - mathematical modeling of thermophysical processes occurring in liquids in the metastable state.

Dear Colleagues,

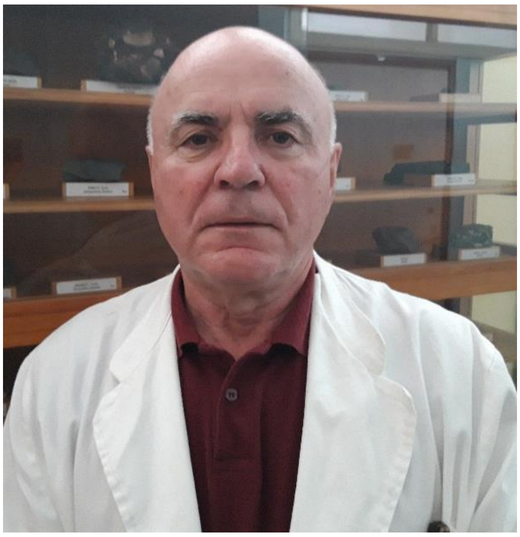
I cordially greet all participants of our 5th international conference.
For five years now our Conference traditionally consolidates all those best scientific ideas, that authors present in their speeches.
All these years we examines the questions and problems of modern power engineering, energy technology of power-consuming industry branches, alternative sources of energy, resource conservation, questions of modeling the process of industrial equipment, processes and equipment of various branches of industry, questions of automated control systems and information processing, heat-and mass-exchanging processes and equipment of special technique, questions and problems of electricity and power control.
I see these directions also in the submitted works this year.
Unfortunately, the restrictions associated with the coronavirus did not allow us to traditionally meet for discussion. I think that next year we will be able to do this at our university.
I wish everyone good health and new creative achievements.

Conference Chairs
Anatoliy Pavlenko

Aleksander Szkarowski – profesor dr hab. inż., Kierownik Katedry Sieci I Instalacji Budowlanych Politechniki Koszalińskiej. Zainteresowania naukowe i kierunki badań – poprawa sprawności i ekologiczności spalania paliwa w energetyce i przemyśle oraz efektywności wykorzystania energii w ciepłownictwie, ogrzewnictwie i wentylacji.

Drodzy Koledzy i Przyjaciele – Uczestnicy naszej Konferencji!

Po raz piąty spotykamy się, żeby w towarzyskiej atmosferze a także i ostrych dyskusjach wymieniać zdania na temat wyników badań, pomysłów i wynalazków, które szanowni Autorzy przedstawią w swoich referatach. Nie każda konferencja pochwalić się może wysokim poziomem naukowym w tak obszernym zakresie tematów. Dowiadujemy się o nowych badaniach związanych z procesami i urządzeniami różnych gałęzi przemysłu, zagadnieniami zautomatyzowanych systemów sterowania i przetwarzania informacji, procesami wymiany ciepła i masy oraz urządzeniami specjalistycznej techniki, jak również zagadnieniami i problemami regulacji energii i mocy elektrycznej.
Nie zabraknie tych ważnych tematów także w naszych spotkaniach.
Życzę wszystkim Uczestnikom dobrego zdrowia i humoru oraz nowych osiągnięć twórczych.
Współprzewodniczący Konferencji
Aleksander Szkarowski



Ladislav Lazić - professor of University of Zagreb, Faculty of Metallurgy, Department of Mechanical Metallurgy, Laboratory of Thermal Technique and Mechanical Engineering. The main areas of scientific activities: Computation of high-temperature radiative heat transfer, Thermodynamics of fuel utilization and pollutant formation, Design and efficiency of industrial furnaces and heating equipments, CFD approach to combustion modelling and numerical simulation of furnace processes, Finite element technique in simultaneous transient conduction and thermal stress analysis, Techniques in the reduction of NOx emissions and, in particular, Metallurgical archeology.

Dear colleagues, organizers and participants of the 5th Conference,

it is my great pleasure to be a member of the Scientific and organizing committee of the conference for several reasons. The main reason is that in today's world, energy and ecology are in the focus of interest, and I am glad that the organizers of the conference are institutions of countries with whose scientific and educational institutions I have many years of successful cooperation. I regret that due to the Coronavirus Pandemic we will not have the opportunity for personal contact, but we remain in the hope that we will realize this next year.

I greet you from my city of Petrinja, where I live, and from Sisak, where I work, cities that were badly damaged in two strong earthquakes in December last year. However, that did not destroy our hope for recovery.

I wish you all the best in business and personal life,

Ladislav

Table of Contents

MUNICIPAL HEAT ENERGY OF UKRAINE: ADAPTATION TO GLOBAL WARMING

Boris Basok[1], Anatoliy Pavlenko[2], Evgen Bazeev[1]

[1] Institute of Engineering Thermophysics National Academy of Sciences of Ukraine;
[2] Kielce University of Technology, Poland

Introduction. Global warming and the "greenhouse effect" are among the most discussed issues in physics and geopolitics, causing environmental protests in the world. Both anthropogenic (greenhouse gas emissions) and natural concepts of global warming with dramatic consequences of the global climate change and its individual regions are widely discussed. The remarkable trend of global temperature rise, recorded since the middle of the twentieth century, stimulated the expert community to begin active discussion of possible ways to address the global warming situation, its consequences and problems related to adaptation to climate change. The problem of adaptation to global warming concerns various economic and human life spheres, including public energy service. Therefore, Ukraine has launched innovative energy-efficient technologies to mitigate the effects of climate change effects, including renewable energy sources, as well as developed engineering system equipment for the energy supply of the housing and communal services. Architectural and construction solutions have been optimized, making it possible to reduce the use of energy resources and resulting greenhouse gas emissions.

Results. The scenarios of anthropogenic enhancement of the global greenhouse effect, thermal, hydrological conditions and the main conclusions for possible temperature climate changes in Ukraine were considered in the early 90s of the 20th century and presented in [1]:

1. with global anthropogenic warming by about 1 ºC (in the first quarter of the 21st century) in the southern regions of Ukraine the level of warming will almost coincide with the global one, and in the north it may increase up to 40%.
2. the latitudinal temperature gradient on the territory of Ukraine will decrease in absolute value to 10% (in the modern era the latitudinal gradient of surface annual temperature is about 0.8 ºC per 1º latitude).
3. if the annual global temperature rises by 3 ºC (around the middle of the 21st century), the temperature regime in the northern regions of Ukraine may become similar to the temperature regime of its southern regions.

It is pointed out that the results, obtained by many researchers, dealing with mathematical modeling of the terrestrial climate system dynamics to develop RCP scenarios of regional climate change with anthropogenic enhancement of the greenhouse effect are still unreliable. The problem is insufficient deep study of the whole set of physical processes, forming the latitudinal-longitudinal distribution of temperature fields and other climatic parameters (total precipitation, weather variability, etc.).

The publication of forecasts ten years later showed that in Ukraine the increase in the average surface temperature in the period during 1900-2000 was 0.4-0.6 ºC, and by region: north-east about - 1 ºC, woodland and forest-steppe - 0.7-0.9 ºC, steppe - 0.2-0.3 ºC. By seasons: in winter - 1.2 ºC, in spring - 0.8 ºC, in summer - 0.2-0.3 ºC. A negative trend in the number of frosty days and climate decontinentalization were detected. Predictive estimates of changes in the average surface temperature in Ukraine by 2050 are reduced to the fact that the increase will be 1.5-2.0 ºC, and in January for the south - 2.0 ºC, for the north - 2.8 ºC and in July for Ukraine - 0.5-1.0 ºC [2].

The recent forecast for warming in Ukraine for a number of periods up to 2100 is shown in Fig. 1. It is evident that the results of instrumental measurements of temperature rise and warming forecasts in Ukraine generally correspond to the trends of global temperature change, and Ukraine faces the same challenges and risks that the world is concerned about [3].

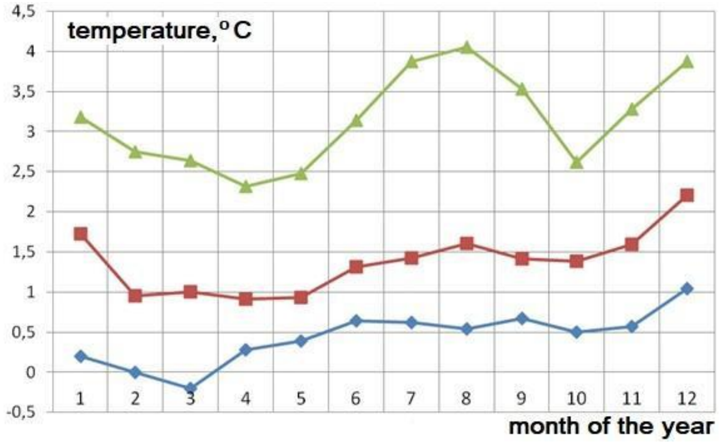


Fig. 1 - Projections of changes in average monthly air temperatures in Ukraine with confidence intervals for an ensemble of 10 RCMs (regional numerical climate models) as compared to 1991-2010 yrs. [3]. Designations: ◆ - 2011-2030 yrs. (average value - 0.44 ºC), - 2031-2050 yrs. (average - ■1.57 ºC), ▲- 2051-2100 yrs. (average - 3.15 ºC).

The adaptation to global warming in the economy is relevant to, mainly, the agro-industrial complex, as a food security guarantor and a major exporter of products, as well as the energy sector of Ukraine, communal energy in particular. The climate change results in a change in temperature characteristics - degree-days of warm and cold periods, energy costs for heating and air conditioning of the housing and communal sector, etc.

The strategy purpose of climate change adaptation for municipal energy is to ensure the consistent municipal energy supply (heat supply and air conditioning) to guarantee a comfortable and reliable present and future heat and cold supply, technically and economically sound, while complying with environmental requirements.

It is vital for housing and communal services and district heating to know how the demand for heat supply may change in the coming years due to global warming. According to observations, the parameters of low-frequency variability of the average heating temperature in the south of Ukraine and the corresponding change in energy consumption for population heating needs were assessed, changes in average monthly temperatures in the regions of Ukraine from 1961 to 1991 were analysed, the trends from October to April were determined and energy consumption for building heating was calculated [13-15].

Further assessment, determination of global warming effects and its impact on the heating period characteristics was expanded and deepened. In studies [2, 16–19] temperature characteristics and dynamics of the heating period for 1900–2013 and in separate periods of this range are determined and a noticeable reduction of the heating period duration is shown. The previous publications on changes in climatic characteristics of the heating period were briefly analysed using data that are important for the study of change features in the heating period parameters [2]. The climatic effect on man and socio-economic systems is analysed, in particular, climate changes related to global warming and their impact on the efficiency of activities in various economic sectors of Ukraine until 2050. [20, 21]. The change in the heating period duration, solar and wind energy development options were assessed [16–21]. The results of these studies can serve as the basis for further activities on the adaptation of municipal energy supply to climate change.

The main volume of anthropogenic greenhouse gas emissions (about 70–80% [23]) falls on the fuel and energy complex. It is generated mainly during the combustion of hydrocarbon fuels in "large" energy (thermal power plants) and in municipal thermal power (thermal power plants, district and autonomous boilers). Regarding this energy sector, greenhouse gas emissions can be reduced as follows:

1) increase of energy efficiency in the use of energy resources by:

– improvement of organizational and economic mechanisms of fuel and energy sector;

– introduction of energy-efficient-oriented innovative technologies in the whole technological chain - from production to final use of energy resources;

2) optimization of hydrocarbon fuel combustion processes with improved environmental performance (low-emission environmentally friendly combustion);

3) use of economically and environmentally sound renewable energy sources - the so-called low-carbon energy, as well as efforts to achieve carbon-neutral energy.

Conclusions. In view of the above, it is necessary to make adjustments to the management system of many economic sectors in Ukraine, like communal energy, heat supply of buildings and air conditioning of housing and communal services. Therefore, it is utterly important to develop measures for the formation of municipal energy adaptation policy - identify and justify the main directions of solving key problems to adapt municipal energy supply (heat and cold supply) to global warming, taking into account norms and standards in energy efficiency, public construction and reconstruction with harmonization of normative documents with the European Union directives in the field of municipal energy supply and energy efficiency of buildings. It is also necessary to change the applied temperature characteristics, especially the number of degree-days and terms of warm and cold periods, improve engineering systems and equipment of power supply of buildings, optimize thermal characteristics of enclosing building structures, in particular in the construction of passive houses “zero-energy","green” houses, eco-houses and improve the architecture of buildings, etc.

Provisions of adaptation strategy to climate change of municipal energy supply shall contain:

– strategic objectives of adaptation to global warming;

– characteristics of conditions and identification of threat factors for global warming for municipal energy supply (heat supply and air conditioning);

– determination of measures, indicators, their threshold values, posing a threat to municipal energy supply;

– principles of adaptation to global warming;

– development of measures and mechanisms that eliminate or reduce the impact of negative conditions and factors due to global warming;

– development of monitoring as an operational-analytical and diagnostic control system for adaptation measures and mechanisms to eliminate emerging deviations from the threshold values of indicators that pose a threat to municipal energy supply.

Implementation of the strategy for adaptation of municipal energy to climate change will contribute to the economy and rational use of fuel and energy and material resources, reduce energy costs of the housing and communal sector, introduction of architectural, innovative engineering solutions and improvements in living standards, as well as reducing the country's dependence on imports of fuel and energy resources, and therefore will increase the country's energy security.

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CHANGE OF EMULSION STRUCTURE DURING HEATING AND BOILING

Anatoliy Pavlenko[1], Ladislav Lazić[2]

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Abstract. The article proposes methods for evaluation of the emulsion media structure stability during their heating and boiling. The dynamic effects, accompanying processes of boiling and changes in the vapour phase volume are evaluated. Methods for determination of thermodynamic parameters of the vapour phase is proposed. It was demonstrated that intensity of heat transfer processes in the boiling emulsion can be considerably higher due to evaporation or condensation of the thermolabile phase within emulsion structure. Intensification of heat and mass transfer processes will increase the efficiency of heat and mass transfer technologies and devices, on their basis the development concept of clean energy-efficient heat technologies is proposed.

Introduction. The tasks aimed at efficiency increase of heat and mass transfer processes are very important for almost all heat technologies. One of effective methods used to increase intensification of heat and mass transfer processes is use of emulsions as operating media (B.M.Gasanov and N.V.Bulanov, 2015, Matthew L. Roesle et al., 2015, Roesle, M. L. and Kulacki, F. A., 2012, C.Y. Lin and L.W. Chen, 2008, Alexander K. Rozentsvaig and Ch. S. Strashinskii, 2015, V. Califano et al, 2014), containing various mutually insoluble fluids with certain thermophysical characteristics. When such media are heated, thermolabile fluid may boil over, accompanied by an abnormal increase in pressure and temperature fluctuations. (Pavlenko, A.M., 2018, Pavlenko, A.M. et al, 2019). An increase in the emulsion temperature is accompanied by their structure change, breakage of a dispersed fluid droplets or their merging. When thermolabile fluid boils in presuperheating, conditions, heat and mass transfer processes get intensified due to abnormally rapid change in the vapour phase volume and turbulence transition of the medium flow; it undoubtedly increases efficiency of these processes. In its turn, this phenomenon makes it possible to reduce energy costs for implementation of the emulsion homogenization processes. Processes of deformation and breakage of the dispersed phase have been studied by many researchers (E. Mura et al, 2012, Aktershev S.P. and Ovchinnikov V.V., 2013, J. Shinjo et al. 2016), nevertheless, these phenomena require further consideration in terms of heat and mass transfer efficiency. Besides, the process of deep dispersion (homogenization) itself is a separate important task. In general, the problem of fluid dispersions breakage in a continuous medium is divided into two directions (A.M.A. Attia and A.R. Kulchitskiy, 2014, J. Shinjo, et al, 2014, V. Califano et al., 2014): breaking of fluid droplets in emulsions and gas flows. When considering these processes, it is important to determine that the droplet is resistant to action of destructive forces. The calculations, presented in the literature, in most cases are based on Bond's and Weber's criteria (V. Califano et al, 2014, Dolinsky, A.A. et al, 2005), i.e. only instabilities of Rayleigh-Taylor and Kelvin-Helmholtz, which are most characteristic for emulsification or destabilization of dispersed media. Moreover, no existing model considers the process of the secondary fluid breakage, taking into account the vapour layer formation at the interface of two phases. Basically, possible processes of deformation and breakage under the action of boil-off, growth of vapour bubbles, or under the action of broken bubbles, cavitation pockets at the moment of their collapse are described.

Thus, we consider dispersed particles of the emulsion secondary phase as centres of heat and mass transfer process initiation, their number determines the intensity increase degree, and therefore, efficiency of heat and mass transfer processes. For proper evaluation of heat exchange processes intensity, it is necessary to provide a reliable method for determination of the number of these centres, dispersion conditions (homogenization), and thermodynamic conditions for formation of these centres.

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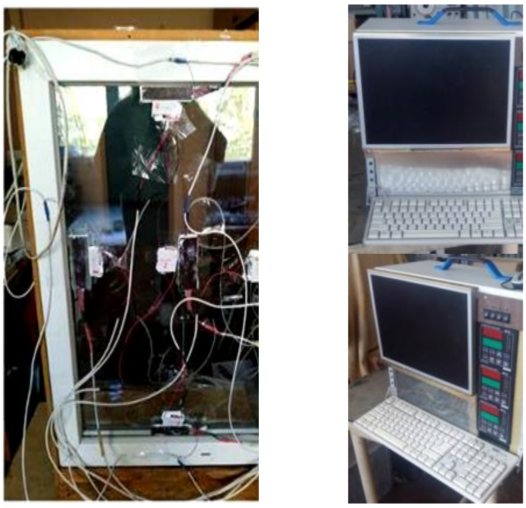
INNOVATIVE METHOD OF IMPROVEMENT OF TRANSPARENT STRUCTURE BY USING ELECTRIC HEATED GLASS

Boris Basok[1], Boris Davydenko[1], Anatoliy Pavlenko[2], Svitlana Goncharuk[1]

[1]Institute of Engineering Thermophysics National Academy of Sciences of Ukraine;
[2]Kielce University of Technology, Poland

Introduction. Housing and communal economy occupies a leading position in terms of energy consumed in the country. Issues of improving the energy efficiency of final energy consumption, namely in residential and office buildings are quite relevant today. In addition, the issue of creating and maintaining a comfortable heat and humidity regime in the premises, which depends on the thermal characteristics of heating and ventilation systems, as well as on the thermal insulation capacity of enclosing structures, becomes important. The temperature regime of the heated room is formed as a result of interaction of aerodynamic and heat exchange processes which accompany heat transfer from the heating device to external air space through enclosing structures of the building. An effective way to reduce heat loss through the enclosing structures of buildings is to replace the old window structures with low heat transfer resistance in modern windows on energy-saving glass. An innovative direction is the use of windows with additional electric heating.

Results. In order to study measures to improve the window structure at the Institute of Engineering Thermophysics of NAS of Ukraine, an experimental stand was created to determine the factors to increase the energy efficiency of translucent structures [1]. In particular, appropriate studies were conducted to establish the features of heat transfer through a translucent enclosing structure with electric heating (fig. 1 a). To this end, two types of heat flux sensors (semiconductor and thermocouple-battery type) and digital semiconductor temperature sensors have been installed at specific locations on the surface of the object. The indoor air temperature was also recorded in different places of the room.



a) b)

Fig. 1. Experimental installation for the study of heat transfer through a translucent enclosing structure with electric heating (a) and a 48-channel thermal recording unit based on semiconductor heat flux sensors (b)

To measure these thermal characteristics, a specially modified portable thermal registration unit was used, the appearance of which is presented in Fig. 1b. The main feature of this unit is the complete set of three secondary devices UKT-38 and four secondary devices Expert with the appropriate number of heat flow sensors of different types and temperature sensors, with connections and data adapters combined in one measuring unit with a special case. With the help of the thermal registration unit, experimental studies of the heat transfer of the translucent enclosing structure with electric heating were performed. Using heat flow and temperature sensors, the distributions of heat flux densities and temperatures on the glass surfaces at different set thermostat temperatures (20, 25, 30, 35, 40 and 45 ºC) were obtained (fig. 2-3).

In Fig. 2 shows data on the comparison of heat flux density obtained using thermocouple-battery type sensors (1, 2) and semiconductor heat flux sensors (1k, 2k). The change in the temperature characteristics of the surfaces of a window with electric heating, which was compared with the set temperature of the thermostat, is presented in Fig. 3.

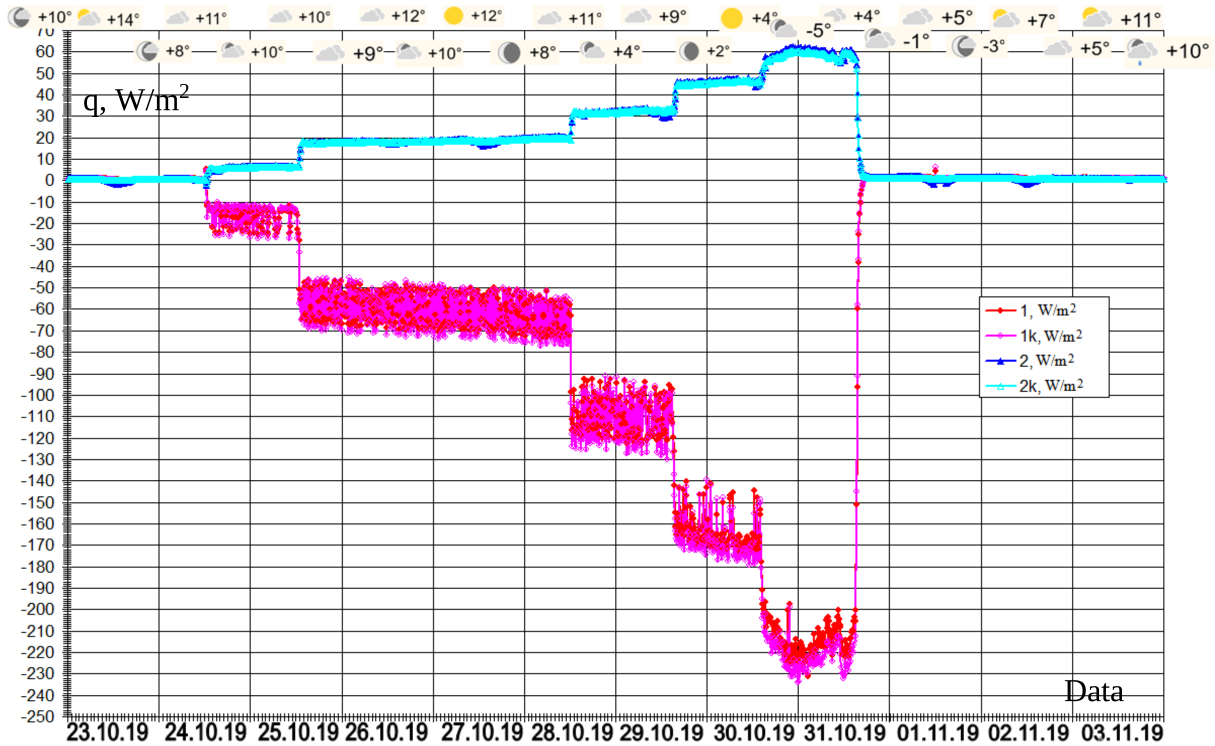


Fig. 2. Dependence of heat flux densities on time at different modes of operation of a windows with electric heating (the sensors are located in the center of the glass)

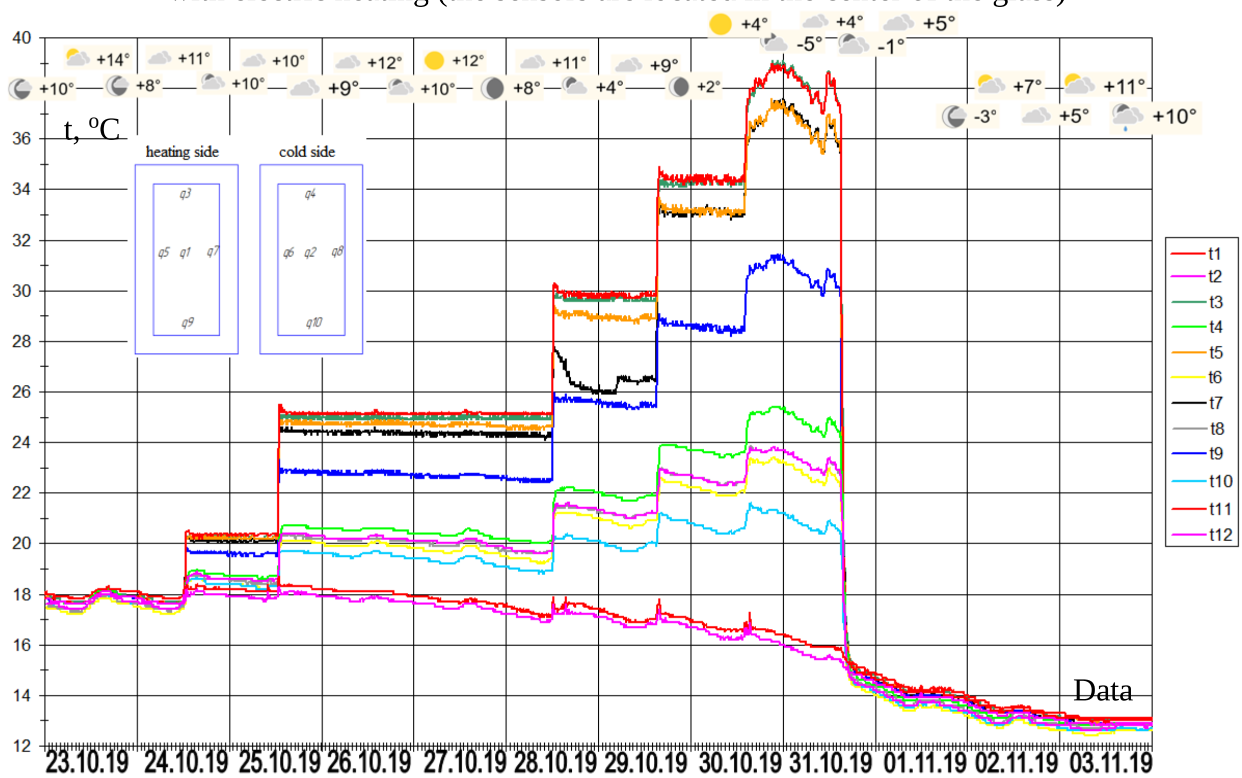
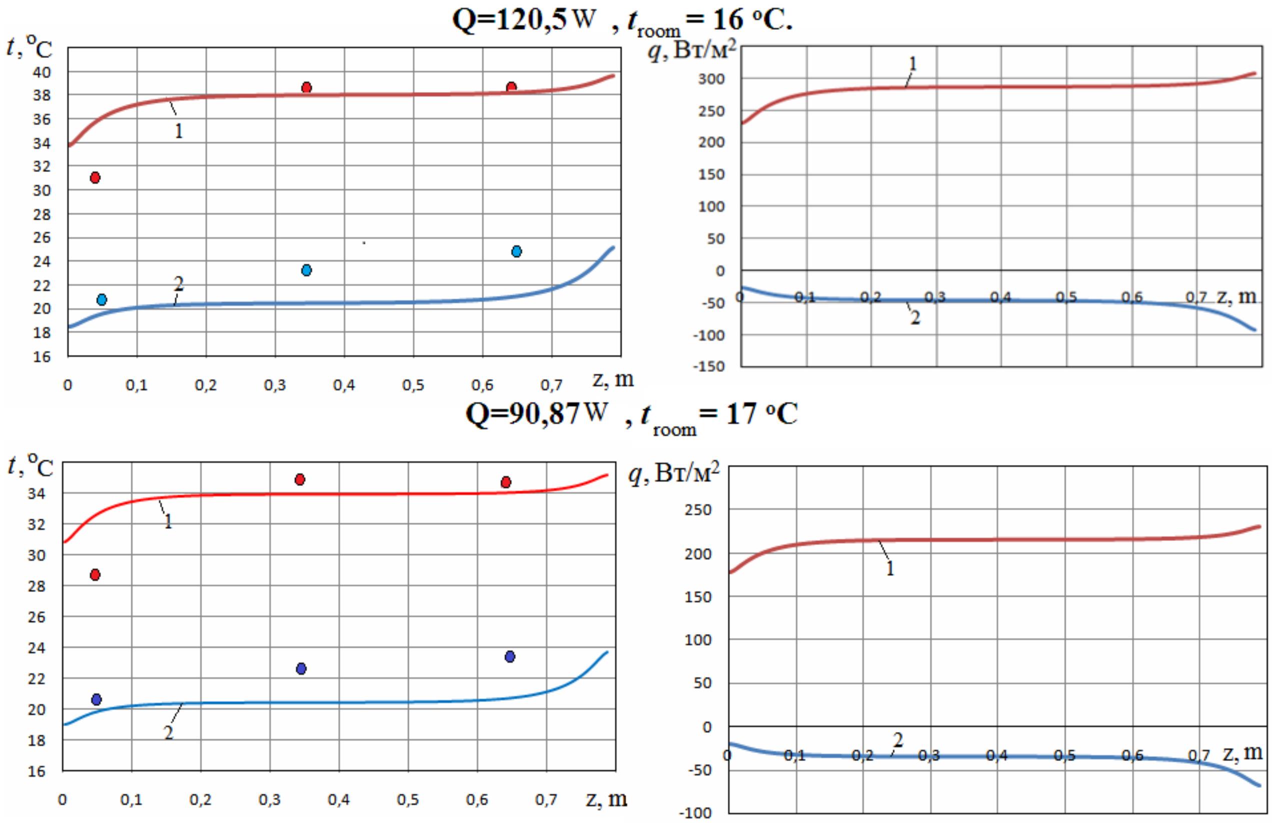


Fig. 3. Dependence of temperature on time at various modes of work of a window with electric heating

An analysis of heat input from the windows with electric heating at a given voltage and current was also performed. Operating modes at different fixed temperature of the thermostat which is built in a window are established.

In addition, the obtained experimental studies on heat transfer through the windows with electric heating were validated using thermophysical modeling. An appropriate mathematical model was developed. Data comparing the obtained theoretical and experimental data are presented in Fig. 4.



a) b)

Fig. 4. Validation of experimental temperature distribution data (a) and heat flux densities (b) on glass surfaces: 1 – a heated glass, 2 – opposite a glass without heating

Conclusions.

Experimental studies of heat transfer of a translucent enclosing structure with electric heating using heat flow and temperature sensors have been performed. Physical modeling of the heat transfer process through a window with electric heating is carried out.

The use of translucent window constructions with electric heating makes it possible to use them as a backup heating system of the building and create a comfortable temperature and humidity conditions in the room by heating the area of the room near the outer enclosing structure.

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NUMERICAL MODELING OF SOLID AND GASEOUS FUEL COMBUSTION IN THE TP-14A BOILER FURNACE TO REDUCE PCDD / F AND GREENHOUSE GAS EMISSIONS INTO THE ATMOSPHERE

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The issues related to the emission of atmospheric pollutants during the provision of energy supply services and the circulation of household waste in settlements are considered. The ways of air pollution by toxic compounds and the formation of greenhouse gases with existing methods of waste heat treatment are analyzed. The issues of reducing the content of toxic emissions in combustion products are studied on the basis of a numerical experiment by means of computational fluid dynamics (CFD). The combustion processes in the power boiler TP-14A (E 220/100) are considered and adequate boundary conditions for the processes of aerodynamics, heat transfer and combustion of gas fuel are determined. The temperature, velocity and concentration fields in the furnace of the investigated boiler have been determined. According to the results of the calculations performed, the formation of chemical underburning and nitrogen oxides is predicted.

Urban air pollution is a major risk factor for public health. The negative impact on the state of atmospheric air is the result of the constant interaction of people with the environment. A wide variety of sources contributes to the loss of air quality. These are transport, energy, industry and the entire complex of housing and communal services - residential complexes and consumer services for the population, down to the smallest. It is believed that from stationary sources objects of large energy and municipal infrastructure (sewage treatment plants and MSW landfills) of the given settlement make the largest contribution to urban air pollution. The main pollutants in terms of mass emission (excluding greenhouse gases) are sulfur dioxide SO_2, nitrogen oxides NOx and carbon monoxide CO. They account for up to 80%, the share of other homogeneous pollutants is less than 10%, the rest is solid suspended particles.

THE INTENSITY OF SOLAR RADIATION FORECASTING BASED ON ARTIFICIAL NEURAL NETWORKS

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Introduction. Due to the problems of global warming (Basok, B., et al., 2020) a large number of countries are trying to increase the quantity of renewable energy sources in their energy strategies. However, one of the biggest complications in such systems operation is their dependence on weather conditions. To operate systems based on renewable sources, such as the solar and wind power, short-term forecasting is required for the timely inclusion of the electrical system balancing.

Recently, in addition to traditional modelling methods, artificial neural networks have been used to solve these problems. Artificial neural networks, as a tool for modelling and forecasting, are widely accepted as an alternative way of solving complex and undetermined problems. The modelling technique with using artificial neural network offers a solution for developing a more generalized model for predicting large arrays of various experimental data, for instance, using climatic and meteorological parameters. In fact, the real object modelling of the surrounding world is accompanied by significant difficulties arising at the stage of problem statement (Novotarskii M.A., Nesterenko B.B., 2004).

In the scientific literature, one can find articles which offers artificial neural networks as a tool for renewable energy forecasting, such as a solar radiation (Fermín Rodríguez, et al., 2018, Hameed W.I., et al., 2019), a wind (Ümmühan Başaran Filik and Tansu Filik, 2017), an aggregated electricity demand (Hernández Luis, et al., 2013), etc. Besides, the review of articles on the artificial neural networks use in renewable energy, available before the year 2000, is given in (Soteris A. Kalogirou, 2001).

The prediction method based on artificial neural networks allowing short-term (10 min) predictions of the solar radiation intensity is presented in (Fermín Rodríguez, et al., 2018). An artificial neural network trained on the Euskalmet database is described in this work. It is proved that artificial neural network prediction accuracy of the solar radiation intensity is reliable and can be used to instantly control the microgrid.

Results. The monograph (Basok B.I. and Veremiichuk Yu.A., 2018) is devoted to the estimation of the solar electric power resource potential in Odessa region. Among other issues the monograph presents insolation measurements in Odessa region, modern models of insolation, and the results of insolation calculations for conditional areas of Odessa region.

This work is aimed at a short-term forecasting attempt (1 hour) of the solar radiation intensity in the city of Odessa based on artificial neural network which was trained on the experimental data of a ground weather station.

The techniques used in this work included four main stages: monitoring, modelling, validation and estimation.

The monitoring was carried out with the help of Davis 6162EU meteorological station, installed on the roof of the main educational building of Odessa National Polytechnic University (Kravchenko, V.P., et al., 2018). Data on the values of solar radiation intensity in 2016 were analysed (from 01.01.16 to 31.12.16). Data were recorded with 1 hour interval.

Since the aim of the work is a short-term forecasting of solar radiation intensity, the intensity of solar radiation in the previous 24 hours was set as model parameters. As solar radiation intensity varies depending on the season, one of the parameters, the array of data was divided into seasons. The data were marked as follows: (1) - winter, (2) - spring, (3) - summer, (4) - autumn.

Besides, since summer has got a longer day compared to winter, an analysis of the correlation between the length of the daylight hours and insolation during the day was performed. Approaching the correlation coefficient to +1 or -1 indicates the presence of positive or negative linear correlation of data sets. Approaching 0 indicates no linear correlation of the data. There is a positive correlation between insolation and daylight hours. Pearson's correlation coefficient between these two values is 0.85. In this regard, the following input parameter of the artificial neural network divided the day into periods as follows: (1) - night, from 0 to 6 (2) - morning, from 6 to 12, (3) - day, from 12 to 18, (4) - evening, from 18 to 23.

For analysis there were taken 26 values as the input values for the artificial neural network, and only one output value, which is the intensity of solar radiation. The first of the input values was the season, the second one was the period of the day, and the other 24 values were data of the intensity of solar radiation taken from the previous day-normalized to the maximum value of the intensity of solar radiation per season.

Modelling, validation and testing of experimental data were performed using the software package MATLAB (R2016a). The Levenberg-Markwatt model was used in the calculations. One hidden layer and 10 neurons were present in the model. The array of analysed data was divided into proportions of 70%, 15%, 15% for neural network training, validation and testing, respectively

The root mean square error (*RMSE*) given by equation (1) was used in this work to analyse the deviation between real and predicted data.

$$RMSE = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(x_i - x_i'\right)^2}'$$

Where x_i - experimental values, x_i' - values predicted by an artificial neural network, and N is the number of predicted points. The lower the *RMSE* value, the more accurate the data forecast.

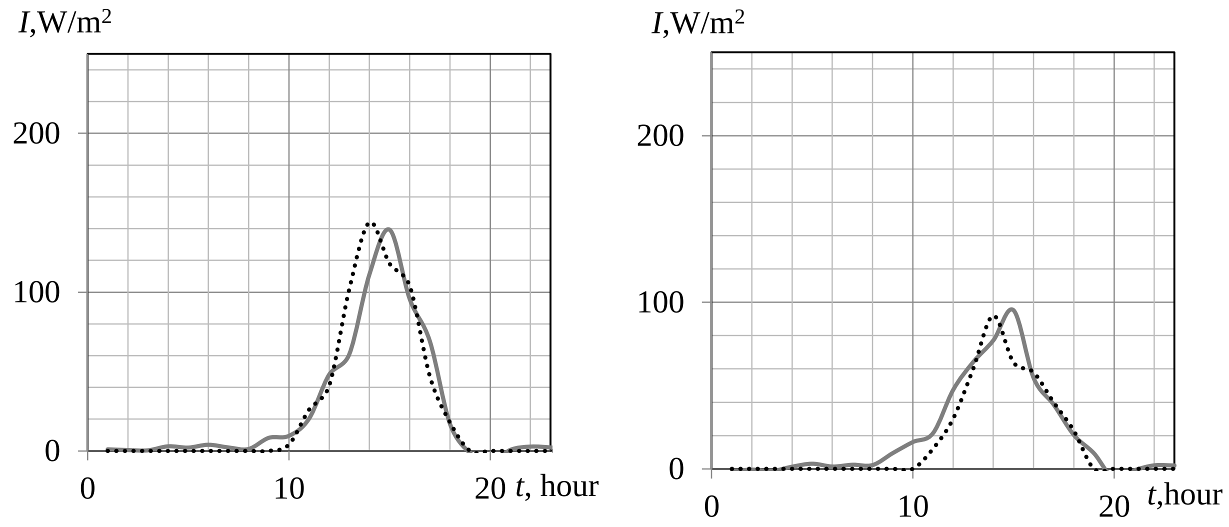

Fig. 1. Forecasting the intensity of solar radiation in the city of Odessa with the help of artificial neural networks *a* - December 26,2016, *б* - December 27, 2016 _ _ _ _ _ - data of the ground meteorological station installed on the roof of Odessa National Polytechnic University (Kravchenko, V.P., et al., 2018); ———— - forecasting with the help of an artificial neural network.

The root mean square error, calculated for the entire array of data, is 46.11W / m^2.

After completing the process of network training and analysis of the deviation between real and forecasted data, an attempt is made to predict the intensity of solar radiation outside the array of data. In fig. 1 it is shown a network forecasting for two days of the year that were not included into artificial neural network training. The root mean square error, calculated for December 26, 2016 was 13.03 W/m^2, and for December 27, 2016 - 9.44 W/m^2. It can be concluded that the artificial neural network predicts with sufficient accuracy even outside the array of data selected for its training.

Conclusions. Artificial neural networks can be successfully used for forecasting renewable energy problems. In particular, forecasting the volume of electricity generation for the following hour is important for sustainable operation of the unified energy system of Ukraine.

Data representing the description of a real system are required for forecasting the parameters based on artificial neural networks. In the formulation of this problem, these are annual meteorological data, which are obtained daily with high accuracy and measurement frequency, for example, one measurement of each parameter every hour, or even more often.

Evaluation of the perfection of the artificial neural network has shown its effectiveness in predicting the intensity of solar radiation, the discrepancy between the forecast and real data for the end of December (the period with the lowest insolation) did not exceed 10%.

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PHOTOSENSITIVITY OF CDTE THIN FILMS

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Light sensitivity is an effective parameter in determining photoconductivity. Photosensitivity is highly dependent on natural and external imperfections, which can act as trapping centers, or as recombination centers. The polycrystalline film contains a large number of grains and grain boundaries. A large number of defects and dangling atomic bonds appear at the grain boundaries, which form additional energy states. These energy states effectively capture charge carriers, thereby creating a potential barrier at the grain boundary. Taking into account the uniform structure of the grain boundary with an average barrier height E_{bd}, the dark conductivity in the film is expressed as [1]:

$$\sigma_D = N_C e \mu_0 e^{-(\Delta E + E_{bd})/kT} = N_C e \mu_0 e^{-\Delta E_D/kT} \tag{1}$$

where N_C is the density of states in the conduction band, μ0 is the carrier mobility in the grain.

Under illumination, the conductivity of the film can increase due to an increase in excess carriers, as well as due to a decrease in the barrier height, depending on the energy of the incident photon. In [1], it was found that the change in conductivity as a result of carrier generation is insignificant in relation to thermally generated carriers, and the conductivity under illumination increases, mainly due to an increase in the carrier mobility at grain boundaries (mobility activation), which arises due to barrier modulation. The total photoconductivity in films under illumination can be expressed as:

$$\sigma_L = N_C e \mu_0 e^{-(\Delta E + E_{bl})/kT} = N_C e \mu_0 e^{-\Delta E_L/kT}, \tag{2}$$

$$\Delta E_L = (\Delta E + E_{bl}) \tag{3}$$

Here ΔE_L - is the energy of photoactivation, E_{bl} - is the height of the barrier under lighting.

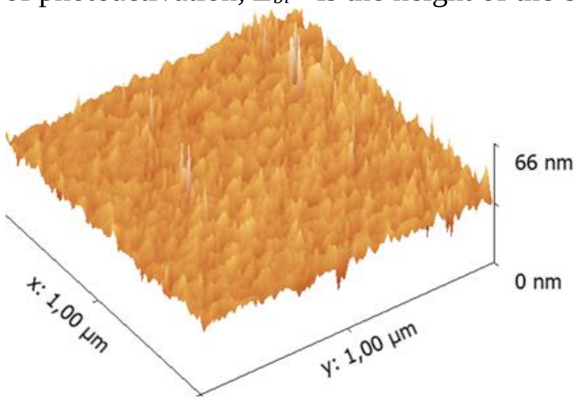

Fig. 1. AFM image of the surface of a 300 nm thick CdTe film on a polished glass substrate.

It should be noted that photosensitivity is highly dependent on structural defects, which can act as trapping or recombination centers. In polycrystalline CdTe films, photoconductivity is mainly determined by processes at grain boundaries. Based on the results of AFM studies, the average crystallite size is determined, shown in Fig. 1 [2].

The decrease in photosensitivity with increasing temperature is explained by a decrease in the photoexcitation process. It is revealed that the lifetime of minority carriers varies inversely with the light intensity and thereby confirms the defect-driven photoconductivity in thin CdTe films.

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EFFECT OF FILM THICKNESS ON MICROSTRUCTURE PARAMETERS OF CDTE THIN FILMS

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The structural perfection of CdTe films substantially depends on the temperature and type of the substrate. Accordingly, the choice of the substrate material and the correlation of technological parameters (deposition time, substrate and evaporator temperatures) can smoothly change the complex of physicochemical and optical properties.

CdTe crystallizes in a face-centered cubic structure of zinc blende with a zinc blende (sphalerite) unit cell size (Fig. 1) of type B3 with tetrahedral coordination of atoms with lattice parameters a = 6.481 Å, d (A-B) = 2.8 Å [48] . Space group - *T*α 2 (F43m), melting point 1092 ° C

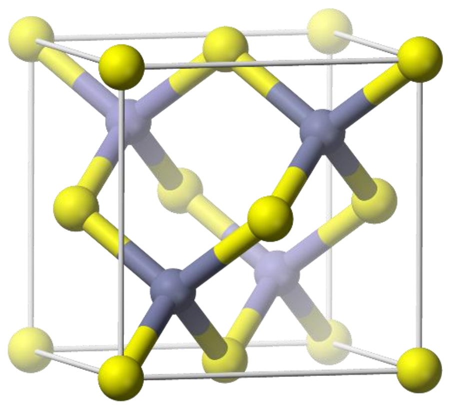

Fig. 1. Zinc blende type CdTe unit cell.

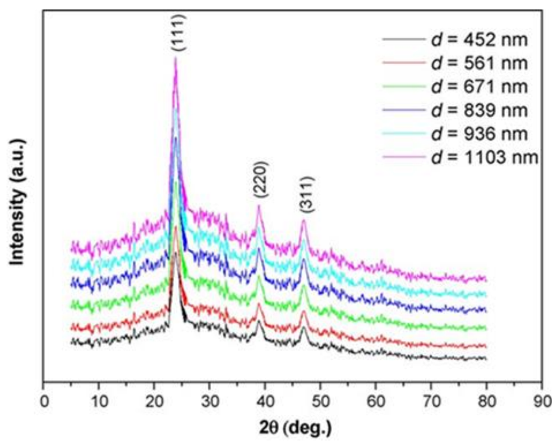

Fig. 2. X-ray diffraction patterns of CdTe films of various thicknesses on glass substrates.

In fig. 2. X-ray diffraction patterns (XRD) of CdTe films of various thicknesses on glass substrates are shown. X-structural analysis showed that the films are polycrystalline with a zinc blende structure with peaks at 2 = 23.80°, 39.40°, and 46.50°, corresponding to the C (111), C (220) and C (311) orientations, respectively. It can be seen that the film thickness affects the XRD of thin CdTe films, that is, the peak intensity increases with increasing film thickness [1].

Each X-ray line XRD profile obtained on a diffractometer expands due to instrumental and physical factors (crystallite size and lattice deformation) [2]. Thus, the first mandatory step in preparing for the calculation of crystallite sizes and grating X-ray scanning is to determine the "clean" profile of the diffraction line for a given image, the width of which depends solely on physical factors [2]. This "clean" line profile is extracted by removing (deconvolution) the instrumental expansion factor from the experimental line profile. Only then can the “clean” line profile be used to calculate the crystallite size and lattice deformation [3].

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IMPROVING THE FLUE GAS CLEANING PROCESS BY UPGRADING THE DESIGN OF THE CYCLONE HOPPER

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The modernization of the cyclone bunker construction is proposed. The idea is to vertically install 2 grids at right angles to each other so that they do not reach the walls and bottom. This will increase the degree of flue gas cleaning by reducing the amount of removal of particles from the hopper.

Environmental protection is an urgent problem today, so ways to clean flue gases before releasing them into the atmosphere or reusing them are gaining ground. Depending on the place of use, conditions and the established ecological norms the process of cleaning of flue gases is developed.

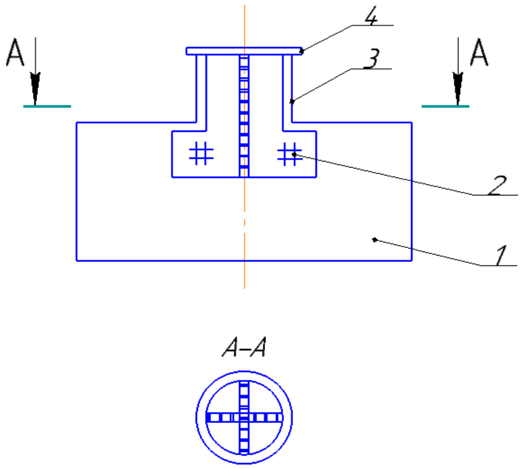
Apparatus for cleaning dusty gas:
- deposition chambers;
- cyclones (separators, multicyclones) and scrubbers;
- filters (sleeve, fabric and electrostatic precipitators)

The most common use of cyclone apparatuses due to their advantages:
- relatively small overall dimensions;
- simplicity of a design;
- ease of operation and repair;
- relatively lower price of the device;
- possibility of use at high temperatures and pressures.

One of the disadvantages of cyclone devices is the reduction of the degree of purification due to the removal of particles from the hopper by the air flow. It is possible to reduce the removal of particles from the hopper by upgrading the design of the hopper. The idea is to install vertically in the pipe and the top of the hopper two grids, which are placed at right angles to each other. This will increase the efficiency of the cyclone [1, 2].

The modernized design of the hopper consists of (Figure 1):

1. hopper body - parallelepiped with closed bottom;
2. vertical grids - installed at right angles to each other so as not to touch the walls and bottom of the hopper;
3. branch pipe;
4. flange - for connection to the cyclone.



1 - the case of the bunker; 2 - grid; 3 - a branch pipe; 4 – flange
Figure 1 – Upgraded cyclone hopper

The process of dust collection is as follows: the particles separated from the flow of dusty gas under the action of centrifugal and gravitational forces, passing between the wall of the pipe 3 and the grids 2 are deposited in the hopper 1. The purified gas flow returns to the cyclone, and due to the grids 2 the swirling flow of air entering the hopper is inhibited by the interaction of air flow and the grid, and rotates, resulting in reduced flow velocity, ie its turbulence during reversal, which, as a result , reduces the number of particles removed from the hopper.

As a result of the modernization of the hopper, the number of particles removed from the hopper will decrease, which in turn will increase the efficiency of the cyclone.

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IMPROVING THE ENERGY EFFICIENCY OF SMALL HYDROPOWER PLANTS WITH TUBULAR PROPELLER HYDROTURBINES

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Abstract: The article analyzes the energy efficiency of low-pressure small hydropower plants with tubular propeller turbines and the factors that affect it. It is noted that in real operating conditions, these turbines often operate with very low efficiency - much lower than nominal. The reasons of discrepancy of real efficiency of the turbine to design are established. Ways to increase the energy efficiency of propeller hydraulic turbines are proposed.

Keywords: tubular propeller hydro turbines, energy efficiency, small hydropower, low pressure.

Introduction

Small hydropower is one of the energy sectors of the future. It does not pollute the environment with carbon dioxide and other harmful emissions. Therefore, it must be comprehensively developed in order to achieve the maximum possible volume of electricity production and the highest possible energy efficiency.

On lowland rivers, where it is not possible to create high differences in water levels at small hydroelectric power plants, simple and reliable tubular propeller hydraulic turbines are most often used. Experience shows that their actual efficiency is approximately 20% lower than the design values. As a result, electricity production is similarly reduced. Therefore, improving the energy efficiency of small hydropower plants is an urgent task, the solution of which is devoted to this article.

Purpose and research objectives

The aim of the work is to analyze the energy efficiency of tubular propeller hydraulic turbines of low-pressure small hydropower plants.

To achieve this goal, the following tasks should be solved:

1. Identify possible causes of reduced energy efficiency of propeller hydraulic turbines (PHT);
2. Suggest possible ways to increase the operational values of PHT efficiency.

Material and research results

On lowland rivers with relatively low water flow, it is most appropriate to build small hydropower plants with tubular propeller turbines. Their advantages are high reliability, speed and efficiency. The main disadvantage is the inability to work with water consumption, which is less than nominal, but it can be minimized through the correct selection of sizes of hydraulic units for each individual HPP. The blades and guides of such turbines are stationary (they cannot rotate relative to their own axes). In Ukraine, hydraulic machines of this type are produced, in particular, by Minihydro LLC (Kharkiv). There are also tubular hydraulic turbines on the market with single (you can change the angular position of either guides or blades) and double (you can change the angular position of both guides and blades) adjustment. Their disadvantages are a much higher cost compared to unregulated turbines, which in many cases makes small hydropower plants with such hydraulic machines unprofitable.

According to the passport data of hydraulic turbines of the vast majority of manufacturers, the nominal efficiency of a tubular propeller hydraulic turbine with an impeller diameter (ID) of 90 cm is approximately 92%. As the ID decreases, the efficiency of such a turbine also decreases, but even for a 30 cm ID it is not lower than 85%. However, in practice the situation is much worse. As a result of experimental research and further processing of their results, we found that the actual efficiency of most propeller tubular hydro turbines at small hydropower plants in Ternopil and Khmelnytsky regions does not exceed 70%. This is the reason for underproduction of electricity and low economic performance of these hydropower plants.

There are two modes of operation of hydraulic turbines:

1. Maximum efficiency mode;
2. Maximum performance mode.

It is unknown what mode of operation this or that propeller hydraulic turbine is set to. This is almost never mentioned in the turbine's passport. Due to global warming and the resulting lack of water resources, all propeller turbines should be set to maximum efficiency. In existing machines, this is difficult but possible.

Other possible reasons for the reduction of operational efficiency of propeller tubular hydraulic turbines in comparison with the design:

1. Wrong length and/or wrong angle at the top of the suction pipe;
2. Insufficient depth of immersion of the suction pipe in the discharge channel of the HPP;
3. The distance between the lower end of the suction pipe and the bottom of the discharge channel of the HPP is too small;
4. Too large a gap between the end surfaces of the impellers and the inner surfaces of the walls of the turbine to which they adjoin (according to existing rules, its value should not exceed 0.1% of the diameter of the impeller);
5. Incorrect blade profile and installation angle;
6. Incorrect guide profile and installation angle;
7. The discrepancy between the angles of installation of the blades and guide vanes;
8. Incorrect speed of the impeller (taking into account the above, this frequency should be equal to the optimal speed of the turbine impeller, provided the turbine operates in maximum efficiency);
9. Leaks in the walls of the suction pipe and suction tract of the hydro turbine, through which external air can penetrate into them.

The Department of Electrical Engineering of Ternopil Ivan Puluj National Technical University offers small hydropower companies cooperation in diagnosing propeller tubular turbines and, if necessary, in increasing the efficiency of these turbines to passport or similar values. Such cooperation will become the foundation for new scientific developments in the direction of improving the energy efficiency of small hydropower.

Conclusion. The study evaluated the energy efficiency of low-pressure small hydropower plants based on tubular propeller hydraulic turbines. As a result, it was found that the operational efficiency of these turbines is approximately 20% lower than the design. A list of possible reasons for reducing the efficiency of these turbines has been developed and analyzed. Ways to increase the energy efficiency of propeller hydraulic turbines are proposed.

MODERN TRENDS IN THE PASSIVE CONSTRUCTION

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Institute of Engineering Thermophysics of the National Academy of Science of Ukraine

Since the development of the passive house standard (1988) [1], the Passive House Institute (Darmstadt, Germany) has introduced new categories for the certification of energy efficient buildings of the passive type: «Passive House Classic», «Passive House Plus» and «Passive House Premium», focusing on renewable primary energy criteria. Energy consumption for heating a passive house still cannot exceed 15 kWh/(m^2a). This criterion remains unchanged for application. But instead of the amount of primary energy consumed, the amount of energy from renewable energy sources is used: for the category of classic passive house this value is 60 kWh/(m^2a), for a passive house plus - 45 kWh/(m^2a), passive house premium - 30 kWh/(m^2a). The number of certified homes is growing every year; there is a database for passive houses on the Internet. The database includes projects of ready-made passive houses and houses under construction [2]. Open days are held [3], when everyone can visit passive type houses and get first-hand information about the living experience, comfort level, heating costs and features of other home systems (in a pandemic, events are held online). Consider some examples of passive buildings (the values of the main indicators are given in the table 1).

1. ID: 5202. The 26-storey apartment building in Cornell Tech, New York, USA is part of the Cornell University campus, located on Roosevelt Island in Manhattan, has 352 apartments for graduate students, teachers and staff.

2. ID: 6397. 22-storey apartment complex (New York, USA) offers residents 140 apartments. Apartments range from studios to two-bedroom. The complex consists of two separate sections. The building is partially based on a monolithic slab and partly on the ground floor.

3. ID: 6234. The 30-storey certified passive building in Tianjin is the tallest high-rise residential building ever built to the standard of a passive house in China. The house has 120 apartments. Each apartment has decentralized ventilation systems with heat recovery, which, if necessary, can be operated by users.

4. ID 6225. Passive house plus on the left bank of the Seine, accommodates the administration of the metropolis of Rouen-Normandy (France). It is the largest public building that has ever had a passive house certificate in France. The building has a special shape and decoration, inspired by the paintings of Claude Monet. It rises to six floors. The colorful facade of glass "scales" has a rainbow ebb. At the roof level, glass gives way to solar panels of different shades. The double facade protects rooms from the sun's heat. In the middle of the building, there is a kind of "gorge" that supplies the interior with natural light.

5. ID: 6158. Reconstruction of a residential building of the 60s according to the passive standard, Colmar (France). Part of the building destroyed, and the rest was rehabilitated, receiving 48 certified passive residential premises. On the roof, there are solar thermal collectors for hot water production.

6. ID: 4329. New building, certified as passive house, Innsbruck, Austria. Residential building and nursing home for 118 places and 23 underground parking spaces. The project is characterized by outstanding quality of architecture. Its feature is its location in the Olympic Village, among the existing buildings, trees, greenery and the embankment of the Inn River, over which the building almost floats on filigree supports.

7. ID: 5948 Passive House Plus certified 6-storey university building located in Roubaix, France. It has 2 auditoriums, a large library and about a hundred rooms. This building is designed to accommodate 1,300 students.

8. ID: 1539. Residential house "House of the Sun", Kyiv, Ukraine. Currently, only one object from Ukraine is present in the database of passive buildings. This is the well-known "House of the Sun" designed by architect T. Ernst. Extremely small plot of land (250 m^2) and the idea of energy saving led to the creation of a very compact, square floor plan (9x9 m)

9. ID 5039. The world's first passive house that moves on the lake and rotates, built on the shores of Lake Weissensee in Carinthia (Austria). Because the orientation of the building significantly affects the microclimate in the room, the house monitors the sun from morning to evening to ensure a comfortable microclimate in the room without excessive use of energy. The advantage of the floating structure is that very little energy is required to rotate in the water. Water is a storage tank with almost no friction and wear, which greatly simplifies the task of creating a sustainable, energy efficient, environmentally friendly and at the same time affordable living space, independent of centralized supply systems. This wooden house produces energy.

10. ID 2504. Certified wooden passive residential building in Nelson (Canada, British Columbia). Number of apartments: 3. The building has a large solarium and covered terraces that protect from the prevailing winds. Particular attention was paid to the choice of materials that provide healthy quality indoor air.

11. ID: 5277. Certified passive sports arena construction (44.75 m x 22.50 m and roof height 7 m, Krakow, Poland). You can hold sports events in various sports: handball, volleyball, basketball, tennis, wrestling, gymnastics, badminton, judo and athletics In the central point, in the southern part of the hall, there is a grandstand for spectators for 224 seats and an additional seat for 60 people. In the northern part of the building is a sports arena. Premises that do not require sunlight are located under the podium (ventilation center, air ducts, storage facilities) to minimize the cost of installation as a whole. Inside the hall there is one wall for climbing (length 20 m and height 7 m) and the second - outside (length 20 m and height 2 m). Social rooms, locker rooms, showers and training rooms (60 m^2) and others are located at the bottom of the building.

12. ID: 6282. House (Passive house plus) for one family, Dabrowa Hotomowska (Masovian Voivodeship, Poland). The highlight of the project is the winter garden, which is a continuation of the living room. The house with a winter garden is the first Polish building certified by the passive house plus standard. On the roof is a photovoltaic system with a capacity of 10 kW. The wooden frame building is built using I-beams to reduce thermal bridges.

Table 1.

Position number in the text	1	2	3	4	5	6
Characteristics of the house						
Year of construction	2017	2019	2019	2018	1966	2013
Area, m^2	18426	9258	7992	6400	3552	7406
Annual heating demand, kWh/(m^{2}a)	15	12	15	11	15	15
Primary energy demand for heating, hot water supply, household electricity and auxiliary electricity, kWh/(m^{2}a)	120	122	116	114	103	174
Air tightness, n50/hour	0,1	0,6	0,1	0,64	0,54	0,41
Renewable energy production, kWh/(m^{2}a)	-	-	-	239	-	-
Position number in the text	7	8	9	10	11	12
Characteristics of the house						
Year of construction	2018	2008	2007	2013	2014	2016
Area, m^2	6183	328	104	202	1666	210
Annual heating demand, kWh/(m^{2}a)	12	15	8	14	14	15
Primary energy demand for heating, hot water supply, household electricity and auxiliary electricity, kWh/(m^{2}a)	42	50	-	119	106	53
Air tightness, n50/hour	0,43	0,6	0,4	0,24	0,18	0,22
Renewable energy production, kWh/(m^{2}a)	58	-	-	-	-	65

The Institute of Engineering Thermophysics of the National Academy of Sciences of Ukraine developed the concept of creating an experimental energy efficient building and built it [4]. The house was created as a scientific, technical and technological thermophysical laboratory with the consistent implementation of the following chain: high energy efficiency house (75 kWh/m^2a) - passive house (15 kWh/m^2a) - zero energy house - smart building - a house as a micro smart-grid system. The total area of the experimental building is 306 m^2. The specific heat consumption of the house is 14.3 kWh/(m^2a). Estimated values of thermal power of systems: heating - 2.6 kW; hot water supply - 3.4 kW; supply and exhaust ventilation - 5.7 kW.
Renewable energy sources are widely used: soil heat, solar energy, wind energy. An automated measuring system has been created, which includes automated continuous measurements of temperature fields, heat fluxes, humidity, pressure, external climatic parameters, etc. Sensors and measuring devices placed in building constructions, in the premises, in the surrounding soil and air.

Conclusions

- The number of certified buildings is growing every year. The reason for this success is that the standards of the passive house are clearly defined, work in all climatic zones, are environmentally friendly and provide minimal energy consumption.
- The passive type building standard is the basis and component for other types of energy efficient construction, such as energy efficient buildings of the «zero energy» type (and close to them), energy active houses, ecological houses, green houses, «smart» houses, etc.
- More and more countries around the world legislate the standards of passive construction of new buildings and reconstructions of old houses.
- Passive type houses have a variety of architecture and purpose.
- The investment component spent on construction to comply with passive standards is decreasing.
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WASTE HEAT RECOVERY IN INTEGRATED ENERGY PLANT FOR FOOD TECHNOLOGY

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Abstract. Efficiency of integrated energy plant on the base of cogeneration gas engine modules with absorption lithium-bromide chiller for combined energy supply of the factory "Sandora"–"PepsiCo Ukraine" is analyzed. The reserves of reducing the heat exhausted into atmosphere are revealed and their realization through deeper conversion into refrigeration with cooling the engine cyclic air is investigated.

Introduction

The integrated energy plants (IEP) for combined refrigeration, heat and power supply (trigeneration) have growing application. The reciprocating gas fueled engines are used as prime engines. The manufacturers develop gas engines as cogeneration engine modules equipped with heat exchangers for producing hot water or steam by using heat of exhaust gas, intake air and charged gas-air mixture of engines, engine jacket and lubricant oil cooling water.

A widespread application of gas engines in IEP for combined supply of electricity, heat and cold is caused by easy adaptability of cogeneration gas engine modules, equipped with heat exchangers for producing hot water from engine recoverable heat, to the customer's conditions. A cold generated by thermotransformer, in general, of absorption lithium-bromide chiller (ACh), using engine recoverable heat, is applied to cover cooling demands. However, the analysis of the IEP efficiency reveals the presence of large heat losses caused by discrepancy of temperature conditions for efficient performance of ACh and gas engine cogeneration module. So, issuing from the condition of engine maintenance at the appropriate thermal rate providing its reliable operation, the temperature of a return hot water (hot coolant) entering the engine cogeneration system from ACh, is limited to design value of 70 °C. If it excess, a surplus heat is rejected to atmosphere, and at its lower temperature a part of the supply hot water which leaves cogeneration engine module mixes with return hot coolant, passing ACh.

The purpose of this work is increasing the efficiency of conversing the gas engine recoverable heat into a cold by matching the performance of absorption chiller and cogeneration engine module.

Results

The problem of increasing the efficiency of conversing the gas engine recoverable heat into a cold is solved for a IEP for combined energy supply of factory "Sandora"–"PepsiCo Ukraine" (Nikolaev, Ukraine). The trigeneration plant, equipped with 2 cogeneration Jenbacher gas engines JMS 420 GS-N.LC (electric power N_e = 1400 kW, heat power N_h =1500 kW) and a single-stage ACh.

The ACh recovers the heath of engine jacket and lubricant oil cooling water, high temperature charged gas-air mixture and engine exhaust gas to produce a chilled water for technology process cooling and conditioning of air in engine room.

The scheme of the existing system of conversing gas engine recoverable heat into a cold by ACh is presented in Fig. 1.

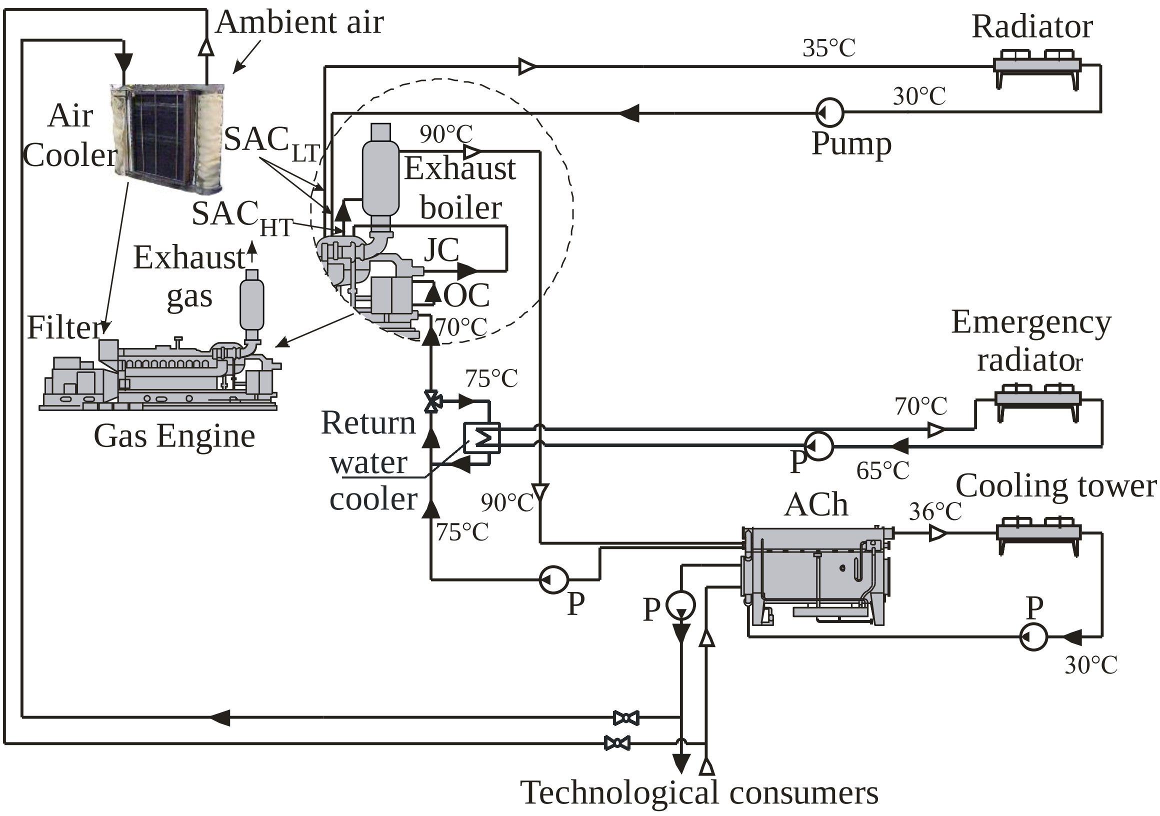
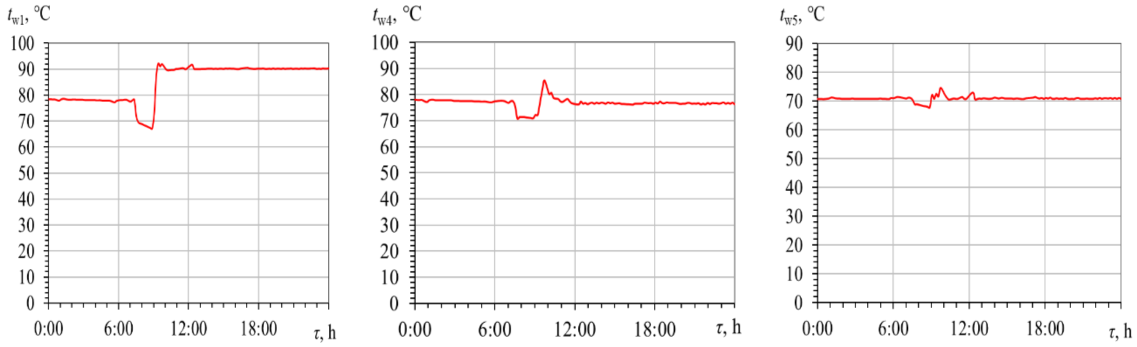


Figure 1. The scheme of the existing system of conversing gas engine recoverable heat into a cold by ACh: OC – oil cooler; JC – jacket cooler; SAC_{LT} and SAC_{HT} – low- and high-temperature scavenge air coolers (of charged gas-air mixture)

According to the scheme in Fig. 1 at temperature of return hot water from ACh t_r is about 75 to 80 °C, i.e. above its design value of $t_{r.sp}$ = 70 °C at the inlet of heat exchangers of cogeneration engine module, providing appropriate thermal engine rate. Therefore, a part of return water is cooled in a return water cooler with rejecting excess heat by emergency radiator into the atmosphere.

In order to estimate a magnitude of excess heat rejected to the atmosphere, and, hence, thermal resources for design improvements by return this waste heat to the cycle of heat conversion to produce an additional cold, the analyses of the data on hot water temperatures t_w, received during monitoring of IEP parameters, have been made. The temperatures of supply hot water t_{w1} to ACh from cogeneration engine module, of return hot water after ACh t_{w4} (before return water cooler for rejecting excess heat by emergency radiator to the atmosphere) and of return hot water, cooled by rejecting excess heat to the atmosphere (after RWC), i.e. hot coolant at the inlet of gas engine module t_{w5} are given in Fig. 2.



a b c

Figure 2. Temperatures of supply hot water t_{w1} to ACh (a), return hot water after ACh t_{w4} (b) and cooled return hot water at the inlet of gas engine module t_{w5} (c)

The share of heat wasted to the atmosphere Q_w caused by necessity to maintain the temperature of cooled return hot water (hot coolant) at the input of gas engine module at the rate of 70 °C, is about 40 % of heat consumption of ACh and actually third of cogeneration engine module thermal capacity 1400 kW.

The innovative heat recovery system with using the excessive heat (normally rejected to the atmosphere) in ejector chiller (ECh) to produce addition cold for gas engine cyclic air cooling was developed and its efficiency was estimated based on monitoring data.

Conclusions. The analysis of conversing the heat available from cogeneration gas engine module into a cold by ACh has revealed presence of considerable losses of heat about 40% caused by discrepancy of temperature conditions of effective performance of the ACh and cogeneration engine module. The innovative heat recovery system with using the excessive heat (normally rejected to the atmosphere) in ejector chiller (ECh) to produce addition cold for gas engine cyclic air cooling was developed and its efficiency was estimated based on monitoring data.

GAS TURBINE INTAKE AIR COOLING SYSTEMS OF COMBINED TYPE AND THEIR OPTIMUM DESIGNING

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Abstract. Turbine intake air cooling (TIAC) by absorption lithium-bromide chillers (ACh) utilizing the exhaust heat is considered as the most effective fuel saving technology for temperate climatic conditions. But the cooling potential of TIAC systems based on ACh of a simple cycle is limited by a comparatively increased chilled water temperature of about 7°C excluding cooling intake air lower than 15°C. The application of a refrigerant as a coolant enables deeper cooling intake air to 10°C or 7°C and lower. The application of two-stage hybrid absorption-ejector chillers (AECh) with a refrigerant ejector chiller (ECh) as a low temperature stage makes it possible to increase the annual fuel saving approximately by 50 % in temperate climate due to deeper cooling air as compared with ACh. Furthermore, this effect can be achieved with the sizes of TIAC system reduced by about 20 % due to determining the optimum refrigeration capacity of AECh providing a maximum rate of annual fuel saving increment and the use of the current excessive refrigeration capacities to cover peaked loads.

Key words: turbine, intake air, two-stage cooling, combined chiller, fuel efficiency.

Introduction. The ambient air temperature makes a considerable influence on the efficiency of gas turbines (GT). The absorption lithium-bromide chillers (ACh) are the most widespread for turbine intake air cooling (TIAC) by using the exhaust heat. But their application is limited by the values of ambient air temperature of about 15 °C because of comparatively raised temperature of chilled water from them of about 15 °C for ACh of a simple cycle. Therefore, their application in temperate climatic conditions is not so prosperous. But they can be applied as a stabilizing boost high temperature stage to minimize the fluctuations of current loading at increased ambient air temperatures above 15 °C. In order to enhance the effect in fuel saving due to TIAC the further subcooling the air, precooled in ACh, might be conducted by the chillers which efficiency is effected by changing thermal loads. Thus, the ejector chillers (ECh) could be applied as a low temperature stage and the most simple in design and cheapest. The application of such combined absorption-ejector chillers (AECh) can be supposed as the novel prosperous trend in TIAC in temperate climatic conditions.

Practically all the generally accepted methods for TIAC system designing issue from the assumption to cover the maximum cooling needs over the full range of yearly operating conditions. Such approach inevitable leads to considerable overestimation of cooling capacities and TIAC systems oversizing that requires to define a correct design cooling load without overestimation.

A reduction of the chiller sizes with maximum annual fuel saving is possible due to defining the optimum design cooling capacity and its rational distribution providing small deviation of current loads from a design value.

The purpose of the study is to develop the advanced combined AECh TIAC systems and the advanced methodology of their designing with rational distribution of the overall design cooling capacity between ACh to cover the unstable thermal load range for ambient air precooling and ECh operating in a comparatively stable load range of further air subcooling, that provides practically twice reduction of a design boost thermal load and about 50% higher annual fuel saving as compared with ACh.

Methodology. A reduction of the chillers design cooling capacity is possible by determining its rational value to provide closed to maximum annual fuel saving as the first step of the methodology for designing the TIAC system and further distribution of the available cooling capacity in response to the current demands as the second step.

The annual fuel saving ΣB_e of the GT due to inlet air cooling is assumed as a criterion to determine a rational design cooling capacity Q_0 of the TIAC system. With this the current fuel reduction B_e have been summarized over the year:

$$\Sigma B_e = \sum(\Delta t_a \cdot \tau)\cdot b_{et}\cdot N_e\cdot 10^{-3}, \text{ t}, \quad (1)$$

where: $\Delta t_a = t_{amb} - t_{a2}$ – current intake air temperature drop, K or °C;

t_{amb} and t_{a2} – ambient air and air temperature at the air cooler outlet, K or °C;

N_e – turbine power output, kW; τ – time interval, h;

b_{et} – specific fuel reduction for 1K (1°C) air temperature drop, assumed 0.7 g/(kWh·K) for UGT10000.

The cooling capacity Q_0 referred to air mass flow rate G_a = 1 kg/s:

$$Q_0 = G_a\, \Delta t_a \quad \xi\cdot c_{ma}, \text{ kW(kg/s) or kJ/kg}, \quad (2)$$

where: ξ – specific heat ratio; c_{ma} –specific heat of wet air, kJ/(kg·K.

Results

A developed combined two-stage TIAC system with AECh is presented in Fig.1.

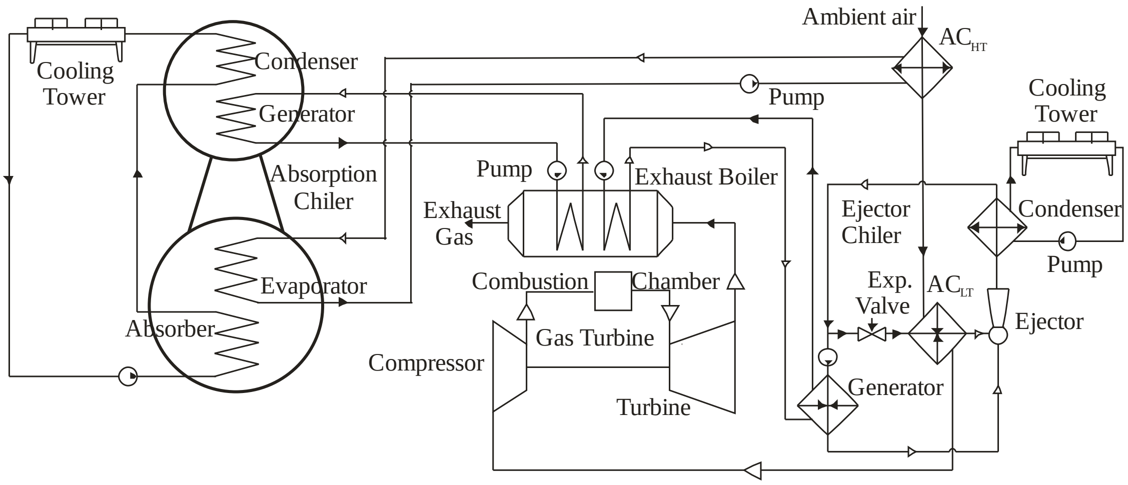
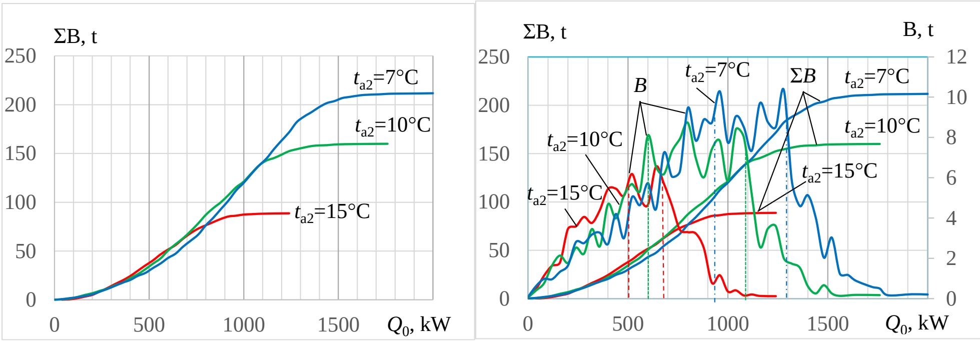


Figure 1. A combined two-stage TIAC system with AECh: AC_{HT} – high-temperature air cooler stage; AC_{LT} – low-temperature air cooler stage; Exp.Valve – expansion valve

According to the method developed the fluctuations of the current effect in GT fuel saving B_e are considered by the rate of their annual increment $\sum B_e$ as relative annual fuel saving increment $\Sigma B_e / Q_0$ referred to design cooling capacity. A such methodological approach makes it possible to increase the accuracy of the calculation results by excluding the approximation of the current changeable values of B_e (Fig.1).

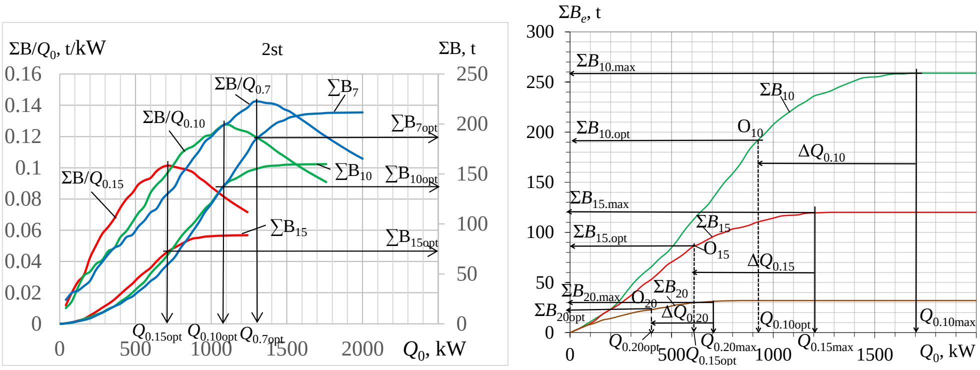
Because of considerable fluctuation of cumulative fuel reduction values B within a wide range of needed cooling capacities Q_0 it is practically impossible to determine its optimum value that would provide a maximum rate of annual fuel saving ΣB increment without TIAC system oversizing (Fig. 2). The calculations are performed for GT UGT 10000.



a b

Figure 2. Cumulative values of annual fuel saving $\sum B$ (a) and current summarized value of $\sum B$ versus cooling capacities Q_0 needed for cooling ambient air at GT inlet to t_{a2} = 7, 10 and 15 °C during 2017 in Mykolayiv region

The optimum design cooling capacity of the chiller $Q_{0.opt}$ provides a maximum rate of annual fuel saving $\sum B_e$ increment $\sum B_e / Q_0$ and minimum sizes of the chiller and TIAC system (Fig.3,a).



a *b*

Figure 3. Annual fuel reduction $\sum B_e$ and its relative values $\sum B_e / Q_0$ referred to design cooling capacity Q_0 (a) and annual fuel reduction $\sum B_e$ (a) for cooling ambient air to t_{a2} = 7, 10 and 15 °C

A maximum rate of annual fuel reduction $\sum B_e$ increment $\sum B_e / Q_0$·for t_{a2} = 10 °C takes place at the optimum design cooling capacity $Q_{0.opt}$ of about 900 kW (Fig.3,a).

As Fig. 3,b shows, optimum designing of TIAC systems provides decrease of installed cooling capacities of the chillers and TIAC systems in the whole by the values of $\Delta q_{0.10,15,20}$, i.e. by 15 to 20 % compared with their maximum magnitudes $q_{0.10,15,20max}$, calculated according conventional practice of designing.

In temperate climatic conditions the application of optimally designed hybride two-stage TIAC systems with combined AECh enables to provide about 50% higher annual fuel saving $\sum B_{10opt}$ at $Q_{0.10opt}$ as compared with $\sum B_{15opt}$ at $q_{0.10opt}$ for ACh (Fig. 3,b) and can be considered as a novel prosperous trend in TIAC.

Conclusions

A novel trend in TIAC by two-stage air cooling in combined AECh is proposed to provide about 50% higher annual fuel saving in temperate climatic conditions.

An advanced methodology is developed to determine the optimum design refrigeration capacity of TIAC systems that provides maximum rate of annual fuel saving increment and decrease of installed capacity by about 20% as compared with conventional designing practice.

FORECASTING THE ELECTRICITY GENERATION OF PHOTOVOLTAIC PLANTS

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Introduction

Since 2019, the electricity market in Ukraine has moved to a new model of operation. Producers from Renewable energy sources (RES) make up an increasing share of the market. According to Ukrenergo [2], in March 2021 the installed capacity of RES in Ukraine was 6.97 GW, of which the largest share falls on solar power plants (SPP) – 5.51 GW, namely 10% of the total installed generation capacity in the country. RES producers sell electricity at a "green tariff", which is the highest among all others in Ukraine. From the beginning of 2021, RES producers began to compensate for the imbalance of electricity in the market a day in advance relative to the day-ahead hourly generation forecast that they provide before the start of the market day.

All these circumstances have increased the importance of the tasks of forecasting the generation of electricity from RES producers, in particular SPP, both at the level of the power system of Ukraine as a whole and at the level of producers.

The day-ahead hourly forecast of SES generation is referred to as short-term forecasts and it is used physical, statistical, and intellectual methods [1]. Among the intelligent forecasting methods, the most popular are models based on the Artificial Neural Networks (ANN) and Adaptive Neuro-Fuzzy Inference Systems (ANFIS), as well as combinations thereof. For example, in [5] and [6] new combined methods are presented, which include ANN and ANFIS for short-term forecasting of SPP generation.

The above publications describe the classical and new methods of forecasting and include assessment of their accuracy, but it is paid not enough attention to the study of the properties of the models themselves, in particular their structure and sensitivity to input error. The latter is especially true for models that use intelligent technologies, such as ANN and ANFIS, where it is impossible to explicitly express the relationship between input and output variables.

The aim of this work is to form an approach to the study of the forecast error dependence on the number of input membership functions and their form, as well as the sensitivity to input data error of the ANFIS-based day-ahead hourly forecast SPP generation model. The sensitivity of the model to input data error is particularly important because the input data for generation forecasting can come from meteorological forecast providers, which indicate the predefined forecast error.

Materials and Methods

In the study it was built a numerical model of the dependence of solar panel generation power on current solar irradiance based on ANFIS [3]. This model can be used for hourly day-ahead generation forecast based on solar irradiance forecast values from weather forecast providers, taking into account the cloud opacity.

The ANFIS methodology is based on a network of special structure that allow you to create and configure a set of fuzzy rules such as Takagi-Sugeno to approximate the relationship between multiple inputs and a single output. The author of ANFIS in [3] showed that it is a universal approximator of continuous functions of several variables defined on compact sets.

Data from the open dataset Photovoltaic (PV) Solar Panel Energy Generation from UK Power Networks from the London Datastore repository were used to train and test the model [4]. The input data of the model – current solar radiation (W/m2), measured at the local weather station, and the output data – the power of the solar panel (kW) from the location of Forest Road. For research, the sample was divided into training and test data.

The research was performed using numerical simulation in MATLAB with the use of Fuzzy Logic Toolbox. The first part of the research was devoted to the choice of the number of input membership functions and their form. In the second part of the study, the sensitivity of the model to the input data error was determined for the selected number and form of membership functions from the first part. The results were evaluated by errors root mean square error (RMSE), mean absolute error (MAE) and normalized mean absolute error (NMAE).

Results and Discussions

It was calculated the table of dependence of RMSE, MAE and NMAE errors on the number of input membership functions and their forms for training, test and the whole sample. The number of input membership functions varied from 2 to 30, and the type of membership functions was chosen from the set (MATLAB notation): gbellmf, gaussmf, gauss2mf, trimf, trapmf, dsigmf, psigmf, pimf.

To determine the optimal number and form of membership functions, the NMAE error of the test sample was used, which was calculated to be in the range from 3.92% to 4.15%. A minimum was achieved on 5 triangular trimf membership functions.

The model was tested for sensitivity to the error of the input data using the chosen optimal number and the form of membership functions. A generated random sequence was added to the input data, which added an NMAE error of 1.81% to 8.19% to the input data. The NMAE error of the initial data in the test sample varied from 4.19% to 5.78%, i.e. the model studies showed a sufficiently low level of variation in the output values relative to the error of the input data.

Conclusions

The study demonstrates an algorithm according to which it is proposed to study the structure and sensitivity of ANFIS-based models for the problem of approximating the dependence of SPP generation power on solar irradiance. In this study, the best type of membership function was trimf, the number and shape of functions had little effect on the result - within 0.23% of NMAE.

Regarding the sensitivity of the model to the input error, it can be noted that for 5 input trimf membership functions, increasing the input error to 8.19% NMAE leads to an increase in the output error in the test sample to 5.78%, NMAE. The rather low sensitivity of the model to the input data error allows us to conclude that it can be used for forecast meteorological data with a pre-known fixed forecast error.

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ANALYSIS OF CONSTANT AND INTERMITTENT HEATING MODES USING BEM AND CFD SIMULATION

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Abstract: *Thermal characteristics of the room in steady-state and transient modes on the basis of CFD model in ANSYS Fluent software environment were analyzed in the present paper. The first part of the study is devoted to the room heat load determination using created steady-state model, and its comparison with EnergyPlus modeling results. In the second part of the work the influence of thermal inertia characteristics of enclosing structures on cooldown and heating-up over time were investigated using transient model. To analyze the heating-up of the room two cases with different radiator surfaces temperatures were studied.*
Keywords: *intermittent heating, BEM, CFD, cooldown, heating-up.*

Introduction

The construction sector is the largest consumer of energy in the world [1]. Energy consumption of the household (residential) sector represents the second-largest share in the energy balance of Ukraine. Through the state support programs, residential energy efficiency improvements are focused primarily on measures related to centralized heat supply and building envelope thermal insulation. Less attention is paid to the organization of the heating mode of local devices, which would provide comfortable thermal conditions and efficient use of energy. To implement the correct heating mode, it is necessary to take into account a large number of controlled and uncontrolled factors, including external and internal factors as well as thermal inertia properties of building envelope constructions over time [2].

Purpose and research objectives

The purpose of the present study is to analyze the building structures thermal inertia in case of the intermittent heating operation mode in the considered room using CFD modeling software.

According to the purpose the following tasks are solved:

1. creation of the room 3D simulation model in ANSYS Fluent software environment;
2. analysis of the room cooldown when the local heating device is not in the operation;
3. analysis of the room heating-up with local heating device in operation using different surface temperatures

Material and research results

Initial data. An existing room in an apartment building was chosen for the research. The room dimensions are 4.5 m × 2.34 m × 2.7 m height; the gross volume is 28.43 m^3. The exterior wall (3.92 m^2) as well as window (2.4 m^2) face east. The interior walls are contacting the other conditioned spaces. Room ventilation is natural and air change rate is 0.56 h^{-1}. The studies were performed at an outdoor air temperature of -22˚C (heating design temperature for Kyiv).

Model description. ANSYS Fluent solver package was used to perform the CFD computations. In considerable number of studies CFD modeling has been used to analyze thermal comfort, indoor air quality, and energy saving potential [3]. Based on continuity, momentum, and energy equations program allows to evaluate the thermal and 3D flow fields [4].

Fig. 1 shows the computational area for CFD modeling.

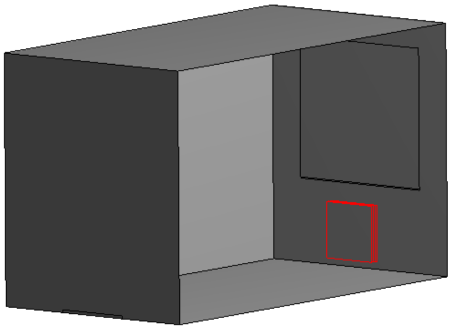

Figure 1. Room 3D simulation model

The heating device (radiator) is located at a distance of 0.1 m from the exterior wall and from the floor respectively, and consists of four surfaces with dimensions of 0.6 m × 0.6 m and a thickness of 2.5 mm. To take into account the possibility of heat accumulation in the radiator, the following conditions were set on each surface:

$$\frac{dt}{d\tau}C = Q\big|_{-} - Q\big|_{+}$$

$$t\big|_{-} = t\big|_{+}$$

$$C = (cm)_{steel} + (cm)_{water}$$

Since the ventilation in the room is natural, we assume that air enters the room by infiltration through leaks in the external enclosing structures. Therefore, in this model, the ventilation inlet is located under the window with the air mass flow rate of 0.0057 kg/s, which corresponds to an air change rate of 0.56 h^{-1}. The air outlet to other rooms is provided through an opening above the floor at the location of the interior door.

For the computational area the following conditions were adopted when conducting simulation:

– thermal conductivity and viscosity of air do not depend on temperature in contrast to density;

– turbulent air flow was taken into account with “k-epsilon” turbulence model;

– radiative heat transfer was taken into account with “surface to surface” radiation model.

The boundary conditions on both window and exterior/interior walls were defined by using convective thermal transfer conditions (t_{ext} = -22˚C, t_{int} = 22˚C α_{ext} = 23 W/(m^2·K), α_{int} = 8.7 W/(m^2·K)). On the interior walls two different conditions were tested: adiabatic and convective. But the second option was chosen to take into account thermal inertia properties of building structures. The heat flow from the radiator was set using temperature on its surfaces ($t_{rad.}$ = 51˚C).

The CFD model was implemented with the following simplification: enclosing structures and radiator surfaces were modeled as a homogeneous equivalent walls (Table 1).

Table 1. Characteristics of the homogeneous equivalent walls

Construction type	Thickness, m	Density, kg/m^3	Specific heat, J/(kg·K)	Thermal conductivity W/(m·K)
Wall exterior (east)	0.47	1271	874	0.535
Wall interior (south)	0.42	1795	878	0.813
Wall interior (west/north)	0.165	1549	840	0.42
Ceiling/floor	0.22	2436	840	1.939
Window	0.038	1000	450	0.0483
Radiator (4 surfaces)	0.0025	7007	946	55,17

Before the start of the transient mode simulation the temperature of radiator surfaces that provides an internal temperature of about 22˚C was investigated under steady conditions. The transient condition was simulated considering the geometric characteristics of the computational area and a time step of 60 seconds was chosen for modeling.

Analysis of the study results. To check the quality of the simulation results, the room heat load in ANSYS Fluent were compared with the simulation results for similar model created in EnergyPlus software environment. So at an internal temperature of 22˚C the heat load according to the results of CFD modeling was 44.75 W/m^2, and according to BEM results was 48.73 W/m^2.

Fig. 2 shows the results of the transient modeling over time. With the beginning of the transient mode simulation heat supply was disconnected and during the cooldown simulation the average mass-weighted temperature in the room has dropped by 5.5˚C in 180 minutes. It was also noted during the study that mean radiant temperature for the first 60 minutes has been lower than air temperature, but with further cooldown it has changed to the opposite.

Two cases with different temperature of the radiator surfaces (55˚C and 65˚C) have been studied in order to analyze the heating-up of the room.

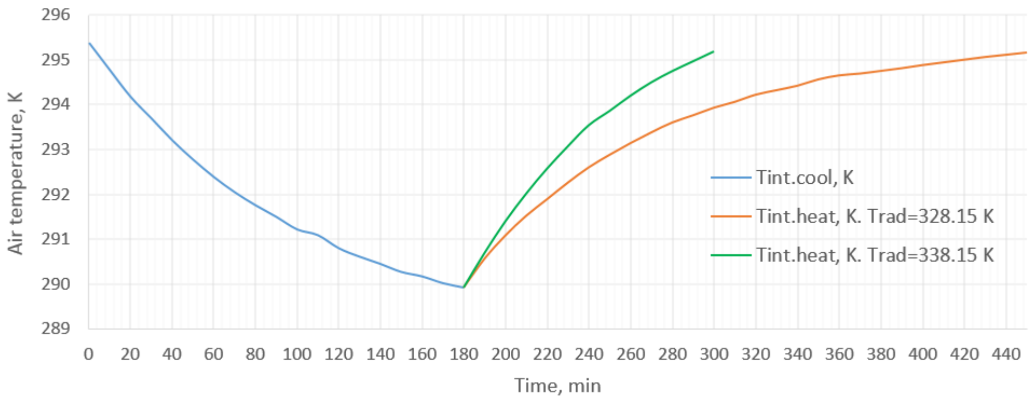

Figure 2. Transient simulation results of internal air temperature

Conclusion

In the study, CFD modeling in transient conditions on the basis of the room 3D model created in ANSYS Fluent software environment was carried out. According to the modeling results the room cools down faster than it heats up, but additional heat capacity of the radiator has a significant effect on the heating-up process. At a radiator temperature of 55˚C the room has been heated up to 22˚C in 280 minutes, but at a radiator temperature of 65˚C it tooks only 120 minutes. Therefore, the results for this room may form the basis for local programming of the controllers. Consideration of other factors such as occupant's activity, dynamics of air exchange and solar heat gains impact may be the purposes of future studies.

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SIMULATION OF FORCED CONVECTIVE HEAT TRANSFER OF A BUILDING FACADE WITH WINDOW RECESSES

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Introduction. The convective heat transfer coefficient (CHTC) on building external surfaces is essential factor for heat losses appropriate calculation. The constant heat transfer coefficient value is widely used for heat flux determining of building and construction envelopes. Although, in fact, it changes along these surfaces and depends on many factors, including the shape and the size of the building, as well as wind velocity and its direction, temperature difference, presence of façade appurtenances, etc. Thus, using constant value of heat transfer coefficient on external and internal building surfaces can cause inaccuracies in building heat losses determining.

The calculation of distribution function of heat transfer coefficient on external building surfaces, based on the solution of wind flow turbulent momentum and energy transport, should be regarded as promising approach. CFD research of stand-alone building and entire array with dense urban area are contained in literature.

1. Along with CFD simulation, experimental investigation methods of the real building and model measurements in wind tunnels are developing simultaneously. Most of the known dependences are primary developed for smooth façade and do not generally include effects of façade appurtenances. For instance, in (Meseret T. Kahsay et al., 2019), high-rise buildings with external elements modifying the airflow pattern on the surface of the building are considered. The result of this study shows that local and surface averaged convective heat transfer coefficient values on the surfaces of each building are dependent on building aerodynamics and forms of the facade appurtenances. A review of the impacts of such an architectural element as a balcony on the indoor environmental quality and energy consumption of dwellings are represented in (Ribeiro, C.et al, 2020)

The purpose of this article was a numerical simulation for office buildings with window insets and a comparison of their aerodynamic and heat transfer characteristics with the same plain wall building envelopes. Reynolds-averaged Navier–Stokes equations with k-ε model of turbulence were chosen as the research technique. The choice was based on a validation process using experimental data from the Architectural Institute of Japan (Yoshie,R. et al.,2007).

Results. **Computational** model validation. The calculations given in (Yoshie,R. et al.,2007) were repeated, and also there was an attempt to establish how strongly the kinetic energy profile at the computational domain inlet affects the integral computational results.

Kinetic energy profiles at different measuring lines are presented in fig 1 b. Experimental and CFD calculated data with experimental inlet kinetic energy profile and its constant value at the inlet of computation domain is compared. As a result, it was found that the use of a suitable value of the kinetic energy constant along the height at the inlet does not significantly increase the calculation error. This circumstance is very important, taking into account the fact that inflow kinetic energy profile is not always known in real conditions. In next simulation task we assumed that kinetic energy was constant above the height and its value was counted using Richards and Hoxey equation (Richards P.J. and Hoxey R.P., 1993).

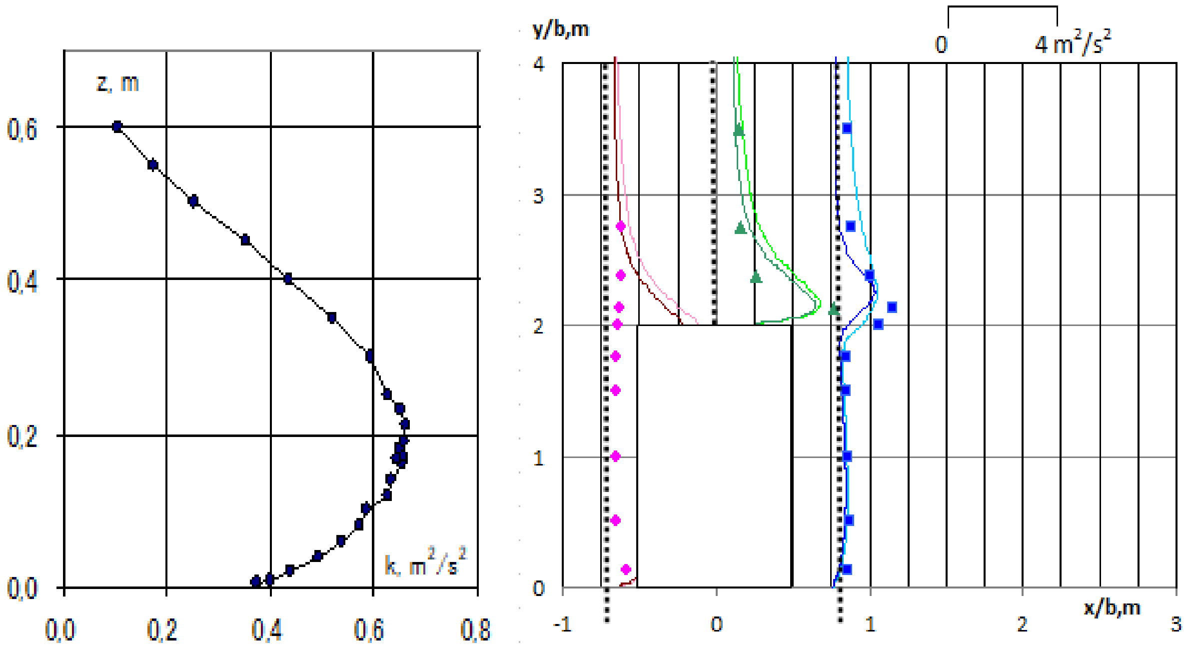

Fig.1. *a* – experimentally measured kinetic energy profile at the computational domain inlet (Yoshie,R. et al.,2007), *b* – comparison of experimentally measured kinetic energy profiles (● – in section x/b = -0.75; ▲ – – in section x/b = 0; ■ – – in section x/b = 0,75) with calculated, at different kinetic energy profiles at the computational domain inlet: (—— —— ——) constant value k=0.48 m^2/s^2; (—— —— ——)- experimentally measured profile at the inlet of computational domain.

Based on the results of the validation, the following conclusions were drawn: 1 – the kinetic energy profile at the inlet of the calculation domain insignificantly affects the calculation results; 2 - The use of low-cost and widely used two-parameter *k-ε* turbulence models is quite possible with the correct setting of the boundary conditions at the inlet of the computational domain; 3- In absence of experimental kinetic energy profile the use of a constant along the height suitable value of kinetic energy at the input of computation domain introduces acceptable error.

Computational results. Comparison of the obtained distributions of heat transfer coefficients on the lateral surface of the facade of a building 60.0 m long with near-window recesses and heat transfer coefficient of the smooth building is presented in fig.2. As can be seen from Fig. 2, with other conditions being equal, the shape of the surface of the building envelope has a significant effect on the heat transfer coefficient. The tendency of increasing the heat transfer coefficients to the edge of the building, from the windward side, is preserved. However, the heat transfer coefficient at the edge of the building becomes lower than that on smooth walls. At the same time, the values of the local heat transfer coefficients in the center of the building increase, on the surface areas outside the window inset. Local heat transfer coefficients inside the window recesses are reduced due to complex aerodynamics near such a surface. The presence of window insets significantly affects the aerodynamics around the building because it leads to additional vortex formation and, respectively, to the redistribution of the flow energy, Fig. 3. The influence of the near-window insets on the aerodynamics of the flow extends to a distance of up to 1 meter outward from the surface of the enclosing structure.

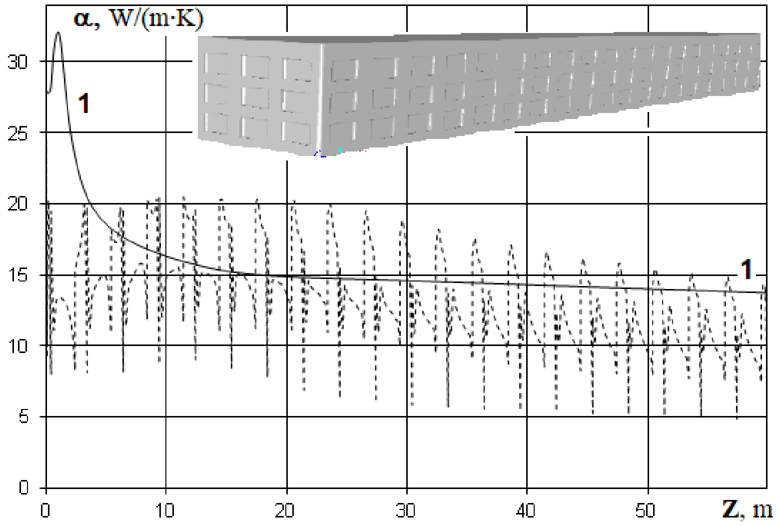

Fig. 2. Distribution of heat transfer coefficients along the facade surface at the height of 5.0 m (half-height of the building) from the ground surface. The building (model insert) is three-storey 60.0 × 18.0 × 10.0 m, along each floor there are 20 evenly spaced windows measuring 2.0 × 1.5 m, recessed into the facade by 0.12 m. Solid curve 1 - distribution of heat transfer coefficients for a smooth building.

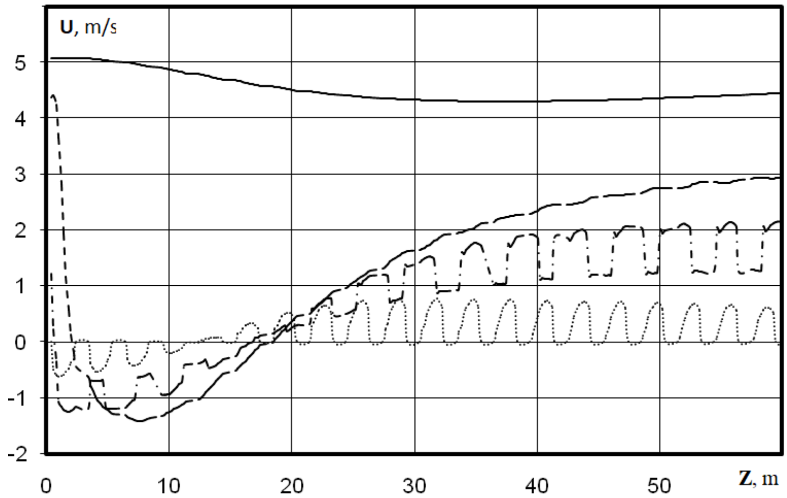

Fig. 3. Dependence of an average speed along the building at a distance from the facade to the outside: × × × × × - 0.01 m; — - — - 0.11 m; - - - - - - 1.1 m; ——— - 11.1 m from the building envelope in a plane parallel to the ground plane at a height of 5 m from its surface.

Conclusions. The presence of near-window recesses window significantly affects the heat transfer from the surface of the building. In the future, it is necessary to investigate the effect of the depth of the near-window recesses on heat transfer and aerodynamics around the building and determine its optimal value for reducing heat transfer from the building surface.

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THREE-DIMENSIONAL NUMERICAL MODEL OF HYDRODYNAMICS AND HEAT TRANSFER IN THE SYSTEM "SOIL – HEAT EXCHANGER – HEAT CARRIER"

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Introduction. An important step in the development of modern heat pump systems for heat supply (including combined) is the choice of the type of heat pump. Studies show that for the climate of Ukraine it is advisable to use heat pumps such as "soil - water". This is explained by the fact that the soil layers below the freezing depth (1.0 ÷ 1.3 m) have a stable temperature, about +8 ^{o}C. Thus, the heat pump uses a source of low-potential heat with almost constant potential, which allows to achieve a high equivalent energy conversion factor (COP ≥ 3.0) throughout the heating period. For combined heat pump systems, two main types of soil heat exchangers are used - horizontal or vertical orientation. The advantage of horizontal shallow groundwater heat exchanges is the natural recovery in the warm period (due to solar insolation and / or passive conditioning) of the temperature state of the soil mass after the operation of the heat pump installation during the heating period.

When using horizontal shallow ground heat exchangers, the main problem is the lack of national standardized methods for calculating their thermal characteristics. Thus, at the moment, such heat exchangers are designed and implemented without thermophysical justification of their optimal operating parameters.

The difficulty is that a cumbersome mathematical apparatus is used to describe the processes of thermal conductivity in an unlimited space. The fundamental equation of thermal conductivity is known to have the form [1]:

$$\frac{\partial T}{\partial t} = a_m \, div \, grad \, T \tag{1}$$

The initial and boundary conditions are added to it, which is the physical and mathematical formulation of the tasks of soil heat accumulation. The main difficulties in solving such problems are that:

1. The solution is in the form of infinite Fourier series, cylindrical Bessel functions, etc. The coefficients included in the series are found by complex mathematical operations and have limitations related to the nature of the change in boundary conditions.

2. At change of boundary conditions that is characteristic of the horizontal soil accumulator, there is no analytical solution of problems of this type.

3. With the accumulation of solar energy (in summer), the operation of solar collectors depends on climatic, meteorological and geographical factors. The heat source has a sinusoidal change in intensity during each day of operation. Such a change in the supply of heat to the ground accumulator creates waves in the environment where it is pumped.

4. In addition, in soil accumulator, heat is distributed in an unlimited space.

Results. These difficulties indicate difficulties in obtaining an analytical solution to the problems of soil accumulation of thermal energy. In the case of variable boundary conditions, taking into account the distribution of heat in an unlimited space, we can conclude that the use of numerical methods is problematic. The Institute of Engineering Thermophysics of the National Academy of Sciences of Ukraine has developed and proposed for use a three-dimensional numerical model of the temperature state of the soil mass during the operation of the soil collector. The model considers a computational domain having the shape of a rectangular parallelepiped with sides x_{max}, y_{max} and z_{max}. At a depth h of the considered area is a horizontal flat tubular coil heat exchanger (soil accumulator) with circulating coolant. The values of x_{max}, y_{max} and z_{max} are chosen so that the heat transfer processes to the soil accumulator have a minimal effect on the temperature conditions at the boundaries of the calculation area. The temperature field of the soil mass is described by the equation of thermal conductivity.

To solve it, boundary conditions must be set on the soil surface ($z = 0$), which depend on the climatic characteristics of the area and a number of meteorological factors. In the general case, these may be the temperatures of soil and air and a coefficient of heat transfer. Such conditions reflect the nature of the thermal interaction of the soil with the air, the temperature of which is constantly changing. Under natural conditions, fluctuations in ambient air temperature, as well as the intensity of radiative heat transfer, lead not only to changes in the surface temperature of the soil T_0 (τ, 0), but also to changes in the law of temperature distribution at its depth T_0 (τ, z). But these changes are observed only to a certain depth, below which the soil temperature can be considered constant (same).

The boundary conditions for the heat transfer equation in the soil massif are met by the condition of no heat input to the calculation area from the sections of the soil massif located outside it. In other words, the collector located in the considered calculation area will "take away" heat only in that part of the soil massif which is limited by the specified area x_{max}, y_{max}. In this case, the solution of the heat transfer problem will meet the condition of the most intensive cooling of this limited area of the soil massif.

Below are the results of calculations of the temperature state of the horizontal soil collector. The horizontal soil collector is located at a depth of 1.65 m on the territory of The Institute of Engineering Thermophysics of the National Academy of Sciences of Ukraine [2-4]. The calculation area has dimensions x_{max} = 17 m, y_{max} = 34 m; z_{max} = 7 m. The total length of the collector pipe is 269 m. The inner diameter of the polyethylene pipe is d = 0.028 m. The wall thickness of the pipe is δ = 0.002 m. The heat carrier is a 30% aqueous solution of propylene glycol. Volume flow of the heat carrier G = 0,756 m^3/h.

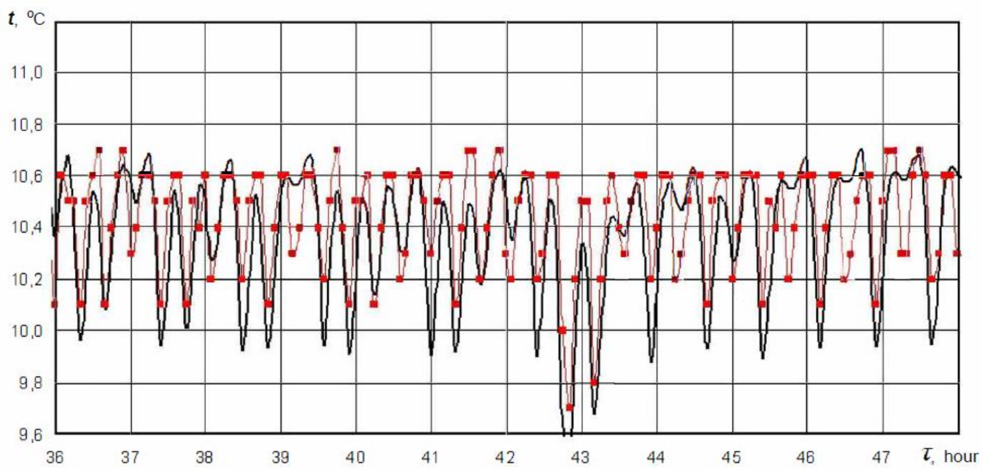

Fig. 1. Change in the temperature of the heat carrier at the outlet of the soil heat exchanger (red - experiment, black - calculation) for the period τ = 36… 48 hours

The graph (Fig. 1) shows that during the operation of the heat pump in combination with a horizontal ground heat exchanger, the temperature of the heat carrier at the inlet and outlet of the collector is constantly changing. However, the results of the calculations correspond to experimental data. Thus, we conclude that this model works adequately and it can be used to calculate the horizontal soil heat exchangers of shallow foundation.

Evaluation of the efficiency of the horizontal soil heat exchangers can be performed on the value of the linear heat transfer coefficient of it. This value is determined from the solution of the above problem in the conditions of stationary heat transfer. It is believed that the temperature of the heat carrier at the inlet to the heat exchanger is constant.

The soil temperature at the boundaries of the calculation area is also constant over time and evenly distributed along the edges of the calculation area. The linear heat transfer coefficient is calculated as the ratio of the amount of heat supplied to the heat carrier in the soil per unit time, to the length of the pipeline and the difference in soil temperatures at the boundary of the calculation area and heat at the inlet to the heat exchanger. The dependence of this value on the flow rate of the heat carrier and the wall thickness of the pipe is presented in Fig. 2.

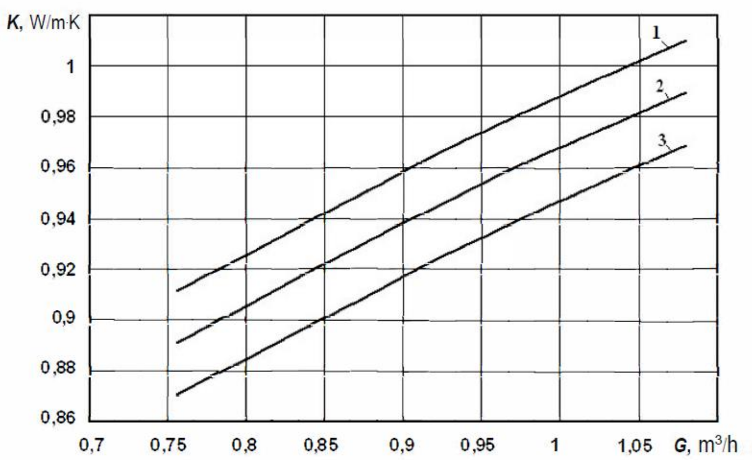

Fig. 2. Dependence of the linear heat transfer coefficient of the horizontal soil heat exchanger on the flow rate of the heat carrier and the wall thickness of the pipe:
1 - δ = 1 mm; 2 - δ = 2 mm; 3 - δ = 3 mm.

As can be seen from the graphs (Fig. 2), the linear heat transfer coefficient increases with increasing heat carrier's consumption and decreases with increasing pipe wall thickness.

Conclusions. The results of calculations according to the numerical model, which describes the operation of the soil horizontal heat exchanger, are satisfactorily consistent with the experimental data. This indicates the possibility of its use for modeling the operation of horizontal soil heat exchangers of arbitrary design in soils with different geology.

With the help of the obtained simulation results it was possible to establish an insignificant influence of the wall thickness of the heat exchanger's pipe on its thermal characteristics. The operating pressure of the heat carrier is crucial for choosing the thickness of the pipeline. The average estimated power of soil thermal energy removal from 1 running meter of the experimental heat exchanger's pipeline was 28 W.

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USING APPS WEATHER FORECAST FOR CALCULATION ELECTRICITY GENERATION BY PHOTOVOLTAIC STATIONS

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Introduction. Solar energy increases it`s impact on countries' energy systems by creating total installed power capacity of photovoltaic stations (PVS), but beside reducing the emissions of coal-fired power plants [1] and increasing the share of "green generation" in energy balance, the problems appear with balancing of energy regimes. Changes in the expected level of outturn of electricity generated into the electricity grid from renewable energy sources (RES) due to weather disasters and anomalies have already led to several accidents, power outages of some parts of consumers. It is important for power system managers to have accurate forecast data on changes in the amount of generation by photovoltaic and wind power plants, so that in case of their discrepancy, they can be quickly replaced by shunting power capacity. A set of input data for the mathematical model can be obtained from PVS monitoring systems, weather applications that forecast the weather and offer a large base of archival data of meteorological parameters.

Depending on the location of PVS, not only the magnitude of insolation changes, but also the typical daily irradiance patterns and cloud cover of daytime (clear sky, instantaneous cloud change, foggy day with limited generation input, high variability due to rapid cloud cover changes), and photoelectric power generation changes need to balance with the power system [2, 3].

Methods and statistical models, as well as time intervals are described in [3,4]. The influence of photomodule temperature, shading, degradation, inverter power, system age are presented in [4,5], so the determination of the coefficients of influence on the generation for specific photovoltaic systems require detailed study.

The results of this study are based on statistical data on the amount of electricity generated at photovoltaic plants in the village of Poberezhia and the city of Tlumach, Ivano-Frankivsk region in Ukraine for the period from 1.04.2020 till 30.04.2021. All calculations are reduced to a power of 1 kW. Meteorological data from the Meteoblue server and the WetterRadar information website were also used [6, 7], which allows the Excel interface to extract data for the project and use it in the LabVIEW virtual devices [8].

The amount of electricity produced by PVS on any day of the year W_{G_i} depends on the length of daylight and the amount of solar electricity that enters the photomodule. The generation curve can be described using the harmonic function (1), as well as using the normal distribution function (2) using formulas 1-2.

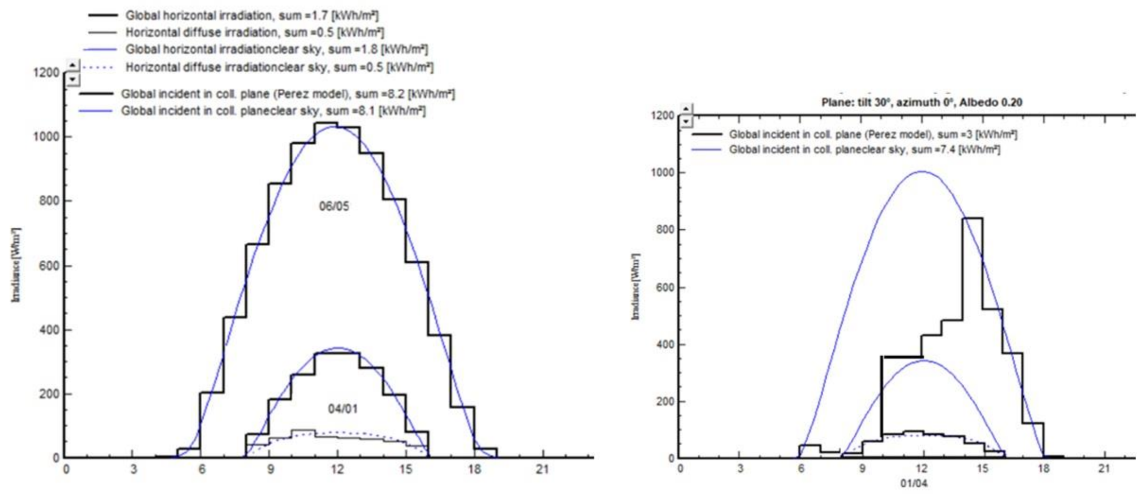
$$W_{G_i} = W_{G0}\sin(\delta+\varphi) \pm k_1 a \pm k_2 b, \quad (1)$$

$$W_{G_i} = W(x,\mu,\sigma) = \frac{1}{\sigma\sqrt{2\pi}} e^{\frac{-(x-\mu)^2}{2\sigma^2}} \pm k_1 a \pm k_2 b, \; -\infty < x < +\infty, \quad (2)$$

Define variables for a given graph:

where $\mu = 179$ - mathematical expectation at value $W_M = 3,6$ кВт, $\sigma = \dfrac{1}{\sqrt{2\pi} W_{G_{MAX}}} = 0,1108$, k_1, k_2 - weights, which take into account deviations from the nominal parameters when changing solar insolation (cloudiness a) and temperature b. Let's make a graph of the dependence of the amount of generated electricity PVS from the day of the year using the normal distribution function (Fig. 1), with the value of the beginning of the graph (X = 178.8) corresponds to the time of daylight growth - December 22, and the range (x = 178,8-179.2) annual period. Assign the days of the year, starting from December 22 (x = 178.8), ordinal numbers m, superimposing them on the value of x. Every day the value of x increases by 0.0011.

Fig. 1 shows an example of a change in the power of the solar system in 1 kW during one day. Due to the increase in the length of the daylight and the amount of solar insolation, the total amount of electricity produced by the PVS increases as a result. When the clouds shade the solar panels, the power of the system drops sharply, respectively, after the cloud cover disappears, the power increases again.



a) b)

Fig. 1. *Change in PVS power capacity in January and May for sunny day (a) and partly cloudy day (b)*

Fig. 2 shows the correspondence of the constructed graph of dependence of the produced electricity of FES during the day (kWh) for April 2021 with the graph constructed according to the forecast data according to the forecast "day ahead" and according to the available values of meteorological data.

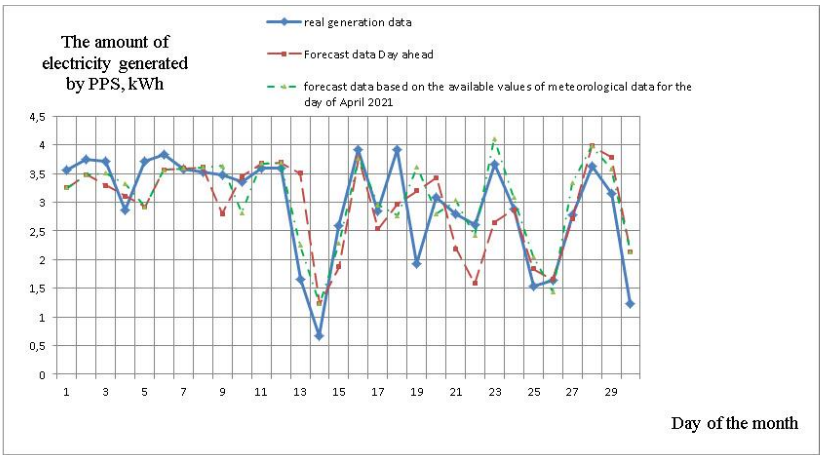


Fig. 2 *Graphs of change generated active power electricity of PVS: 1 - real generation data; 2 - forecast data "day ahead"; 3 - forecast data based on the available values of meteorological data for the day of April 2021*

Conclusions

This article analyzes the possibility of using weather forecast applications to determine the amount of electricity generated by photovoltaic plants. These forecasting tools based on historical measurements will help to more easily calculate the projected amount of electricity generated, which will increase the efficiency of their use compared to complex and expensive forecasting applications.

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HEAT PUMP AND ADSORPTION SYSTEM OF CONSERVATION OF METAL EQUIPMENT

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Inrtoduction. In modern conditions, power equipment operates in shunting mode and has long periods of downtime. It is at this time that the equipment must be protected from corrosion. In the absence of measures to preserve the equipment, its service life is reduced and the financial costs of repair work increase [1].

Humidity is the main catalyst of the corrosion process, and the higher the relative humidity, the faster the destruction of metal parts of the equipment [2]. But at low relative humidity (below 40%) corrosion processes almost stop and do not destroy the metal, so for high-quality preservation of power equipment it is necessary to maintain the relative humidity of the air in contact with the metal below 40% [3].

Results of the research. The main methods of dehumidification are presented in the article [4]. The advantages and disadvantages of all methods of air dehumidification were analyzed and the most effective, to solve the problem of conservation of power equipment, adsorption method of air dehumidification was selected. However, due to the existing shortcoming of the system, namely the high consumption of electricity, it was proposed to improve the system by combining an adsorption dehumidifier with a heat pump system. Thermodynamic analysis of energy efficiency of such a system was performed in [5] under the conditions of theoretical (isoenthalpy) processes of dehumidification of working air and humidification of regeneration air in the adsorption rotor. The actual processes of dehumidification and humidification of air in real designs of adsorption rotors differ significantly from ideal isoenthalpy processes and can be determined for standard rotors by the existing method. This raises the question of the impact of these real processes on the final energy efficiency of the heat pump-adsorption system of air dehumidification for the conditions of conservation of power equipment.

Figure 1 shows the heat pump-adsorption scheme of conservation of power equipment. In this scheme, a silica gel adsorption rotor is used to deep dehumidification of the air, and a heat pump with partial recirculation of regenerated air is used for highly efficient heating of regeneration air, which significantly reduces energy consumption.

Figure 2 shows a general graph comparing the maximum efficiency coefficients for a standard adsorption system and an improved heat pump-adsorption system for two rotor thicknesses - 100 mm and 200 mm. From this graph it is seen that the efficiency of the systems is influenced by: the thickness of the rotor and the parameters of fresh air. It is seen that the greater the thickness of the rotor, the more efficient the systems with and without a heat pump installation. An interesting feature of the systems is the fact that increasing the outside air temperature leads to increased energy efficiency of the standard system and to reduce the efficiency of the heat pump-adsorption system. This is due to the fact that in a standard adsorption system with increasing temperature of fresh air it is need to spend less electricity in the heater, and in the case of heat-adsorption system increase in fresh air moisture leads to increased load on the heat pump evaporator and a corresponding increase energy consumption.

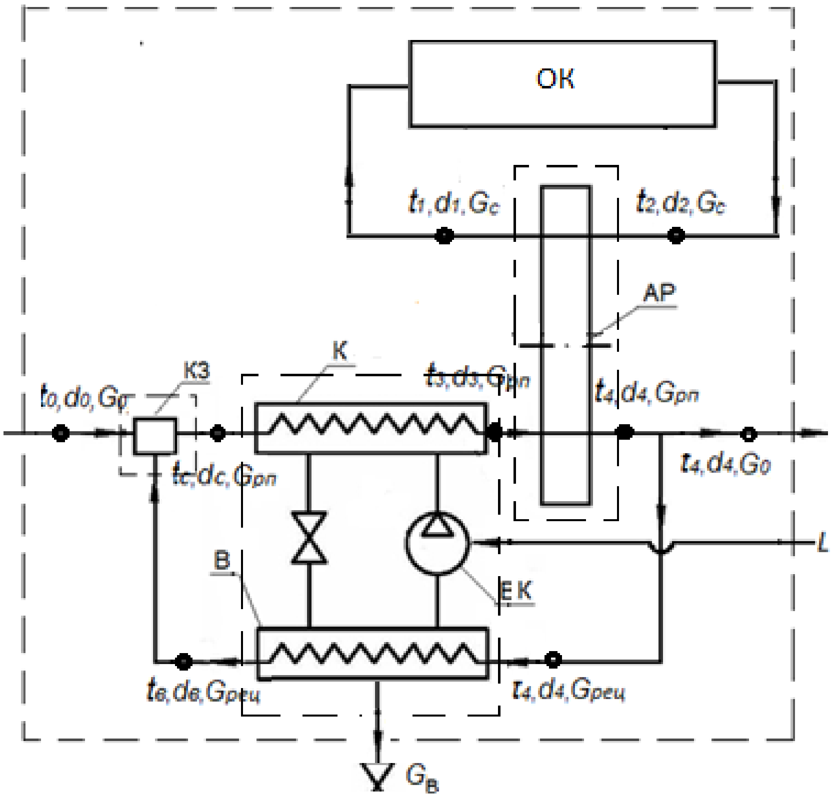

Fig. 1. *Basic heat pump-adsorption scheme of air dehumidification in the system of conservation of metal equipment: K - HP condenser; B - HP evaporator; EK - compressor; AP - adsorption rotor; K3 – mixing chamber; OK - the object of preservation.*

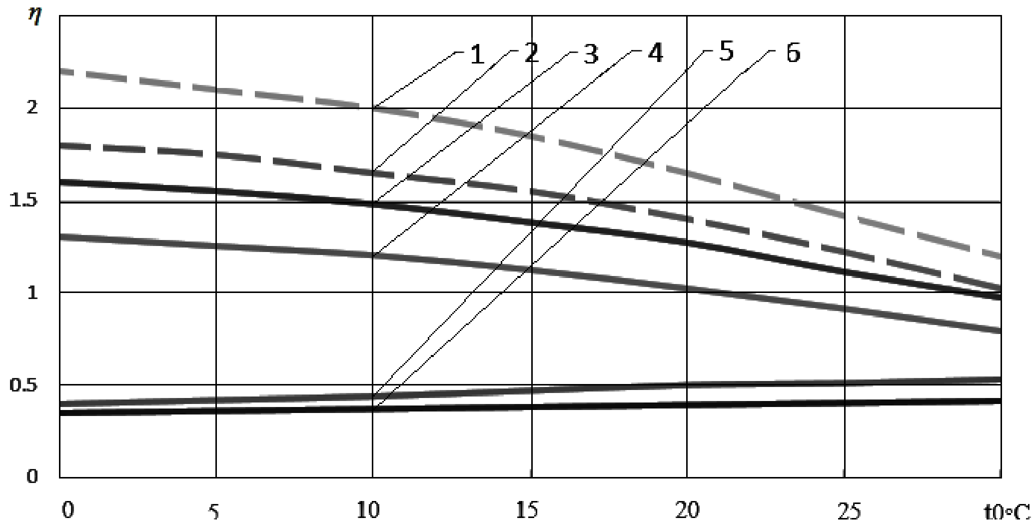

Fig. 2. *Comparison of energy efficiency of heat pump-adsorption and standard adsorption system for canning equipment depending on the temperature of fresh air: 1, 2 - for the theoretical system at a temperature of regeneration air of 60 ^{0}C and 65 ^{0}C; 3, 4 - for a real system at the optimum temperature of regenerative air and rotor thickness of 200 and 100 mm; 5, 6 - for a standard system without HP with a rotor thickness of 200 mm and 100 mm.*

Conclusions. From the above results of calculations, it is seen that the heat pump-adsorption scheme of conservation of equipment, even taking into account the real processes of dehumidification and humidification of the air in the adsorption rotor is more efficient than a standard system with an electric heater. The efficiency of the heat pump-adsorption system is 2-4 times greater than the efficiency of the standard adsorption system, which significantly reduces the cost of electricity for equipment maintenance. Also, the advantage of this system is that its efficiency increases with decreasing fresh air temperature, which is especially useful in temperate climates.

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GENERATION OF PRESSURE OSCILLATIONS IN AIR FLOWS IN VERTICAL CHANNELS WITH INTERNAL HEAT RELEASE

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Introduction. Periodic self-oscillatory processes accompanying the supply of heat into the flow of a compressible medium is an important factor that must be taken into account when designing heat and power equipment. Thermoacoustic self-oscillations can occur when convective heat is supplied to the flow from an external source or when fuel mixtures are burned in various energy devices. Self-oscillations of pressure can create additional mechanical loads on the structure of the combustion device, which can lead to its damage. This mode can also change the heat transfer conditions. Self-oscillations of pressure do not allow increasing the power and economic indicators of power equipment. The regularities of this phenomenon have not yet been sufficiently studied. Therefore, there are problems with the definition of ways to prevent its destructive action. Therefore, an important problem is to determine the dependence of the characteristics of self-oscillations on the level of thermal load on the system and on the geometric characteristics and operating parameters of the furnace devices.

Theoretical studies of thermoacoustic oscillations were carried out mainly at simplified mathematical formulations of problems of hydrodynamics and heat transfer, which make it possible to obtain their analytical solutions. In [1], the results of studies of pressure self-oscillations arising in combustion devices are presented. In this case, it is believed that self-oscillations arise due to the presence of a phenomenological delay in the combustion process. It is noted in [2] that pressure self-oscillations also arise as a result of “negative” viscous frictional resistance. An analysis of this effect on the occurrence of pressure self-oscillations is presented in [3]. A simplified technique for calculating the parameters of self-oscillations excited during unstable vibration combustion in vertical combustion chambers of blast furnace stoves was developed in [4].

The described results were obtained using simplified mathematical models that allow obtaining analytical solutions. Therefore, these results only approximately describe the characteristics of self-oscillations. To determine the quantitative characteristics of self-oscillations, it is necessary to apply more precise formulations of the problems of hydrodynamics and heat transfer and numerical methods for their solution.

Results. To determine the characteristics of self-oscillations of pressure in the flow, numerical modeling of the dynamics of the air flow and heat transfer in the vertical channel with a local internal supply of heat to the flow is performed. Air flow occurs due to natural convection. Heat is released in a certain volume of limited length due to the action of internal sources. This volume is located closer to the inlet section of the channel.

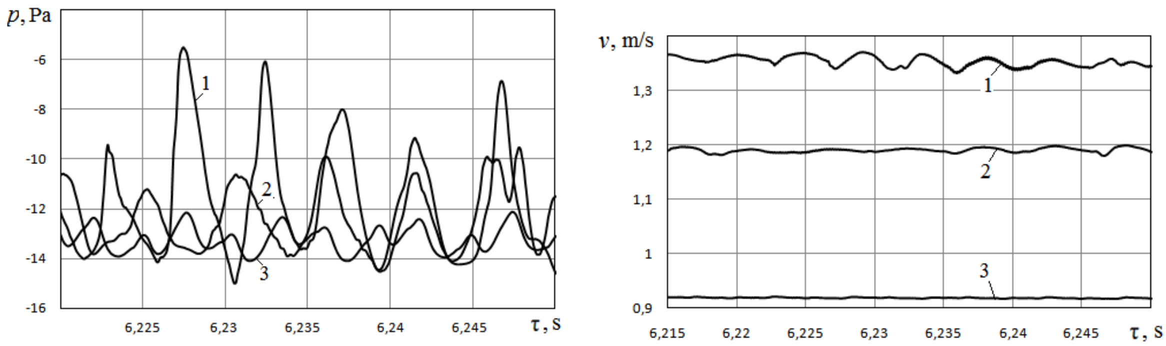
Numerical studies of hydrodynamics and heat transfer in a vertical channel, as well as an analysis of the possibility of occurrence of thermohydraulic instability in the flow, are carried out in the computational domain, which includes a cylindrical channel located in a cylindrical cavity with open lower and upper sections. Air flow is described by a system of Navier-Stokes and energy equations. As the boundary conditions for this system, the values of air pressure and temperatures at the inlet and outlet from the computational domain are set. This system is solved by the finite difference method. Based on the results of its solution, the time-varying fields of velocity, pressure and temperature are determined.

Analysis of the results of numerical studies showed that the time variation of the velocity in the channel has the character of oscillations with variable amplitude. For the case of a channel that has a height of 1.1 m and a diameter of 0.1 m, inside which heat is released with a power of 3 kW in a limited section, which begins at a distance of 0.16 m from the inlet section of the channel, the time average velocity in the inlet section is reaches a value of 1.06 m/s, and in the outlet section

- 2.07 m/s. That is, the average velocity at the exit from the channel is twice the flow rate at the entrance to the channel. The velocity oscillation frequency is ω_v~240 Hz. In this case, the oscillations in the velocity at the exit from the channel are in antiphase with the oscillations in the velocity at the entrance to the channel. That is, at the maximum velocity at the exit from the channel, the velocity at the entrance to the channel will be minimal. This indicates that the gas flow, moving upward under the action of the thermogravitational force, simultaneously carries out oscillatory movements, expanding and narrowing in the direction of the inlet and outlet sections of the channel. The time variation of pressure also has the character of oscillations that occur with variable amplitude from 1 Pa to 2 Pa and with a frequency of ω_p ~ 240 Hz.

With time, the amplitudes of the velocity and pressure oscillations in the channel decrease under the considered conditions. After a while, the oscillations practically damp out due to dissipative effects. The pressure and velocity oscillations have a somewhat different character in the case when there are elements near the outlet section of the channel that create additional local hydraulic resistance to the gas flow. Such elements can be, for example, flame stabilizers. Schematically, they can be represented as a system of concentric rings located in the channel and having a common axis of symmetry with the channel. The results of computational studies show that fluctuations in velocity and pressure in this case do not damp, but continue for a long time with a variable amplitude of ~ 1.0 Pa ... 2.5 Pa and with a frequency of ω_p ~ 240 ... 260 Hz.

These results refer to the case of the heat source power Q=3 kW. To determine the influence of the power of internal sources of heat release on the amplitude of pressure oscillations in a vertical channel, the problem considered above is also considered for Q = 2 kW and 1 kW. The time variation of the excess pressure over the time interval 6.215 s <τ<6.245 s in the cross-section that is located at a distance of 0.1 m from the outlet section of the channel at three values of the heat release source power are shown in Fig. 1a. It can be seen from the figure that with a decrease in the power of the heat source, the amplitude of pressure oscillations also decreases. With a decrease in the power of the sources of internal heat release, the average air velocity in the outlet section of the channel also decreases, as well as the amplitude of its oscillations (Fig. 1b).



a) b)

Fig. 1. Change in time of excess pressure (a) and velocity (b) at different power of sources of heat release: 1 - Q = 3000 W; 2 - 2000 W; 3 - 1000 W.

To determine the effect of the channel height on the characteristics of pressure self-oscillations in the air flow, the problem is also solved for the channel height H = 2.0 m at Q = 3000 W. Comparison of the nature of the change in time of excess pressure in the cross-sections of channels with a height of H = 1.1 m and H = 2.0 m, are shown in Fig. 2a. These cross-sections are located at a distance of 0.1 m from the outlet of the channels. As can be seen from this figure, at the same power of the sources of heat release, with increasing channel height H, the amplitude of pressure oscillations also increases. In this case, the frequency of these oscillations decreases with increasing of channel height. At H = 1.1 m, the frequency was ω_p ~ 240 Hz, and at H = 2.0 m, it was ω_p ~ 135 Hz.

The influence of the channel height on the characteristics of velocity oscillations is shown in Fig. 2b. It can be seen from this figure that with an increase in the channel height H, the average air velocity also increases. The amplitude of the velocity oscillations also increases, and the oscillation frequency decreases, as does the frequency of pressure oscillations.

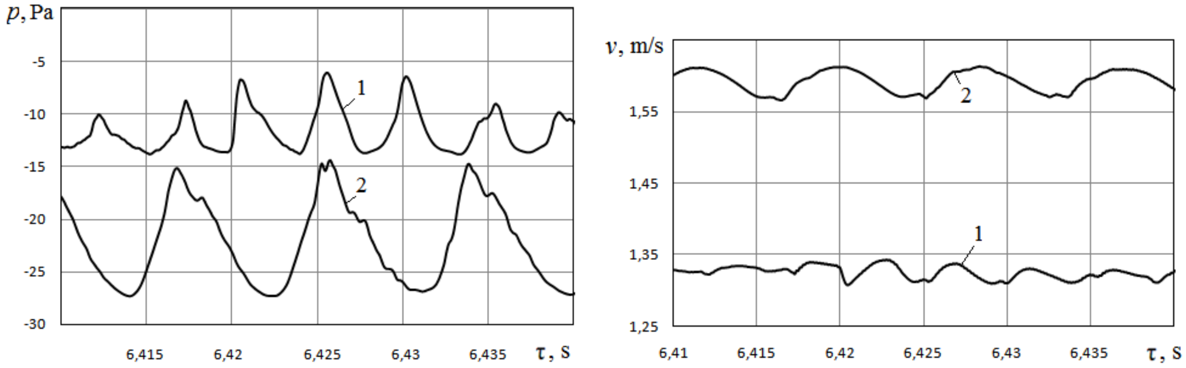


Fig. 2. Change in time of excess pressure (a) and velocity (b) in vertical channels of different heights at Q = 3000 W: 1- H = 1.1 m; 2 - H = 2.0 m

Conclusions. At a local internal release of heat in a compressible gaseous medium located in a vertical channel, a flow arises, which is accompanied by self-oscillations of pressure and velocity. The velocity in the lower section of the channel is always less than in the upper section. Velocity oscillations in the lower and upper sections of the channel occur in antiphase. Self-oscillations of pressure and velocity in a vertical channel with local heat release tend to decay with time. Self-oscillations do not damp and are maintained for a long time at a certain level in the presence of additional local hydraulic resistance in the channel. The nature of pressure self-oscillations is inharmonic with a non-constant amplitude.

The amplitude of pressure and velocity self-oscillations in a channel with local heat release increases with an increase in the power of the sources of internal heat release and with an increase in the height of the channel. The oscillation frequency decreases with increasing of channel height.

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INNOVATIVE AIR CONDITIONING SYSTEM WITH RATIONAL DISTRIBUTION OF THERMAL LOAD

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The performance efficiency of air conditioning (AC) systems depends on the heat efficiency of their air coolers. The intensity of heat transfer of refrigerant, evaporated inside air coils, drops at the final stage of evaporation, that is caused by drying out the inner wall surface while transition of refrigerant two-phase flow from annular to disperse (mist) flow. A sharp decrease in heat transfer coefficient to refrigerant at the final stage of its evaporation in compact air coolers results in lowering the overall heat transfer coefficient and reduction of air cooler efficiency.

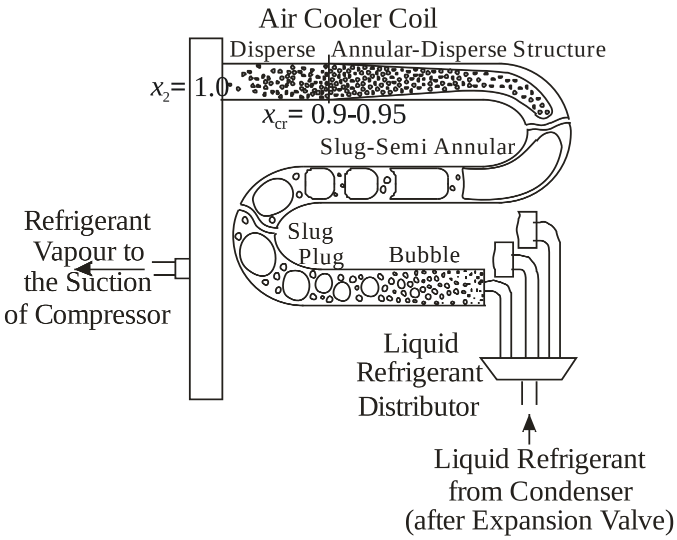
The concept of incomplete refrigerant evaporation with overfilling air coils that leads to excluding a dry-out of inner surface of air coils is developed.

Results of investigation

Typical structures of inside tube refrigerant evaporation and behaviour of refrigerant heat transfer coefficients α_a with the vapor mass fraction x are presented in Fig. 1.

The convective evaporation of refrigerant inside channels is characterized by sharp drop in intensity of heat transfer at the final stage of evaporation when so called burnout takes place. This occurs due to inner channel wall surface drying out with transition of refrigerant two-phase flow from annular-disperse flow to disperse (mist) flow (Fig. 1,a).

In compact air coolers with finned tubes the coefficient of heat transfer to refrigerant α_a at the final stage of its evaporation is much lower than α_{air} to air. This results in decrease in overall heat transfer coefficient k (Fig. 2,b).



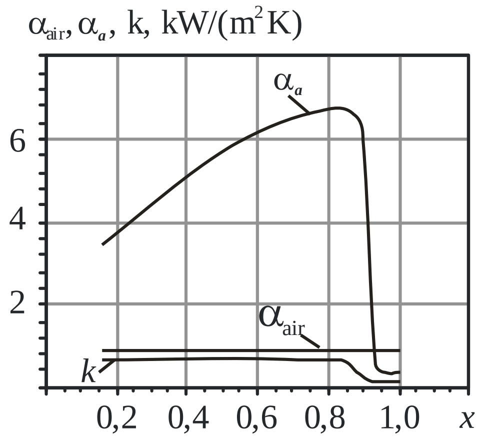


Figure 1. Typical structures of inside tube refrigerant boiling (a) and variation of heat transfer coefficients to boiling refrigerant α_a and air α_{air} and overall heat transfer coefficient k with the vapor mass fraction x (b)

Calculations are performed for the air cooler with plate finned tubes of 12 and 10 mm outside and inside diameters, air temperature at the inlet t_{air1} = 25 °C and outlet t_{air2} = 15 °C, refrigerant boiling temperature at the exit t_{02} = 0 °C, refrigerant R142b.

Considerable lowering the heat transfer coefficient to refrigerant α_a which becomes lower than the heat transfer coefficient to air α_{air} and causes a decrease in the overall heat transfer coefficient k at burnout vapor fraction $x_{cr} \approx 0{,}9$ corresponding to drying the channel wall surface with the transition from annular to disperse flow that leads to the sharp decrease in the heat flux q.

To provide intensive heat transfer on all the length of air cooler coils it is necessary to exclude their ending post dry out sections, i.e. make the air coolers operate with incomplete boiling. The unevaporated liquid should be separated from the vapour in the liquid separator and directed again at the entrance of air cooler.

The results of thermal efficiency comparison of conventional air cooler with complete evaporation and superheated vapor at the exit and of advanced air cooler with incomplete evaporation are shown in Fig. 2.

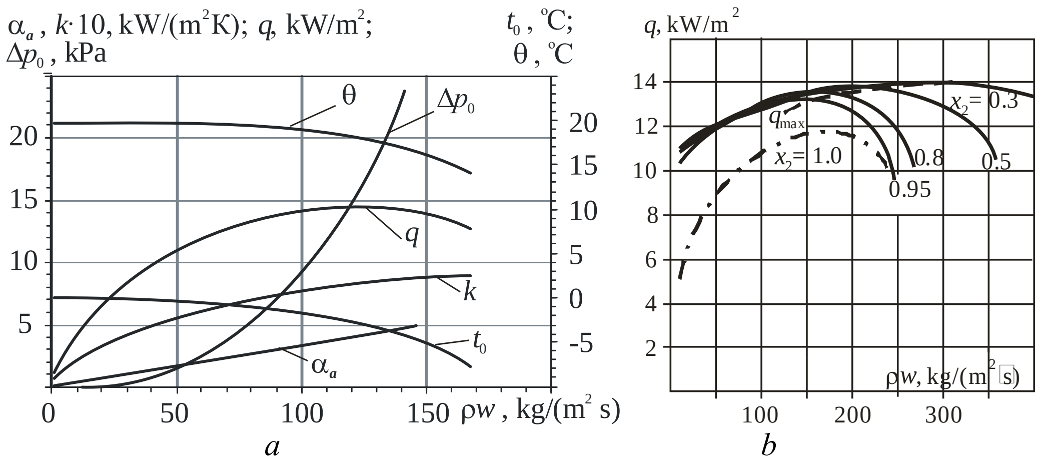


Figure 2. Mean values of heat fluxes q, heat transfer coefficients to refrigerant α_a and overall heat transfer coefficients k , logarithmic temperature difference θ, refrigerant boiling temperature t_0 and pressure drop ΔP against refrigerant mass velocities ρw for complete evaporation (a) and heat fluxes q at mass vapor fraction x_2 at the outlet of air coil for incomplete refrigerant evaporation (b): R142b, t_{02} = 0 °C; air velocity w = 6 m/s

Thus, overfilling the air coils of the air cooler by liquid refrigerant provides an increase in heat flux q by 25…40 % compared with conventional complete refrigerant evaporation and enables a larger deviation of refrigerant mass velocities ρw from their optimum value, providing maximum value of heat flux q, that means that a larger heat load changes are permitted, that gives good perspectives of its application in air conditioning systems.

Conclusions

A proposed novel concept of enhancing heat efficiency of heat exchanges with boiling refrigerants inside channels is intended to solve the problem of changeable actual heat loads on ambient air coolers by over filling air coils that provides excluding the final stage of refrigerant evaporation with low intensity of heat transfer.

Overfilling the air cooler by liquid refrigerant provides an increase in heat flux by 25…40 % compared with conventional complete refrigerant evaporation and enables a larger deviation of refrigerant flows from their optimum value, providing maximum value of heat flux. As a result, a larger heat load changes are permitted, that gives good perspectives of its application in air conditioning systems.

REDUCING THE HARMFUL EMISSIONS AND POROUS POLLUTIONS WHILE COMBUSTION OF WATER-FUEL EMULSIONS

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Abstract. *Based on the experimental and theoretical studies, a scheme of system for complex gas cleaning method of an internal combustion engine was developed. This system reduces the content of NOx in gases by 55%, SO_2 by 50%, and the content of solid particles by 3 times. The use of a complex system ensures that gases are purified from toxic ingredients and heat emissions to the level recommended by IMO.*
Key words: *water-fuel emulsion, internal combustion engine, harmful emissions.*

Introduction

Receiving additional energy due to deep utilization of heat losses of an internal combustion engine (ICE) saves fuel consumed for the operation of a ship's power plant. This, accordingly, leads to a decrease in emissions of harmful substances into the atmosphere, contributes to the satisfaction of the more stringent standards of the International Maritime Organization (IMO), which regulate the limits of these emissions.

According to the MAN specialists, the IMO requirements (III level from SO_2, NOx emissions) can be fulfilled using the following technologies: Water-fuel emulsion combustion (WFE) - WIF (Water in Fuel emulsion); Scavenge Air Monistening (SAM); Exhaust Gas Recirculation (EGR); Selective Catalytic Reduction (SCR).

The use of a combined SAM & WIF scheme to reduce NOx emissions is promising: water vapor in the combustion chamber increases the heat output and reduces the O_2 content. According to MAN, an increase in thermal power and a reduction of O_2 in the charge air provides a decrease of the combustion temperature, which leads to a decrease in NOx emissions. In addition, it should be noted that with a decrease of the combustion temperature, the soot concentration increases, as well as the amount of CO.

The use of WIF technology leads to an increase of fuel consumption up to 1.2% (if the condensing surfaces of isn't used when gases cooling below the dew point temperature of H_2SO_4 and H_2O vapors), SAM technology - 2.3%, EGR technology - 4.6%, SCR technology - 7.5-15% (taking into account the price of urea). But WIF technology provides a reduction of NOx to 30%, SAM technology - 45%, the existing EGR system - 70%; SCR technology - by 80% (at the required level of IMO requirements (III level) - 80% reduction of NOx). In addition, the SCR system must be accompanied by scrubber technology for the removal of SO_2.

The aim of the study is to develop a technology for the integrated purification of ICE exhaust gases.

Methodology

Possibility of solving complex problems in proposed technology is ensured by combustion of WFE with specifically recommended value of water content (W^r = 30%). Such WFE composition substantially affects not only running of thermal and physicochemical processes along the entire path of fuel combustion aggregates (starting from combustion zone and to cut of flue), but also directs them in the required direction. For performing tasks in technology of proposed method, providing solutions to problems of improving economic efficiency, improvement of environmental indicators and reliability, it is envisaged 4 stages of technological process:

1) WFE preparation with a water content of about 30%;

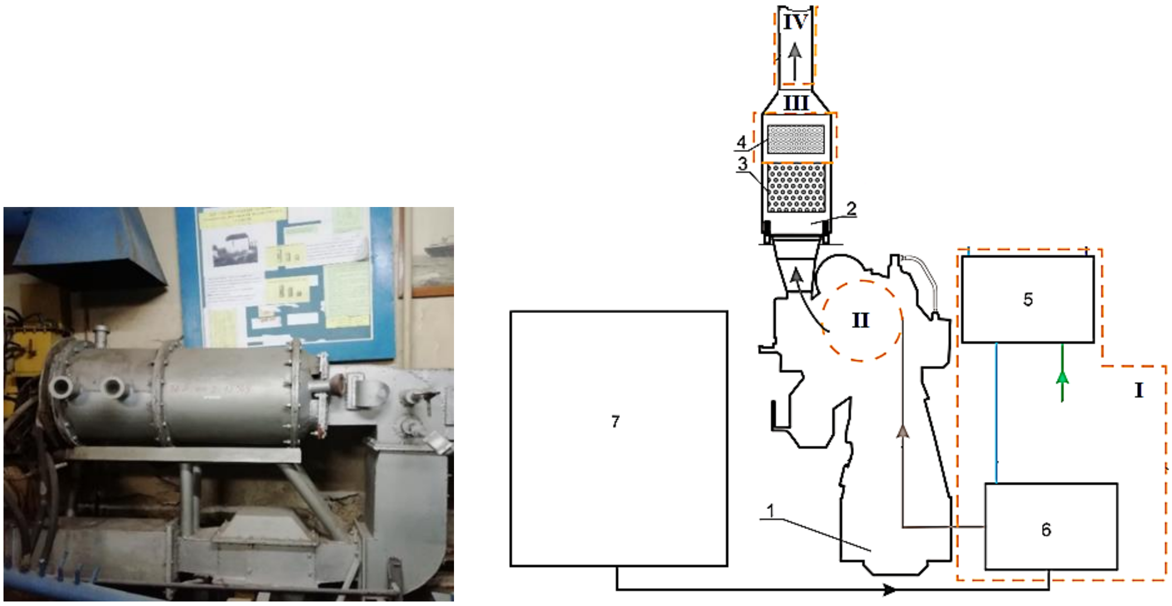
2) WFE combustion with a water content of about W^r = 30% leads to formation of equimolar ratio NO_2 : NO in exhaust gases at outlet of combustion zone (as confirmed by patent and providing low-temperature (LTC) reduction), as well as reducing NOx, SO_2, CO_2 emissions;

3) installation of condensing surfaces, on which conditions are created for passivation of metal and a sharp decrease LTC intensity, as well as conditions from side of gases and in condensate to intensify NOx, SO_2 absorption;

4) continuation of absorption intensification on condensing surfaces of gas flues (providing conditions for reliable operation of their metal) or maintaining temperature of metal of these gas flues above the dew point temperature of sulfuric acid vapour H_2SO_4 without NOx, SO_2 absorption, but ensuring reliability of work (at low LTC level).

Results

Based on the experimental and theoretical studies (Fig, 1, a), a scheme of system for complex gas cleaning method of an ICE was developed (Fig. 1, b).



a b

Figure 1. General view of experimental setup (a) A scheme of system for complex gas cleaning method and stages of cleaning (b)

I – WFE preparation with a water content of about 30%;; II – reducing concentration of toxic substances and solids in gases when WFE is burnt with water content 30 %; III – adsorption processes occurring on condensing surfaces of exhaust gas boilers; IV – processes occurring on condensing surfaces of gas flues;

1 – ICE: 2 – EGB; 3 – dry convective surface; 4 – condensing heating surface; 5 – water preparation unit; 6 – water-fuel emulsion preparation unit; 7 – fuel tank.

The main elements of the power plant, which provides for the combustion of specially prepared WFE with a water content of 30%, are the ICE and the exhaust gas boiler (EGB). A dry convective surface and a condensing surface must be installed In the EGB to perform tasks. It is also mandatory to install a water treatment unit and WFE (the first stage of gas purification). Specially prepared WFE is supplied to the ICE injectors.

As a result of the combustion of activated WFE at the engine outlet we obtain exhaust gases of the corresponding composition with a reduced amount of toxic ingredients up to 35% or more, and most importantly, the equimolar ratio of NO_2 : NO in NOx (which is confirmed by our experimental and literature data). This is the second stage of exhaust gas cleaning, which allows to reduce, for example, the concentration of NOx by 30-50%.

Further, the exhaust gases enter to the EGB, in which a dry convective surface is installed at the inlet (superheater, vapor-generating surface), and a condensing convective surface in the form of an economizer and (or) a hot water supply section with a metal temperature of 70-130 °C at the outlet, which leads to condensation of sulfuric acid vapors in the exhaust gases of ICE.

In the acid condensate under the indicated conditions, an average concentration of about 57% is established. The result is a sharp increase of SO_2 and NOx absorption. The presence in them of an equimolar ratio of NO_2 : NO provides passivation of the condensing surface made of carbon steel. This provides a sharp decrease of the LTC intensity, an increase of the operation reliability of these condensing surfaces and the possibility of a sharp increase of the depth of exhaust gases utilization to ~ 80-90 °C instead of 160 °C (when standard fuels combustion). Thus, the third stage of gas cleaning is carried out.

Further, the gases after the EGB enter to the gas duct. With such a low gas temperature and ensuring the temperature of the gas duct metal after the ICE at a level of 70- 80 °C, that is, in the presence of sulfuric acid condensate on the inner surface of the gas ducts, the process of absorption of toxic substances will continue with reliable operation of the gas duct metal, and the intensity of the mass flow will additionally decrease H_2SO_4 and LTC (this is the fourth stage of gas purification).

This is due to the fact that in gases there is an equimolar ratio of NO_2 : NO, which means that the passivation of the metal surface and a decrease of the LTC will be ensured with a minimum temperature difference between the gases and the metal of the gas ducts.

Thus, the implementation of these stages of gas purification provides a decrease of the content of toxic ingredients in gases by almost 50% (compared to existing technologies, which provide a decrease of the exhaust gas temperature of the EGB to 160 °C) and partial removal of solid particles, contained in gases (when WFE combustion, solid and soot particles are 80% less than standard fuels combustion).

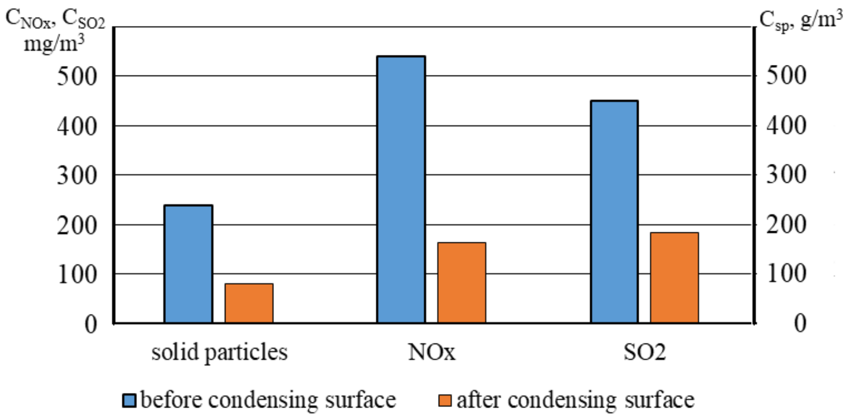


Figure 2. Emission rates of toxic ingredients.

Conclusions

When WFE combustion with a water content of 30%, the LTC intensity decreases, which makes it possible to install condensing heating surfaces in the EGB. The installation of a condensing heating surface in the EGB reduces the content of NOx in gases by 55%, SO_2 by 50%, and the content of solid particles by 3 times. The use of a complex system ensures that gases are purified from toxic ingredients and heat emissions to the level recommended by IMO.

MAIN ENGINE OF TRANSPORT SHIP INTLET AIR COOLING BY EJECTOR CHILLER

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Abstract. *The efficiency of cooling the air at the inlet of marine slow speed diesel engine turbocharger by ejector chiller utilizing the heat of exhaust gases and scavenge air were analyzed. The values of air temperature drop at the inlet of engine turbocharger and corresponding decrease in fuel consumption of the engine at varying climatic conditions on the route line Odesa-Yokogama- Odesa were evaluated.*
Key words: *internal combustion engine, ejector chiller.*

Introduction

Slow speed diesel engines are the most widespread as the main engines of the ships. of sailing change The fuel efficiency of diesel engines are considerable effected by the variations in ambient air temperatures along the route lines. The ejector chiller can be applied as the most simple in design for cooling marine diesel engines intake air.

The purpose of the work is to estimate the efficiency of cooling the intake air of marine slow speed diesel engine by ejector chiller taking into account the variable climatic conditions along the route line.

Methodology

A slow speed diesel engine 6S60MC6.1-TI [2] is considered as an example of the main engine of transport vessel: nominal power N_n = 12.24 MW and continuous service power N_s = 10 MW. For the 6S60MC6.1-TI engine, according to the data of the MAN company (according to the calculations by using "mandieselturbo" software package), cooling inlet air for every 1 °C results in reduction of specific fuel consumption within 0.11 to 0.12 g/(kWh) [1-3].

The efficiency of engine intake air cooling is estimated by decrease in specific fuel consumption Δb_e due to reduce of intake air temperature Δt_a, that depends on the heat Q_h extracted from the exhaust gas and scavenge air and the efficiency of its conversion in refrigeration capacity Q_0 of the chiller, i.e. coefficient of performance COP.

The efficiency of conversion of waste heat into refrigeration capacity is characterized by coefficient of performance $\zeta = Q_0/Q_h$ as the ratio of the chiller refrigeration capacity Q_0 to the consumed heat Q_h , extracted from the engine exhaust gases, scavenge air and others.

The available refrigeration capacity Q_0 of ECh is calculated as $Q_{0.ECh} = Q_h\,\zeta_{ECh}$, where Q_h – the heat, extracted from the engine exhaust gases and scavenge.

The values of the available air temperature drop in the ECh air cooler $\Delta t_{a.ECh}$ due to using ECh available refrigeration capacities $Q_{0.ECh} = Q_h\,\zeta_{ECh}$ is calculated proceeding from the heat balance as $Q_{0.ECh} = G_a\,\xi_a\,c_a\,\Delta t_{a.ECh}$ as $\Delta t_a = Q_{0.ECh}/G_a\,\xi_a\,c_a$, where G_a – air mass flow rate, kg/s; c_a – specific heat capacity of wet air, kW/(kg·K); ξ_a – specific heat ratio of cooling air process in air cooler.

The available temperatures of cooled air at the outlet of the air cooler $t_{a2} = t_{a1} - \Delta t_a$.

The current values of reduction in specific fuel consumption per 1 hour: $\Delta b_e = \Delta t_a \cdot \Delta b_{e1°C}$, g/kWh, and the total fuel reduction per 1 hour: $\Delta B_e = N_s\,\Delta b_e$ or $\Delta B_e = N_s\,\Delta t_a\,\Delta b_{e1°C}$, g/h, where $\Delta b_{e1°C}$ – reduction in specific fuel consumption referred to engine intake air temperature drop in 1 ºC or 1K, $\Delta b_{e1°C} = \Delta b_e/\Delta t_a$ = 0.12 g/(kWh·K); N_s = 10000 kW– diesel engine power output

A refrigeration capacity of ejector chiller Q_0 is defined from available exhaust gas heat Q_G as $Q_0 = \zeta\,Q_G$, where ζ – coefficient of performance of ejector chiller, ζ = 0.35.

A schema of developed engine intake air cooling system with ejector chiller utilizing the heat of exhaust gas in evaporative section of ECh generator and scavenge air in economizer section is shown in Fig. 1.

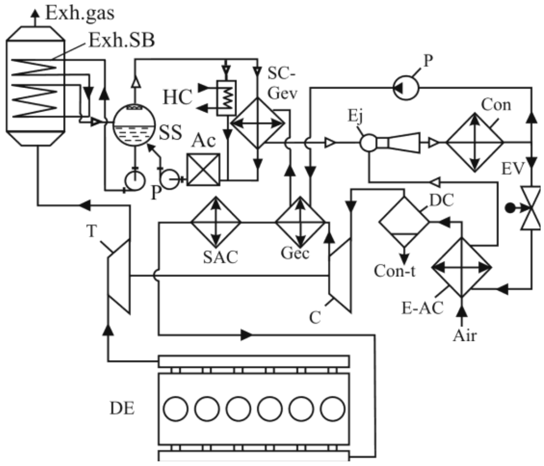


Figure 1. A schema of the engine intake air cooling system with ejector chiller utilizing the heat of exhaust gas and scavenge air: DE – diesel engine; T – turbine, C – compressor of turbocharger; SAC – scavenge air cooler; Exh.SB – exhaust gas steam boiler; SC-Gev – steam condenser-evaporative section of ECh generator; Gec – economizer section of ECh generator; E-AC – evaporator-air cooler; Ej – ejector; Con – condenser; EV – expansion valve; P – pump; Con-t – condensate; DC – droplet catcher; Ac – accumulator of feed water; SS – steam separator; HC – heat consumer.

The ejector chiller consists of power and refrigeration contours. A generator of power contour uses a heat of exhaust gas to produce a high pressure refrigerant vapour as a motive fluid which energy is used in ejector to compress the low pressure refrigerant vapour, sucked from evaporator-intake air cooler of refrigeration contour, up to the pressure in the condenser.

Results

A route line Odessa-Yokogama-Odessa (June-July) 2019 is considered (Fig. 2).

For each time interval (3 hours) along the route line Odessa-Yokogama-Odessa the values of ambient air temperature t_{amb} and relative humidity φ_{amb} were fixed by applying the well known program "mundomanz.com" to calculate the processes of cooling intake air in the air cooler at the inlet of turbocharger of diesel engine and define the required temperature drops Δt_a and refrigeration capacities $Q_{0.ECh}$, as well as the available air temperature drop in the ECh air cooler Δt_a due to using the available refrigeration capacities $Q_{0.ECh}$ of ECh utilizing the heat of engine exhaust gas and scavenge air.

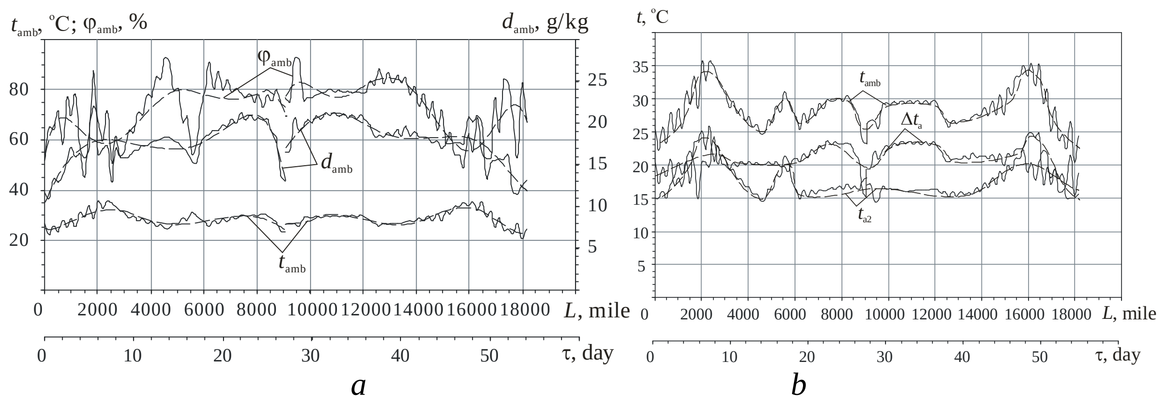
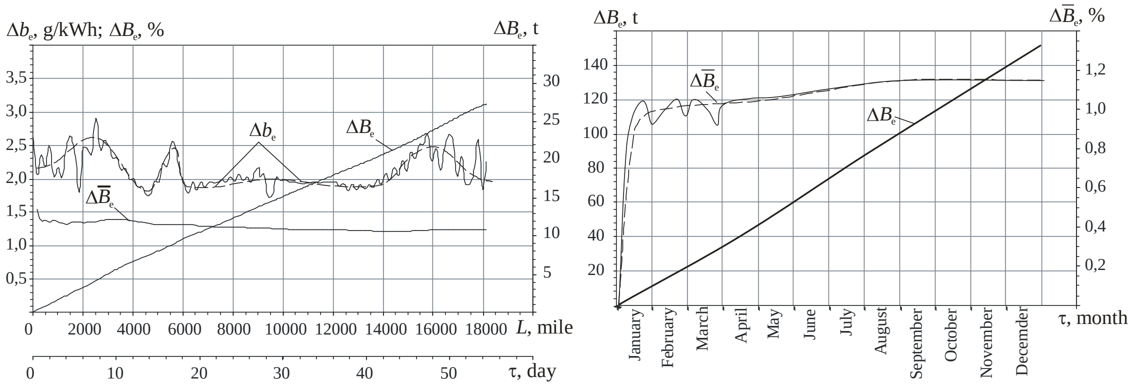


Figure 2. Variation of ambient air temperature t_{amb} , relative humidity φ_{amb} and absolute humidity d_{amb} on the route line Odesa-Yokogama-Odesa

Decrease in specific fuel consumption Δb_e , g/(kW·h), of diesel engine, fuel saving in absolute B_e , t, and relative B_e' , %, values on the route line Odessa-Yokogama-Odessa and their summarized annual absolute ΔB_e and relative $\Delta B_e'$ values due to cooling intake air by ejector chiller, using exhaust gas and scavenge air heat are resulted in Fig. 3.



a *b*

Figure 3. Decrease of specific fuel consumption Δb_e, total fuel consumption for engine power N_s = 10000 kW in absolute ΔB_e, t, and relative $\Delta B_e'$, %, values referred to the engine total fuel consumption on the route line Odesa-Yokogama-Odesa, June-July 2019 (a) and their summarized annual absolute ΔB_e and relative $\Delta B_e'$ values (b)

As Fig.3 shows, a decrease of specific fuel consumption due to intake air cooling by ejector chiller Δb_e = 2.0…2.5 g/(kW·h), absolute fuel saving during the routes Odesa-Yokogama-Odesa, June-July 2019, is ΔB_e = 26 t (Fig.3,a) and annual fuel saving ΔB_e = 159 t (Fig.3,b) and the relative fuel saving $\Delta B_e'$ is about 1.3 % for diesel engine 6S60MC6.1-TI (continuous service power N_s = 10 MW). In order to provide a deeper engine intake air cooling to the temperature t_{a2} of about 10 ºC and lower it is necessary to apply two-stage cooling air in hybrid water-refrigerant air cooler by combined absorption-ejector chillers with higher COP.

Conclusions

The efficiency of application of waste heat recovery ejector chiller system for cooling the intake air of marine diesel engine has been analyzed for real changeable climatic conditions on the routes Odesa-Yokogama-Odesa. The application of ejector chiller provides reducing the engine intake air temperature by 20…23 ºC with corresponding decrease of specific fuel consumption by 2.0…2.5 g/(kWh). In order to provide a deeper engine intake air cooling to the temperature t_{a2} of about 10 ºC and lower it is necessary to apply two-stage cooling air in a hybrid water-refrigerant air cooler by combined absorption-ejector chillers with a higher COP.

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EFFICIENCY ANALYSIS OF THE AEROTHERMOPRESSOR APPLICATION FOR INTERCOOLING BETWEEN COMPRESSOR STAGES BY USING CFD MODEL

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Abstract: A study of the aerothermopressor operation for air intercooling between the stages of a multistage compressor as part of a modern gas turbine (LMS100 brand from General Electric) was carried out in the article. A calculation method has been developed using numerical modeling for the evaporation of fine water droplets in the air flow. The main characteristics of the two-phase flow at the aerothermopressor outlet have been determined. It has been found that jet apparatus provides efficient atomization of the liquid, and hence, more efficient isothermal compression process in a high-pressure compressor. The aerothermopressor applying allowed to reduce the temperature of the compressed air between the compressor stages to 50–70 °C. Such a decrease in temperature under the thermo-gas-dynamic compression conditions allowed to increase the pressure at the aerothermopressor outlet up to 12–28 kPa (4-9%).

Keywords: Pressure Increase, Two-Phase Flow, Gas Turbine.

General Electric has brought into commercial operation the first modern LMS100 gas turbine with a nominal capacity of almost 100 MW using air intercooling technology in 2005. This gas turbine provides the highest efficiency in an open circuit to date. A special feature of the LMS100 is the use of intercooling within the air compression section of the compressor. Today it is the only mass-produced unit of this type in the world. The use of intercooling in the LMS100 made it possible to increase the air pressure degree up to $\pi_c = 40$, while the efficiency was $\eta_e = 45.5\%$ [1].

An alternative way to inject water into the air flow between the compressors is to use an aerothermopressor (Fig. 1). The apparatus belongs to the type of jet devices and consists of the following main elements:

1) a confuser (designed to accelerate the air flow to a speed close to the sound speed);

2) a nozzle (designed to inject water into the flow);

3) an evaporation chamber (the process of thermo-gas-dynamic compression is taken place);

4) a diffuser (designed to equalize the flow, reduce the velocity and increase the pressure of the flow).

The advantages of using the aerothermopressor include the following:

- increasing the pressure and cooling of the working fluid will reduce the compression work in the compressor;
- ensuring effective atomization and humidification of liquid (water) between compressor stages;
- reduction of the additional work of the compressor during the evaporation of water droplets in the flow path during compression;
- an increase in the amount of the working fluid in the cycle [2].

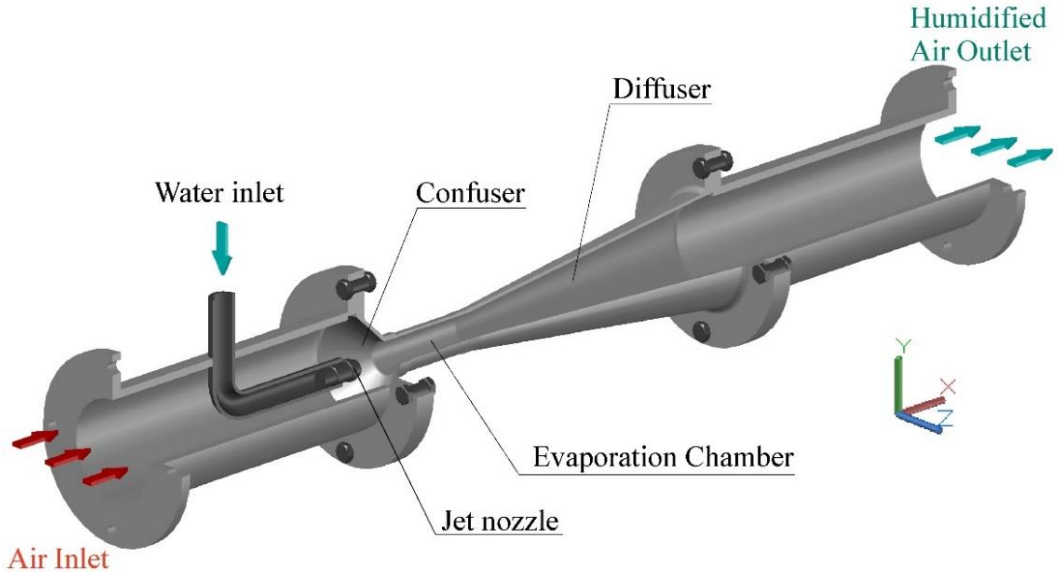

Fig. 1. 3D model of the aerothermopressor

To carry out numerical modeling, the finite volume method was applied, which is implemented in the ANSYS Fluent software package. The Eulerian-Lagrangian approach was used to simulate the interaction of injected water droplets and air flow. A two-parameter k-ε Realizable turbulence model from the RANS group of models was used to investigate the behavior of the air flow [2, 3]. Discrete Phase Model was used to simulate the movement of water droplets [4].

To analyze the gas turbine cycle, the well-known calculation methods were used [5]. The calculation of the gas turbine cycles was carried out for the degrees of pressure increase π_c = 12–40, while in the circuits, instead of an air cooler (surface or contact with nozzle injection), it was proposed to install the aerothermopressor.

As this chart illustrates the use of the aerothermopressor made it possible to reduce the air temperature between the compressor stages by t_{2atp} = 50–70 °C (Fig. 2), that is, up to 50–110 °C. Such a decrease in temperature under thermo-gas-dynamic compression conditions made it possible to increase the pressure by ΔP_{atp} = 12–28 kPa, that is, up to 4–9% (Fig. 2). Contact air cooling by using the aerothermopressor allowed to reduce the compressor compression work by 2.5-3.0%.

The amount of water injected into the aerothermopressor can exceed the value required for evaporation in the evaporation chamber (up to 10% relative to the amount of air). This solution will allow to obtain evaporation during compression in a high-pressure compressor and, as a consequence, bring the compression process closer to isothermal with the lowest value of the work in compression. Thus, the use of the aerothermopressor can be an alternative to the traditional contact cooling of compressed air when injected by nozzles.

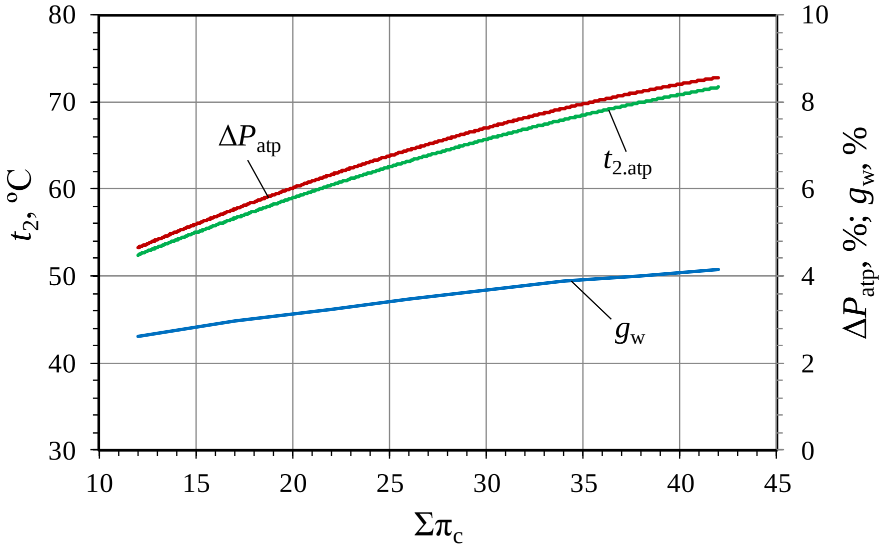

Fig. 2. Dependences of the outlet air temperature ($t_{2.atp}$), the relative water flow rate (g_w) and the relative pressure increase at the aerothermopressor outlet (ΔP_{atp}) on the total compressor pressure increase $\Sigma\pi_c$

Conclusion: The paper analyzes the efficiency of using an aerothermopressor for contact cooling of compressed air in the LMS100 gas turbine circuits. Aerothermopressor provides effective fine atomization of water, and hence, a more efficient compression process in the high-pressure compressor.

It has been determined that the aerothermopressor allows to increase the air pressure between the compressor stages by 4-9%, as a result of which the compression work in the compressor stages decreases; increase the amount of the working fluid in the cycle by g_w = 2–4%, and, as a consequence, increase the specific power of the gas turbine by 3–10%.

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INTEGRATION TECHNOLOGIES OF HEAT STORAGE INTO DISTRICT HEATING SYSTEMS

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Introduction

Improving the efficiency of heat supply contributes to the overall energy savings. District heating (DH) and cooling networks are the communal the pipeline of transport infrastructure used to transfer thermal energy from the central boiler house to numerous users. The main consumers are the residential sector and industrial facilities. It should be noted that the housing sector uses about 26.1% of final energy consumption in the European Union [1]. In Ukraine, 1.6 times more primary energy is used for heat production than for electricity production. Electricity generation accounts for 39%, while high-potential heat production outlay for 21% and low-potential heat (below 100 ° C) intended for 40%, respectively. Thus, increasing the efficiency of the DH system can potentially effectively contribute to overall energy savings.

The advantage of the DH system is that it is possible to provide heat to huge areas at the same time, by using a variety of cheap fuels and provides centralized control the temperature of the coolant. In addition, the system is controlled in distance by specialized organizations, which provides high reliability, environmental friendliness and ease of use.

Despite many positive aspects, this system has a number of disadvantages. The main disadvantage is the large losses in the transportation and distribution of heat, the lack of quantitative regulation and violation of the temperature regime at peak load. The present-day state of heating networks does not meet the technical requirements. In the EU, about 11% of heating systems are inefficient [2]. In Ukraine, this figure reaches 17%.

Thermal energy accumulation and storage (TES) systems can solve the problems of unstable operation of the DH system during the peak period of heat consumption, ensure stable operation of boiler equipment with the highest possible efficiency, reduce the consumption of electricity and organic fuels. In addition, its allow to attract to multi-fuels balance systems of renewable energy and secondary energy resources, as well as significantly reduce harmful emissions into the environment.

These works are especially relevant in today's socio-economic environment to ensure energy security and flexibility of DH systems, decarbonization of generation, reducing the cost of transportation and distribution of thermal energy and the prospects for the transition to new advanced technologies of the fourth generation. Fourth generation DH systems are a modern trend in heat supply, which is being actively implemented in the EU, USA and China. In particular, their implementation can solve environmental problems, as the European Union has announced a "green agreement" aimed at reducing greenhouse gas emissions by 50% by 2030 compared to 1990 [3].

Classification technologies of heat saving

TES technologies make it possible to security heat or cold and use excess heat energy during several hours, days or even months. Global trends in efficiency, energy savings and security of energy systems are based on the principle of energy storage. Depending on the form of energy storage, general heat storage technologies are divided into the following categories:

1. Technologies for the transformation of mechanical energy into heat are widespread, but are characterized by low efficiency approximately 40%.

2. TES, which are based on single-phase capacitive accumulation for seasonal or short-term accumulation of thermal energy, which allow the use of heat or cold from natural sources and secondary energy resources. Thermal accumulation is carried out by solid or liquid substances due to the heat capacity of the material. The capacity of heat saving in such accumulators is from one hundred kilowatts to hundreds of megawatts. Payback period from 4 to 6 years.

3. The accumulation of thermal energy in the so-called latent TES systems is based on the use of materials with a phase transition of organic (high molecular weight paraffin's, paraffin's and glycols) or inorganic origin (crystalline hydrates, salt hydrates and eutectic water - salt solutions), which have a high latent heat capacity. The limited use of phase exchange heat accumulators can be explained by the low coefficient of thermal conductivity and excessive corrosive activity of inorganic materials, as well as a change in volume during melting of materials of organic origin. The payback period is approximately 3 years.

4. General interest is providing technologies for the use of own chemical energy based on adsorption processes. As a working solid are used the zeolites and silica gels, as a heat carrier is used the air. Thermal energy storage can be daily, weekly, monthly or even seasonal, depending on the volume of the working solid. Payback period till 7 years.

5. Heat accumulators based on photochemical and thermochemical reactions, thermoelectric and triboelectric principles of operation have not found wide application and are currently at the stage of research tests.

Application of heating storage technologies in District Heating Systems

District heating and hot water supply in Ukraine has great potential for the use of main pipelines as heat accumulators, which is allowed by current regulations documents [4]. If DH occupies in Western Europe about 10% of the total heat supply market, DH reaches in Ukraine 70% [5].

Our research shows that the use of heat supply networks as accumulators of heat energy and the optimization of the DH system can reduce the total energy costs by approximately 30%.

In addition to promising heat storage technologies in central heating systems include heat pumps - a saving-effective technology that uses electricity to generate heat from the ground or air for heating systems of industrial, commercial and residential buildings. However, investment in electric heat pumps is not attractive due to the lack of subsidies at the legislative level. High-power heat pumps of DH are considered a saving-effective and energy-efficient solution [6].

Widespread stationary heat accumulators have been used in DH since the first generations of heating systems. Stationary TES are used both in homes and in combination with thermal power plants, boiler rooms and heat pumps in DH. Moreover, TES gives possibility optimal boiler load management, reduction of consumption fuel and energy resource and harmful emissions into the environment.

Mobile heat accumulators (M-TES) can be successfully used to combine several sources of thermal energy combined into a single system. It has especially important including increasing the use of renewable and secondary energy sources [7].

In the laboratory "Processes and technologies of heat supply" ITTF NAS of Ukraine in 2020 created and tested a mobile heat accumulator with a thermal capacity of 0.8 MW, heat productivity of 1200 kWh. Charging time is 4 - 6 hours, discharge time 10 - 12 hours, heat storage capacity 200 kWh, average load power 120 kW, average discharge power 90 kW. The mobile container type M-TES is equipped with a block heat point and tanks-accumulators of heat. M-TES ITTF NAS of Ukraine has a number of advantages over analogues, in particular:

- reduction of heat losses in the modes of transportation and waiting, thanks to perfect insulation;

- work without staff maintenance and compactness of heat storage systems;

- high intensity of coolant supply, especially during the first hour;

- the ability to transport excess heat from industrial sources for a distance of more than 50 km, which in our estimation, when implemented in Ukraine, provides an opportunity to cover up to 5.0% of total demand for heat supply.

Conclusions

Global trends in efficiency, energy storage and security of energy systems actualize the task of developing energy stockpiling technologies.

Thermal energy storage can be used to control the load of heating systems, i.e. the equation of generation energy sources, to provide the peak demand for heating and electricity with a high utilization of equipment. The work has considered current technologies of saving and storage of thermal energy that can be used in central heating systems and was done conclusions about the feasibility of their practice.

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THERMOPHYSICAL BASES OF INCREASE OF EFFICIENCY OF HEAT CONSUMPTION OF THE BUILDING AT THE USE OF INDIVIDUAL HEAT POINT

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Introduction. It is known that heat losses in the housing stock are up to 30-45%. Therefore, measures to save heat consumption by the final consumer, in the building, are the most effective. Today, in buildings, especially old buildings, unregulated heat points are used, in which the equipment is already physically and morally obsolete, which leads to significant heat consumption. Therefore, the transition from central heat points to individual heat points (IHP), located directly in the building, with the installation of appropriate automation and monitoring of energy use, as well as information technology is an urgent task. It should be noted that the implementation of IHP should be carried out only after the insulation of the building, because if you do the opposite, the calculated parameters of the equipment that is part of it will no longer meet the requirements. There is also another important problem in Ukraine – a significant uneven daily schedule of electricity consumption in the United Energy System (UES) of Ukraine. One of the technical solutions to this problem is the introduction of automated IHP with electric boilers, which allows not only to effectively regulate heat consumption of the building, but also independently of heat distribution networks to generate heat, which in turn allows the use night electricity at reduced rates. This is extremely important for the sustainable, high-quality and safe operation of the UES of Ukraine.

Purpose of the work. Improving the efficiency of end use of thermal energy by the consumer (building) by effectively controlling its heat consumption based on thermophysical modeling of the process of heat consumption management and conducting experimental research using an individual heat point.

For numerical study of the air-temperature regime of the room with intermittent heating, a model with concentrated parameters is used [1]. The model assumes the presence of a certain number of characteristic calculation elements in the room, the temperature values of which determine the temperature of the room as a whole. For a room with one window and one heating device, the rational number of such elements is 12. For each of them, the corresponding equations of heat balance are compiled. Compiling the equation of heat balance for the inner walls of the room, as well as the ceiling and floor, we assume that the surfaces of these elements receive radiant heat flux from the radiator. From the surfaces of the mentioned elements the radiation flow enters the part of the inner surface of the outer wall and the inner surface of the window. In addition, heat is removed from these surfaces by convection to the indoor air. A system of twelve nonstationary heat balance equations is solved numerically. The results of solving this system are time-dependent temperature values of each of the twelve characteristic elements of the room. Calculations of the thermal state of the room with intermittent heating were performed for a model room 6 m long, 3 m wide and 3 m high. Such premises are the most typical for the administrative buildings of the Institute of Engineering Thermophysics of NAS of Ukraine. A two-panel convector 0.5 m high and 1 m wide, installed under a window at a distance of 0.05 m from the outer wall, is used as a heating device. The window, which has an area of 3.72 m^2, has its own thermal resistance R_{window} = 0.16 m^2·K/W. The thermal resistance of the outer wall is R_{wall} = 2.32 m^2·K/W. The heat in the room comes from a two-panel convector, as well as from office equipment and people who work 8 hours a day (working hours) and have a total power Qv = 200 watts. According to the results of the calculation, the intermittent heating mode of the working space, which involves reducing the temperature of the heater after working hours and increasing it before working hours to ensure the normative temperature conditions during working hours, reduces daily heat consumption.

Effective control of heat consumption of the building is possible with the help of automation as part of the IHP. Experimental studies were conducted on the example of administrative buildings №1 and №2 of the Institute of Engineering Thermophysics of NAS of Ukraine, located in Kyiv, 2, Bulakhovskoho Str. These are the same type of three-storey buildings with a total area of 3240 m^2, built in 1973. Unregulated heat point is located in each administrative building of the Institute of Engineering Thermophysics of NAS of Ukraine, which leads to inefficient and significant consumption of thermal energy, as there are no weather-dependent control devices in it. Therefore, an experimental IHP with a hydraulic arrow according to the dependent hydraulic scheme of connection to the heating network (building №1), as well as an IHP with electric boilers according to the independent connection scheme (building №2) were developed and implemented. They provide automated regulation of heat consumption of the building depending on the ambient temperature. Experimental studies were conducted at different modes of IHP operation and at different control algorithms: by outside air temperature, by air temperature in the control room, by daily-weekly schedule of regulation, in which in periods of human absence the controller set ambient temperature from 0 ºC to +9 ºC to reduce heat consumption. With the help of the measuring complex, the obtained experimental data were archived: temperature and coolant flow rates in the supply pipeline and return pipeline, outdoor and room temperature. As a result of the analysis of the constructed graphs of the basic parameters features of heat consumption of buildings depending on the set algorithm of regulation are established.

To compare the results of the calculation on the basis of the thermophysical model, experimental data from experimental studies of heat consumption of the building №1 using IHP with a hydraulic arrow were selected. To solve the problem of thermal regime of the whole building, a system of 12 equations is used for each room in the building, as well as for corridors and stairways. If the room has 2 or more windows, as well as 2 or more heating devices, the number of calculation elements for such a room increases accordingly.

Comparison of the calculated results with experimental data showed that the discrepancy in the maximum deviation is 6… 7% and is within acceptable limits.

Conclusions. On the basis of the thermophysical model the methods of effective control of heat consumption of the building were studied, experimental researches of features of heat consumption of the building at use of IHP were conducted. Also comparison of results of calculation on the basis of thermophysical model with experimental data was conducted.

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ENERGY EFFICIENCY OF HEAT PUMP HEAT SUPPLY SYSTEM WITH HEAT UTILIZATION OF TECHNOGENIC AIR EMISSIONS

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Introduction. In modern conditions energy saving along with energy efficiency are the common priority direction of each country's energy policy. In order to bring Ukraine out of the crisis situation in the field of heat supply, it is necessary to use alternative sources of energy from the environment in all their diversity, and in particular secondary technogenic waste heat resources. One of the most effective types of modern non-traditional energy technology, which allows the use of renewable and alternative energy, are heat pumps (HP) [1].

Industrial units and systems, where the high-temperature heat-technological process is realized, create the technical base of main productions of the national economy, namely ferrous and non-ferrous metallurgy, production of building materials, food industry and mechanical engineering. They consume almost 80 % of high-potential thermal energy, but the use of thermal energy in these economy productions is characterized by a relatively low fuel heat utilization rate – in the range of 15-35 %. As a result of technological processes at industrial enterprises there is a large amount of low-temperature thermal energy which is not used in the technological cycle. Depending on specific conditions, the spent thermal energy can be used in heat pumps as a lower heat source for heat supply of workshops, production areas, warehouses of industrial enterprises [1-3].

Thus, through the introduction of heat pumps in enterprises with high-temperature processes and units, it is possible to create a combined energy technology system that naturally connects energy and heat technology systems to ensure the highest economic efficiency of energy production and technological products realization.

Almost all scientific or research developments related to the introduction of heat pump technology for technogenic air heat sources recovery are at the stage of individual design solutions and applications. In the available literature there are only isolated studies without comprehension and full analysis of results and attempts to apply them to other systems. Thus, the analysis of research in the field of HP applications in heat supply systems showed that this issue is open and necessary to address.

Based on the analysis of works [1-2, 4] it is established that the use of air-to-water heat pump heating systems has significant advantages, as atmospheric air is a completely free, unlimited and comprehensive source. Such systems require low initial investments compared to other energy sources and can be placed on any object. However, a significant disadvantage of such systems is the loss of power and efficiency with decreasing air temperature. This problem can be solved by using technogenic air heat sources in a heat pump, because they have a temperature that varies in the range of 10-50 ºC and have positive values in winter.

In the article [3] the thermodynamic analysis of simple heat pump heating systems is carried out and it is established that when using low-temperature heat sources (atmospheric and ventilation air) there is an optimum degree of lower heat source cooling in the HP evaporator, which corresponds to the minimal total amount of energy spent in in heat pump systems (HPS).

When using technogenic air emissions in heat pump systems (HPS) for heat supply there is a similar problem – the demand to determine the depth of their cooling, in other words choosing the optimal degree of air emissions cooling which corresponds to the maximum useful effect obtained from utilization of their thermal energy, considering operating loads of the HP compressor drive.

Figure 1 shows a HPS heat supply system that works by utilization of technogenic air heat sources. The operation principle is a follows: low-temperature heat source, i.e. exhausted air with temperature t_1 and mass flow G is fed into the HP evaporator. Here the coolant is cooled and its outlet temperature is t_{ev}. The heat supply of the room is carried out due to the heat flow extraction from the HP condenser Q_{heat} with the temperature of the heating coolant $t_{C_{HP}}$ at the entrance to the heating system.

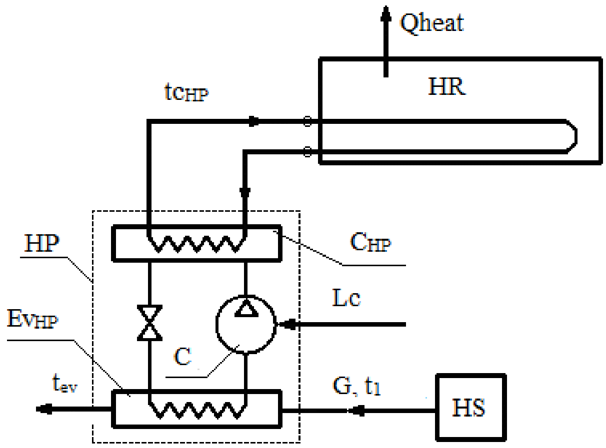

Figure 1. HPS heat supply system that works by utilization of technogenic air heat sources: HR – heated room, HS – heat source, HP – heat pump, C_{HP} – HP condenser, Ev_{HP} – HP evaporator, C – compressor.

The useful effect which can be received because of technogenic air emissions utilization by a heat pump, taking into account energy expenses for the HP compressor drive, can be presented the following way

$$Q = Q_{ut} - \frac{L_c}{\eta_{CPP}\eta_{EPL}}, \tag{1}$$

where Q_{ut} – heat flow utilized while cooling of technogenic air emissions, kW; L_c – energy costs for the drive of the HP compressor, kW; η_{CPP} – energy efficiency of a condensing power plant, it is taken equal to 0.38; η_{EPL} – efficiency of power lines, it is taken equal to 0.95 [3].

Fig. 2, based on the numerical analysis, shows the graphic dependence of specific useful effect which is received as a result of technogenic air emissions utilization by a heat pump, taking into account energy expenses for the HP compressor drive. The calculation was performed at the design ambient temperature of -20 °C.

From the fig. 2 it is established that when utilizing the technogenic air emissions heat at the set temperature inside a heat supply HPS there is an optimum depth of coolant cooling in the HP evaporator, considering conditions for receiving the maximum useful effect and taking into account expenses of primary fuel energy on the HP compressor drive. It is established that the maximum useful effect increases with increasing temperature of technogenic air emissions, which are fed to the inlet of the HP evaporator at a temperature in the range of 20-60 °C. Also noted, that the specific useful effect increases with decreasing design temperature of the coolant in the low-temperature heating system t = 40, 50, 60 °C.

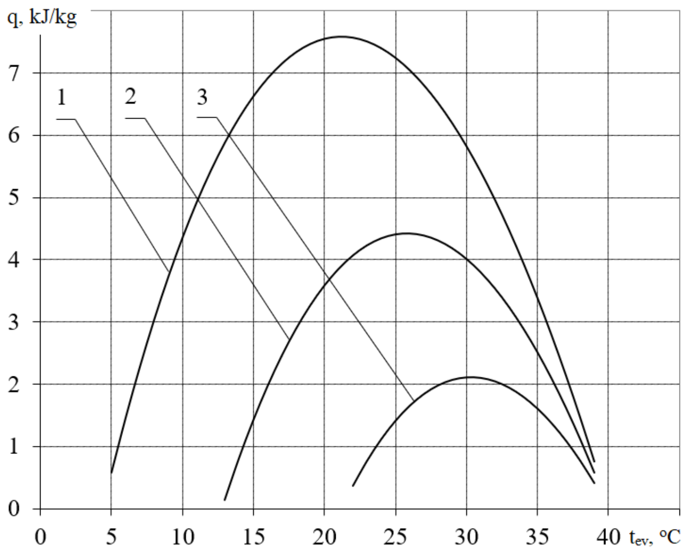

Figure 2. Specific useful effect received as a result of technogenic air heat sources utilization by a HP depending on the temperature at the outlet of the HP evaporator, at the temperature of technogenic air emissions at the inlet to the HP t_1 = 40 ºC:
1 – 3 - at the coolant design temperature in the heating system t = 40 ºC; 50 ºC; 60 ºC.

The dependence of the technogenic air emissions heat utilization optimal degree in the HP heating system on the input parameters is determined: emission temperature, design temperature of the coolant in the heating system and ambient temperature. It is established that the optimal value of the technogenic air emissions temperature at the outlet of the HP evaporator increases with increasing emission temperature and the design temperature of the coolant in the heating system and decreases with increasing outdoor air temperature.

Conclusions. The thermodynamic analysis of conditions for efficient applications of technogenic air emissions heat in the heat pump system for low-temperature water heating is implemented. The main condition of the analysis was optimization of the system parameters based on the received maximum useful effect taking into account the energy costs for the drive of the heat pump compressor.

The implementation of a heat pump heating system with optimal conditions for utilization of the lower heat source will provide the maximum energy effect from the utilization of secondary energy resources like technogenic air emissions heat.

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FEATURES OF RESIDUAL SERVICE LIFE DETERMINATION FOR HIGH PRESSURE ROTORS OF K-200-130 AND K-1000-60/3000 TURBINES

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Abstract: So far, certain approaches have been developed to extension of service life of equipment in the different stages of metal physical exhaustion. The possibility of defining operating conditions of plant equipment beyond the fleet service life becomes even more relevant with increased operating time. The service life is determined as an individual one and is assigned based on the results of individual an inspection of a separate element or the largest group of single-type equipment elements of the considered plant. The fleet service life being reached is followed by diagnostics of specific units of power installations and analysis of their operation, measurement of actual dimensions of components, examination of structure, properties and damage accumulation in the metal, non-destructive testing and estimate of stress strain state and residual service life of the component. The results of performed studies are used to determine an individual service life of each element of energy equipment.

Keywords: *service life, steam turbine, temperature, boundary conditions, ANSYS*

Introduction

The present situation in the energy market of Ukraine demonstrates the need to increase operating capacities, thus requiring renovation or complete replacement of equipment of thermal power plants during the next years. This would allow lifetime extension of thermal power units, including expanding installed capacity and load-following range, as well as decreasing the specific fuel consumption per kWh production.

The problem of defining service life of nuclear power plants considering life cycles of their major equipment is becoming increasingly important each year. This raises questions relating to reasonable decision-making scheme on the due time for decommissioning of NPPs and feasibility of replacing of any major equipment considering safety and economic factors.

Purpose and research objectives

The purpose of the paper is to justify a comprehensive scheme to assessment of residual service life of steam turbine rotors and extension of the operating life [1].

The set purpose is to be achieved by reaching the objectives as follows:

4. analysis of the known ways for service life extension of energy equipment that has reached the end of its fleet service life;
5. the results of metal inspection throughout the entire operating lifetime and analysis of technical audit data relating to damages and geometry changes during refurbishment of steam turbine elements;
6. analysis of the results of experimental researches and estimate of residual service life of steam turbines considering actual operating conditions and local damages of separate turbine components;
7. elaborating proposals regarding approaches to extension of service life of steam turbines.

Material and research results

Initial data. The condensate steam turbines of thermal and nuclear power plants with high temperature elements in 3-D setting are considered. The boundary conditions are established for heat exchange on the rotor surfaces using ANSYS digital model based on built geometrical 3-D models corresponding to operating modes by start-up types from cold, hot and warm conditions and stationary mode.

Model description. In the first phase of the calculation a method for building spatial analogues of turbine machine elements was developed using Solidworks for high and medium pressure rotors.

In the second phase a method for solving non-stationary thermal conductivity equation was developed and boundary conditions of heat exchange on the surfaces of rotors were established using the ANSYS digital model based on 2-D and 3-D geometrical models. The boundary conditions (BC) correspond to operating modes by start-up types from cold, hot and warm conditions, stationary regime [2].

The problem of non-stationary thermal conductivity of steam turbine elements is solved with the equation type as follows:

$$div\left(\lambda grad\ \ t\right) = c\gamma\frac{\partial t}{\partial \tau} \tag{1}$$

where λ, c, γ, - are temperature functions and coordinates under the initial conditions $t_0 = t(x,y,z,0) = f_0(x,y,z)$ and the boundary conditions of the I, II, III or IV kind.

The non-stationary boundary conditions of the I - IV kind were set with due account of operating transients for the surfaces of spatial geometrical models of LPC and IPC.

When defining the BC for non-stationary operating modes the estimate of steam temperature in transient modes on the surfaces of steam turbine elements was applied. Under rapid changes of operating mode of the turbine one can observe a fast change of steam temperatures in its flow section. It has been established by experimental means that in the initial phases of power unit start-up the temperatures of main and reheat steam measured by regular sensors are lower than real temperature values of the steam both by their change speed and statically. Therefore, it was suggested to use the following method for estimation of temperature under transient modes of steam turbine (based on example of calculation of steam temperature in the control stage chamber that almost coincides with the temperate after the control stage).

The third phase implies application of the ANSYS digital model to determine the stress strain state of the HPC and IPC rotos with account of their complex spatial geometry, damages during operating period, repair and restore modifications of design geometry. The temperature gradient is used as a criterion to determine stresses when analyzing their behavior in the high temperature elements of steam turbine for operating modes. The distribution of stress strain state was calculated for the moments when temperature gradients reached extreme values.

The stress strain state was determined for:

- pressure loads;
- temperature loads;
- loads of centered forces (ω = 3000 rpm);
- loads of gravity forces;
- reaction resistance.

The calculations included defining principal stresses, stress intensity for the entire period corresponding to start-up and stationary operating modes in all division points of the high temperature elements of the steam turbine.

That start-ups of the turbine from different thermal conditions can be accompanied with thermal stresses on the rotor surface, which increase the yielding limit leading to residual strain of the metal. If physical and mechanical properties are come out in different values of the yielding limit σ along the cross section area of the shaft, then after reaching σ multiple times a summing of residual axial strains appears, and this is one of the causes of rotor bow.

In the fourth phase a methodological approach was developed to calculate the low cycle fatigue using a software complex of NTUU KPI and ANSYS digital models with application of the calculated change of the stress strain state of HPC and IPC shells and rotors and with account of optimized strength margin ratios by the number of cycles and deformations [3].

Analysis of the study results. The capability of forecasting the value of the residual life is provided under the conditions as follows:

- known parameters defining technical condition of the equipment;
- known criteria of the boundary state of the equipment;
- it is possible to perform periodical or continued inspection of the technical state parameters.

The remaining operating life before cracks $[G]_{\text{ост}}$ (in years) is determined with the formula:

$$[G]_{\text{ост}} = \frac{1-\Pi'}{\Pi''_{\Gamma}}, \tag{2}$$

where Π' - total damage accumulated in the metal of rotos and shells operating under combined creep implication in the different modes of q′ types and cycle loading under different transient modes of k′ types;

Π''_{Γ} - the forecasting average annual loading for the operating period following the analysis that will be accumulated in the considered area of the rotor and shell during the alternation of q" types of sustainable mode and k" types of cycle.

All values relating to the period of operation after carrying out an estimation and continuation of a resource are marked by two strokes.

Conclusion

When forecasting, depending on the service life of the equipment, two approaches are used. Fort short service life (relatively to fleet) and insignificant damage of the equipment only the information on loading is used for forecasting of its residual life. With a service life close to the fleet or significant damage to the equipment elements the degree of equipment damage is additionally investigated. The advantage of the first approach is its lower complexity, the second, which we adhere to – is a more accurate forecast allowing to identify additional reserves of equipment life.

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POWER DISTRIBUTION EFFICIENCY AND RELIABILITY RAISING BY USING THE VACUUM RECLOSERS

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Topical matter of power supply for today is effective increase in the reliability of power supply in medium voltage overhead power systems by sectioning of lines with switching devices, such as disconnectors, controlled disconnectors or sectioning points. In such schemes, the manual approach to emergency management is used. This kind of schemes can be used where the overhead power lines are. Protective device on the outgoing feeder is switched off as soon as damage occurs in any area. As a result, all consumers of the line lose power for a long time. Remotely operated disconnectors or remote controlled sectioning points can also be installed instead of manual line disconnectors.This process of damage localization differs only in that all switching operations are performed remotely. Decision on switching is made by the dispatcher, constant communication with each controlled element of the network is necessary, otherwise it becomes virtually uncontrollable and the entire effect of remote control of disconnectors is eliminated.

Reclosers of E.NEXT-Ukraine Company and "Igor Sikorsky Kyiv Politechnic Institute" are self-contained small-sized complete switchgears with great functionality Fig. 1.

Fig. 1 Installation example for reclosers, switching module and control unit.

The main idea of using reclosers is the following [1, 2]:

- One of the main problems of today's electric power industry is the frequent emergencies on medium voltage overhead lines. This is due to their considerable length and high wear and tear of the equipment of consumers connected to them. Therefore, power supply companies require the installation of sectioning devices on the overhead power lines of consumers, automatically separating this line from the general power grid in case of emergency situations on it. This kind of devices are the reclosers.
- In case of short circuits on the power line protected by the recloser, the fast-switching vacuum circuit breaker protects the fuse link of the tap-off fuse. And only on the 2nd or 3rd automatic reclosing cycle (depending on the setting of the microprocessor protection of the recloser), when it is already possible to talk about the stability of the circuit, the device allows this insert to burn out.
- In addition to the protective and sectioning functions, the reclosers of the E.NEXT-Ukraine Company and "Igor Sikorsky Kyiv Politechnic Institute" can be used for remote monitoring and logging of the quality of supplied electricity, metering its consumption, including it being a part of automatic metering and telemechanics systems. It is possible to enter automatic transfer switches and backup power system with help of them.

Using of reclosers of E.NEXT-Ukraine Company and "Igor Sikorsky Kyiv Politechnic Institute" significantly increases the reliability of the network, reduces the costs of its maintenance and losses from possible undersupply of electricity to the consumer, and allows keeping electricity metering at the border of consumers balance inventory. Currently, about 40% of overhead lines

(OHL) have reach the end of its service life and more than 80% are in need of technical re-equipment.

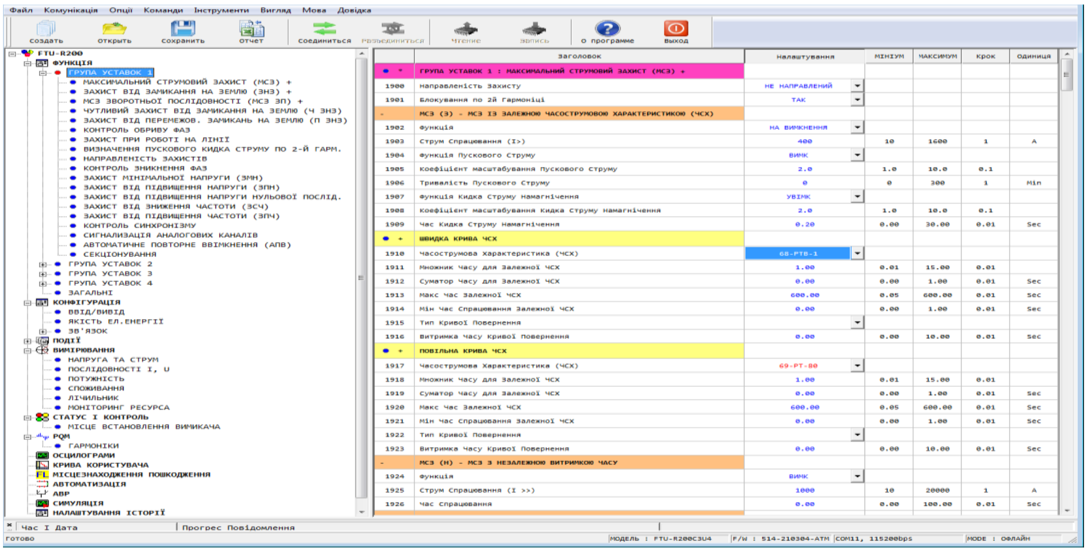

Fig. 2 Mnemonic frame from SCADA system.

The weakest link in the power supply system is 6 (10) kV overhead distribution networks. However, this requires large costs during the construction phase, and sometimes is not feasible due to the complexity of the placement [2, 3]. Also, in order to increase the safety of power supply to consumers, it is possible to repeatedly reserve and section the lines with manually operated disconnectors. But this method also has disadvantages. Therefore, the reliability of power supply of such power lines is increased by sectioning it into several relatively short sections with the installation of intermediate automatic protective switching devices-reclosers. Reclosers of E.NEXT-Ukraine Company and "Igor Sikorsky Kyiv Politechnic Institute" are small circuit breakers located at the top of distribution poles and are usually used on very long distribution feeders.

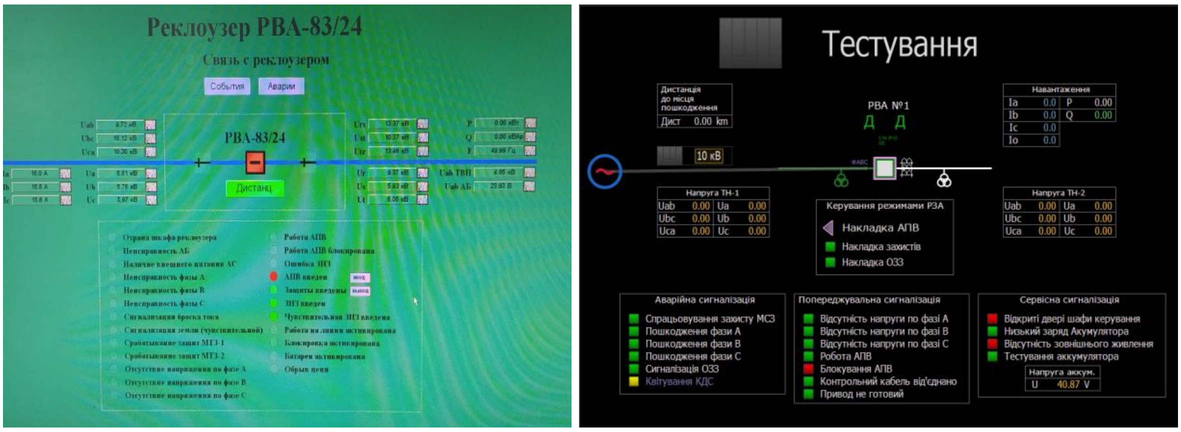

Fig. 3 Mnemonic frame from SCADA system.

Their function is to isolate the feeder section in the event of a malfunction or overload and thus minimize the number of unattended customers. Because they act like small circuit breakers, they have the ability to automatically restore power in situations of temporary failure, hence the name "recloser". This device is remotely controlled and allows the electrical network manager to detect a fault on the overhead line directly at the time of the fault, make decisions quickly and send the emergency repair team to the right area.

Recloser of E.NEXT-Ukraine Company and "Igor Sikorsky Kyiv Politechnic Institute" includes: - vacuum (SF6) switching device; - system of primary current and voltage converters; - autonomous operational power supply system; - microprocessor relay protection and automation system with the ability to connect telemechanics systems; - a system of ports for connecting telemetry devices; - software complex. The advantages of the developed recloser [3, 4].

1. Installation of poles. Reclosers have external (external) pole installation, and due to this:

- Increased level of insulation - the insulation of the poles of the switching module is made of epoxy resin, which has high insulating properties, resistance to ultraviolet radiation, and the ability to self-cleaning from precipitation and pollution.

- No risk of internal short circuit - in the event of an internal fault or lightning strike in the switching module, a short circuit will not occur, since the poles are insulated with solid insulation without the risk of explosion. On the other hand, reclosers with indoor poles have a high risk of explosion.

- Maintainability - in the event of a malfunction of one of the poles, it is possible to quickly replace the recloser pole, which is cheaper and more practical with a long service life, in comparison with the internal version, where this is not possible, in case of a malfunction, the entire switching module is replaced.

2. Drive mechanism. In the proposed reclosers, a spring drive mechanism is installed, which makes it possible to manually turn on and off the recloser in the presence of voltage on the line, while it does not need the presence of an auxiliary power supply, which cannot be done with a magnetic drive mechanism. Also, the latter requires frequent checking of the capacitor, which may lose capacity, which is likely under unfavorable climatic conditions (high temperature). The spring-loaded mechanism of the drive provides a higher mechanical pressure on the power contacts, which minimizes the risks of contact welding, and also withstands a higher short-circuit current compared to a magnetic drive. The spring-loaded drive mechanism is used at high-voltage switchgear / substations, which confirms the reliability and durability of this drive mechanism.

3. Current measurement. Reclosers use built-in current transformers (CTs) to measure current, which provide a whiter class of accuracy than Rogowski coils. The error in measuring the phase currents for CT and Rogowski coil is 0.1% and 1%, respectively, when measuring a single-phase earth fault, the error for CT and Rogowsky coil is 0.01% and 0.2%, respectively, which is a very important factor in networks with an isolated neutral LEP 6-35 kV, where earth fault currents are small compared to phase-to-phase short-circuits.

4. Body material. The recloser body is made of expensive 304 stainless steel, 4 mm thick, powder coated, this will ensure a long service life even in the most aggressive environments, compared to the low grade stainless steel body.

5. Auxiliary transformer (TSN). Complete with reclosers, single-phase TSNs with built-in fuses are used, with the ability to mount on the recloser body, which minimizes the time and material costs for installing the recloser on the power transmission line support.

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THE INFLUENCE OF GEOMETRIC CHARACTERISTICS OF THE BUILDINGS FACADES ON THE HEAT TRANSFER TO THE WIND FLOW

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Introduction. New materials, technologies and architectural solutions in construction can reduce energy consumption during their operation by 50-75% compared to 2000 levels, and the renovationof existing residential and industrial installations can still reduce energy consumption by 30% [1]. Together, this will significantly reduce energy costs, make a significant contribution to reducing environmental impact and climate change, and improve indoor climate conditions.

Ideas for the energy efficiency of buildings are particularly relevant in the design phase of a new building and in the selection phase for renovation of existing buildings. The right choice of architectural solutions for new structures or reconstruction options can provide modern computer modeling, especially modeling of heat and mass transfer processes in buildings and structures in interaction with the environment. Work [2] analyzes the most common programs for calculating the energy consumption of buildings. Using these programs, the thermal, light and acoustic properties of the building are simulated and the thermal and electrical consumption of the building is estimated.

This paper examines the process of convective heat transfer between the external surfaces of buildings and the environment. In most cases, the convective heat exchange of building facades with the environment is estimated by calculating the value of heat transfer coefficients:

$$\alpha_k = \frac{q_s}{T_s - T_a}$$

where q_s is the boundary heat flux on the surface of the enclosing structures; T_s - temperature of the outer surface of the building; T_a - ambient temperature. The more accurately the values of the heat transfer coefficients are determined, the more accurate will be the assessment of the heat taken from the building envelope.

In works [2, 3], the analysis and classification of a significant number of calculated dependences of heat transfer coefficients on building facades is performed. Most of these relationships are used in the above-mentioned software packages for modeling the thermal state of buildings. The stated ratios for the heat transfer coefficients make it possible to take in to account a sufficiently large number of factors that determine the heat flow from the facades of buildings located in the wind flow. However, most of these dependencies do not take into account the type of building (high, medium or low-rise) and its geometry. The dependences of the heat transfer coefficients from the surfaces of building facades on their geometrical characteristics are given in [4, 5]. In these studies, 3D CFD modeling of three groups of free-standing buildings was performed. The first group includes buildings whose height (H) is less than its length L. The second group includes buildings for which H≥L, and the third includes buildings for which H=L. As a result, the power-law dependence of heat transfer coefficient on the wind speed was obtained for the windward facades of buildings.

The main goal of this work is to establish the dependences of the convective heat fluxes on the surfaces of building facades in a wind flow on the basis of three-dimensional CFD modeling with the main architectural features of buildings. In contrast to the works [4, 5], in which the dependence of the heat transfer coefficient on the area of facades, and not on the height and length of buildings, was obtained, in this study the area of facades and roof remains unchanged at various ratios between the height H and the length L of buildings.

Results: Numerical studies were performed for 6 models of buildings with the same surface area (2640 m^2) in wind flow in different ratios between the height H and the length L of the buildings. The width of the building B = 18 m remained constant for all models. It is assumed that the buildings have brick walls 0.3 m thick. The wind flow is directed parallel to the side of the building, the length of which is L. The dimensions of the calculation domain were determined in accordance with the recommendations [6].

The problem of aerodynamics and heat transfer in the wind stream is described by a system of equations consisting of the continuity equation, the equations of turbulent momentum transfer and energy transfer. The system of equations is closed by the equations of the k-ε model of turbulence for the boundary layer of the atmosphere [7]. The heat transfer in the wall of a building is described by the heat conduction equation. To solve the problem at the entrance to the computational domain, the distributions along the height of the average horizontal wind speed U, the kinetic energy of turbulence k and the rate of scattering of the kinetic energy of turbulence ε are specified. The temperature of the flowing wind is T_a = –10 ° C. The temperature is also set on the inner surfaces of the walls of the building + 15 ° C (boundary conditions of the first type). Conjugation conditions (type 4 conditions) are set on the external surfaces of building facades.

Several models of the network were used to numerically solve the problem of aerodynamics and heat transfer in the wind flow around the building: a rectangular uniform network, a rectangular network with reinforcement on the façade surface and a network with polyhedral cells. The determining cell size in all models corresponded to 0.5 m and the number of cells ranged from 2.4×10^6 to 4×10^6. Sensitivity analysis of the solution to the grid models and to the number of cells in the models showed a slight difference in the results.

Numerical modeling using CFD packages allows calculating fields of all functions and unambiguously determining local and average values of thermal characteristics of the modeled object. Fig. 1 shows the average values of the heat flux density q_s over the surfaces of all facades depending on the height of the building H and the wind flow speed. The data suggest that as the wind speed increases, the heat flux density value increases. In the range of building heights from H = 10 m to H = 16 m, the average heat flux decreases with increasing height, and with a further increase in height H at all wind speeds, a slow increase in q_s is observed.

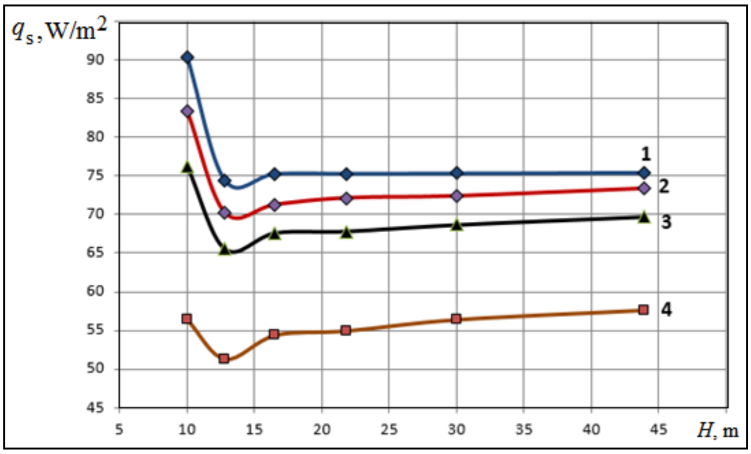

Fig. 1. Average heat flux q_s on the surface of buildings, depending on the height of the building and wind velocity: 1 - U_{10} = 10 m/s; 2 - 5 m/s; 3 - 3 m/s; 4 - 1 m/sec.

The average values of heat transfer coefficients on the facades of simulated buildings obtained as a result of modeling differ significantly from the known models given in [2, 3]. For a building with a height of H = 16.55 m at a wind velocity of U_{10} = 5 m/s at a height of 10 m from the ground surface, the average heat transfer coefficient on the windward side of the building facade was α_k = 36.9 W/(m^2K), while the known McAdams model [2] gives the value α_k = 25.2 W/(m^2K). On the roof of the building, McAdams model provides a 17% lower heat transfer coefficient than the simulated value and 29% higher than on the leeward façade.

Conclusion and findings. Most empirical models, such as the McAdams formula, designed to evaluate the convective component of heat flow on the outer surfaces of building envelopes, take into account only 1 ... 2 parameters from a large number of factors that determine the convective heat transfer of a building in the wind flow of the atmospheric surface layer. At the same time, current CFD modeling packages make it possible to take into account a much larger number of factors influencing convection, including the geometry of buildings and their architectural elements, the influence of which was unequivocally confirmed by the results of this study (Fig. 1).

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THE USE OF ENERGY STORAGE TO CONTROL THE ELECTRICAL LOAD OF THE POWER SYSTEM UKRAINE

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In the energy system of Ukraine in 2020, the share of wind power plants (WP) and solar power stations (PV) in the structure of electricity production has doubled - to 6.8% with a total electricity production of 148.9 billion kWh. The installed capacity of these renewable power plants increased by 1.9 GW during the year.

Increasing the installed capacity, primarily PV, given their stochastic nature of operation, significantly complicates the modes of operation of the power system. In 2020, electricity consumption decreased by 2.7% in the spring months (March-May), when PV energy production increases seasonally. The decrease in consumption at certain times of the day was 7%. The daily schedule of electricity consumption also changed:

• the difference between the minimum and maximum value of daily consumption capacity increased by an average of 300 MW, and in some months - by almost 600 MW;

• the level of electricity consumption during daytime hours decreased the most (for example, in May at 12:00 (in the hour of maximum PV load) the average consumption of working day decreased by 1.1 GW (from 16.1 to 15 GW).

The peak of PV generation due to high solar activity occurs during the day, when the power system shows a decrease in consumption, compared with the morning and evening peaks. To maintain the balance in the power system and ensure operational safety during this period of the day, the dispatchers of the national power company "Ukrenergo" have to:

• apply all proposals of thermal power plant and hydroelectric power plant for unloading within the balancing market

• issue operational safety commands to pump pumped-storage hydroelectricity to increase consumption.

If these measures are not enough, the limits of renewable power plants apply, the maximum total value of the limit was 2178.86 MW (June 7, 2020) (Fig 1.) [1].

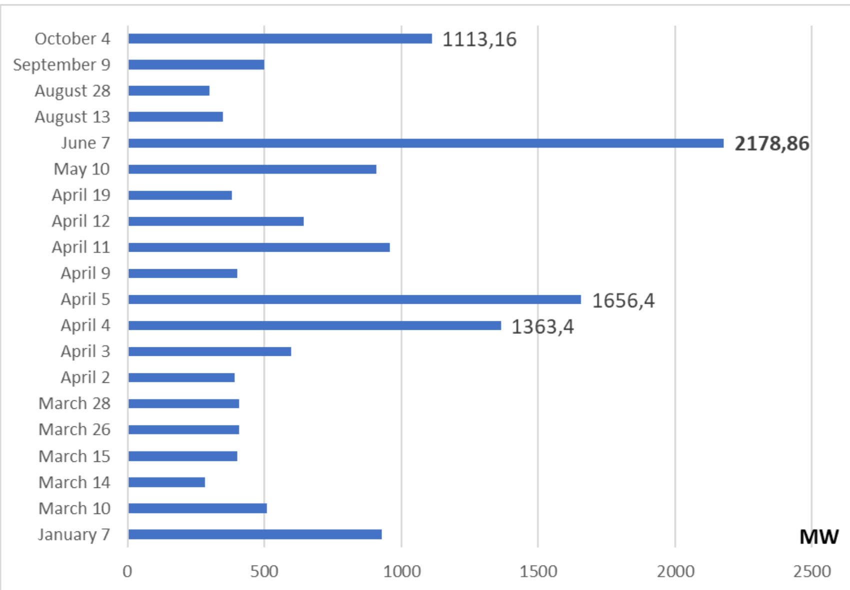

Fig 1. *Restrictions of RE in the power system of Ukraine in 2020*

The balancing of renewable power plants, mainly PV, during the day and remains today the main problem of integration of renewable energy into the energy system of Ukraine.

Therefore, there is a need to increase the flexibility of the power system with the intensive growth of the share of renewable energy in the generation structure, to avoid the need to limit electricity production WP and PV and ensure the safety and reliability of the power system. Therefore, highly maneuverable capacity and sufficient capacity reserves will allow to move to flexible energy management and operational security. Such technologies may include the use of Energy storage (ES) [2,3].

To determine the possibilities of using ES, which can take part in adjusting demand for power. Consider a typical daily schedule of electricity consumption in the Integrated Power System of Ukraine and the generation of WP and PV. Taking into account the variability of renewable energy, we determine the areas in which ES can be used in the charge -CHG and recharge -RCHG mode. (Fig 2.)

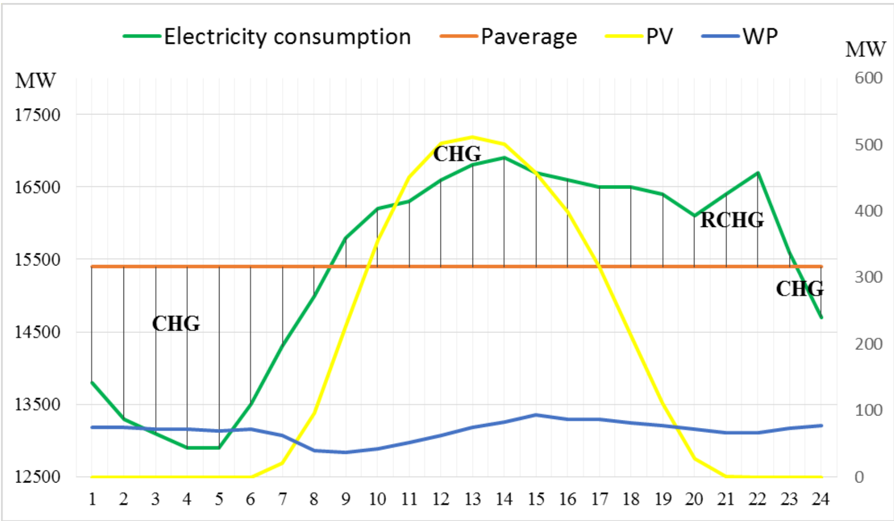

Fig 2. *Graph of electricity consumption, generation WP and PV typical summer day*

The given graph shows one of many variants of possible interaction of the power system with electric vehicles as aggregators. Such work depends on many factors associated with the power structure structure and characteristics of the ES, namely the installed power of the battery and their time of interaction with the network, represented:

$$Pes = (N, C_b, T) \tag{1}$$

where: N – number of ES, C_b – battery capacity, T – network interaction time (charge / discharge).

Therefore, we consider the methodology for assessing the potential power of ES in general generating capacities system for the Integrated Power System of Ukraine.

Thus, in general, the supply-demand balances for the n-degree electrical load for a characteristic day *g*, $n=1 \div N$, $g=1 \div G$, in *t* stage of the calculation period $t=1 \div T$, in a simplified form, are formalized as follows:

$$\sum_{k=1}^{K} Y_{kngt} - \sum_{i=1}^{I} D_{ingt} - HM_{ngt} + HP_{ngt} = 0 \quad (1)$$

where: *Y* – power of *k* type of generation technology, which is used to cover demand at the appropriate level of the GEN and is determined taking into account own needs for electricity generation and availability factors that determine its "availability" for use, *k=1÷K*, *D* – demand for electric power of i-th consumer, taking into account losses on transportation and distribution of electric energy, *i=1÷I*, HP and HM are variables that provide an opportunity to get a solution in the event of the balance disturbance, due to the impossibility of ensuring that it is executed at the given limits for the installed generation power and accepted readiness rates

In more detail, the methodology for modeling the coverage of electrical load using separate (typical) technologies for generating electricity and taking into account the ES [4]

Conclusions. The given method can be used for simulation modeling at the country level and region power system to assess the possibility of ES participation in the management of electricity consumption.The introduction of ES will provide the opportunity to take advantage of all ES benefits and the realization of power consumption flexibility potential and the grid will receive an effective mechanism for solving power supply management issues.

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ENERGY EFFICIENCY ANALYSIS OF THE VENTILATION AND AIR CONDITIONING HEAT PUMP SYSTEM INSIDE A PRODUCTION PREMISE WITH RECUPERATION OF EXHAUST AIR

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Introduction. Following the 2015 UN Framework Convention in Paris, the European Union has set a goal to reduce CO_2 emissions by 80 % by 2050 (Aguilar, et al., 2019). To achieve these goals it was proposed to use renewable energy sources and electricity instead of fossil fuels for purposes of the heat supply and the cold generation. Today, the application of heat pumps (HP) continues to grow, as this technology offers the ability to provide ventilation and air conditioning in buildings, reducing the consumption of primary energy compared to traditional systems (Mazzeo, 2019). However, issues of proper integration and productivity increase of HP are still important for energy saving.

The current research proposes a heat pump system (HPS) for ventilation and air conditioning purposes with recuperation of cold exhaust air to precool fresh supply air. The expected effect of recuperation is the load reduction on the HP evaporator, and, as a consequence, the decrease in the compressor load and the increase in the energy efficiency of the HPS in general. However, the heating of the exhaust air in the recuperator before the HP condenser is a limiting factor, as the air temperature after the condenser can exceed critical values (about 70 °C) which can lead to failure of the HP equipment (Morozjuk, 2006). Therefore, a special research is required to verify advantages of using a recuperator in the studied HPS.

As a result of the study energy efficiency and operating modes of the HPS during the warm season were estimated, depending on parameters of the outside air, the recuperation efficiency and characteristics of a production premise. The theoretical model of the HPS is developed and its thermodynamic analysis to define air parameters in the system's nodal points and maintenance conditions of supply air parameters is carried out.

Research results. A schematic diagram of ventilation and air conditioning HPS with exhaust air cold energy recovery and with a variable ratio of fresh supply air is shown in Fig. 1. The principle of system's operation is as follows: fresh supply air with temperature t_0, moisture content d_0 and mass flow rate G_0 enters the recuperator, where it is cooled at a constant moisture content to temperature t_{pc} by cold ventilation emissions from the premise with temperature t_2, moisture content d_2 and mass flow rate G_{tot}. The cooled supply air flows to the HP evaporator, where it is further cooled with a partial water vapor condensation to a temperature t_{ev} and a moisture content d_{ev}. The preheated exhaust air after the recuperator at a temperature t_{ph} and a constant moisture content enters the HP condenser, where it is further heated and at the outlet has a temperature t_c with a constant moisture content d_2. The heated exhaust air is then divided into two flows: one flow with a mass flow rate G_0 is discharged into the environment while the second one is fed to the mixing chamber, where it mixes with the cooled supply air after the HP evaporator. The resulting mixture of air (t_1, d_1, G_{tot}) is then supplied to the production premise for ventilation and air conditioning purposes (to compensate for internal and external heat flows).

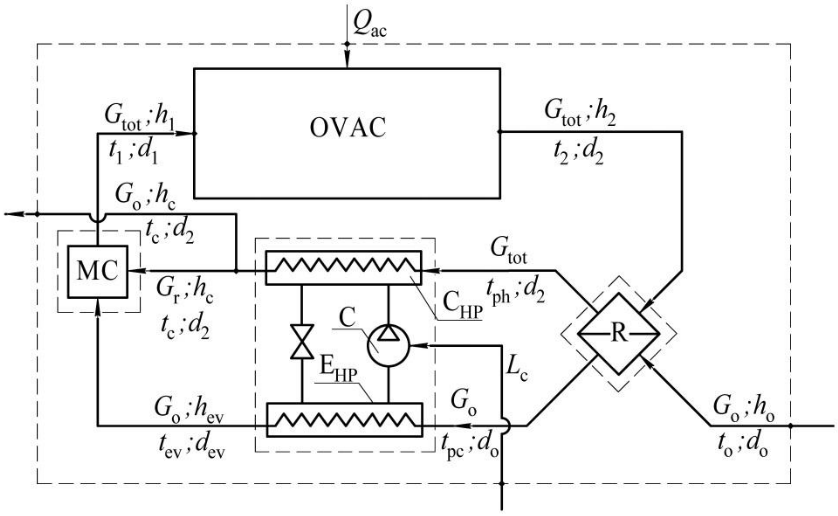

Fig.1. A general design of the ventilation and air conditioning HPS inside a premise with an excessive internal moisture generation: C_{HP} – a heat pump condenser; E_{HP} – a heat pump evaporator; C – a compressor; MC – a mixing chamber; OVAC – an object of ventilation and air conditioning.

Calculation analysis of the HPS parameters was performed for a typical production premise with required technological conditions inside. A workshop of the Roshen confectionery factory in the Kyiv region was chosen as a prototype (Bezrodny, et al., 2018). To ensure comfortable working conditions in the premise, the following indoor air parameters were set: temperature t_2 = 18 °C and relative humidity φ_2 = 50 % inside the premise. Fig. 2 shows graphical relationships between HP and HPS refrigeration efficiencies, temperature and relative humidity of the environment, recovery coefficient and various values of K (a conversion factor used for assessment of premise properties). The case when η_r = 0 corresponds to the HPS without prior recovery of the cold energy from the exhaust air.

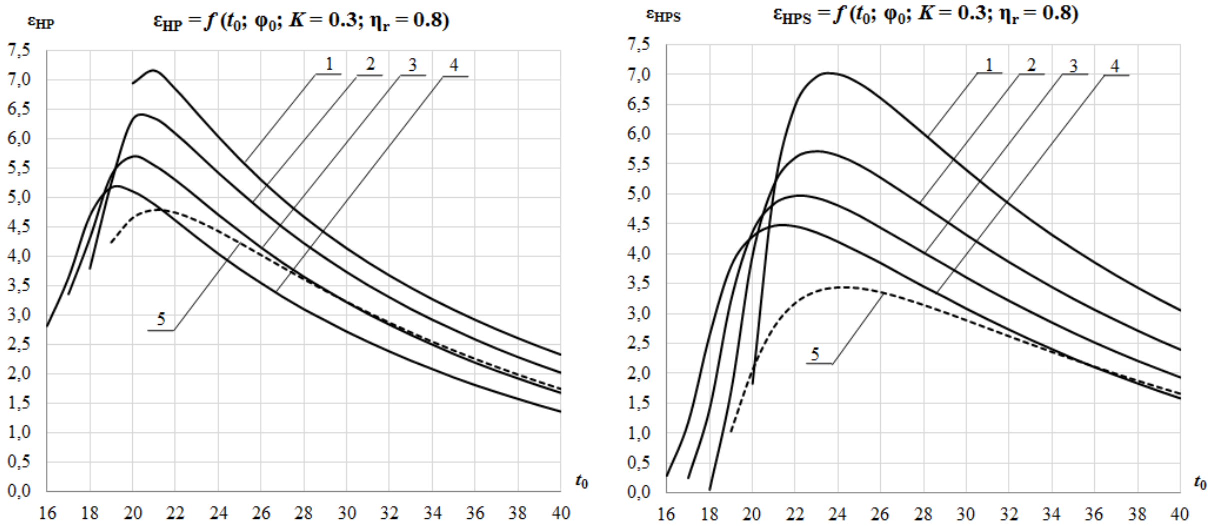

Fig. 2. Relationships between HP (a) and HPS (b) refrigeration coefficients, the environment temperature and relative humidity: K = 0.3 and η_r = 0.8: 1-4 – φ_0 = 40 %, 50 %, 60 %, 70 %; 5 – φ_0 = 50 % and η_r = 0.

Conclusions. The analysis of this ventilation and air conditioning HPS showed advantages of the cold energy recovery, compared to its absence, to provide a higher refrigeration coefficient of the whole system in the operating range of outside air temperatures at considered values of relative humidity. The studied system can work effectively up to some moderate values of external environment parameters which further increase is limited by the maximum temperature of exhaust air after the HP condenser. This HPS has the highest energy efficiency in the area of relatively low outside air temperatures and relative humidity. This suggests that such HPS is suitable for use in countries with temperate continental climates, which are characterized by low relative humidity.

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THERMODYNAMIC ANALYSIS OF THE COMBINED SCHEME USING SOIL AND AIR HEAT PUMPS FOR HEAT SUPPLY OF A PUBLIC BUILDING

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Abstract.

The paper investigates the energy efficiency of using the heating and ventilation scheme of a public building, which consists of a cascade of two heat pumps: air and ground. The main feature of the solution is the series connection of two heat pumps via water and the use of potentially useful properties under different external conditions. In the course of the study, a thermodynamic model of the system was compiled using heat and material balances. Then the system of equations was solved using a numerical method. As a result of the solution, graphical dependences of the determining quantities on the parameters of the system were obtained. In the course of the study, it was decided to consider the efficiency of the system in terms of the depth of utilization of ventilation emissions, namely: minimum external energy costs, equality of conversion factors, equality of ambient air and outlet temperatures. As a result, it was determined that the utilization of exhaust air is advisable only up to the level of equality of conversion factors. Further cooling entails a sharp increase in specific work.

Keywords: heat pump, cascade connection, low-grade energy, utilization, ventilation emissions, ground heat, ambient air, combination of heat sources.

Introduction. According to the statistics of the International Energy Agency (IEA), residential and commercial buildings accounted for 49% of the total final consumption (TFC) of world electricity and 15% of the TFC of fossil fuels [IEA report, 2020]. This indicates that improving the energy efficiency of buildings is critical to addressing energy conservation, emission reductions and global climate change. Currently, heating, ventilation and air conditioning (HVAC) systems in residential and public buildings consume a significant share of the total heat energy consumption [Bezrodny, et al., 2021]. Consequently, energy savings in heating, ventilation and air conditioning systems play an important role in improving the overall energy savings of a building.

In terms of year-round heating demand, the air source heat pump plays an important role in space heating and hot water production [Menberg, et al., 2017] due to its high primary energy efficiency and low pollutant emissions [Christodoulides, et al., 2019]. However, when air heaters are used in winter, low ambient temperatures cause a decrease in the evaporation temperature, which leads to a catastrophic decrease in efficiency, a drop in condensation temperature or even a system shutdown.

However, when using an air-to-water heat pump operating on ventilation emissions at facilities where the consumption of thermal energy by the heating system significantly exceeds that in the ventilation system, the capacity of such a HPU to provide both heating and ventilation may not be enough [Bezrodny, et al., 2018]. This problem can be solved by using cascade heat pumps with air and ground source of heat.

Research results. A schematic diagram of the solution described above for heat supply of heating and ventilation systems of a building using two series-connected heat pumps (air and ground) is shown in Fig. 1.

Analyzing the scheme, you can see that the ventilation emissions with a temperature t_{in} (temperature inside) are injected into the evaporator of the air HP, where it gives off heat to the freon, evaporating it and thereby cooling to the temperature t_{ev}. Next, the coolant is heated in the condenser from the t_r (return water) to the t_g (intermediate). The coolant is heated to the required design temperature t_c by means of ground HP.

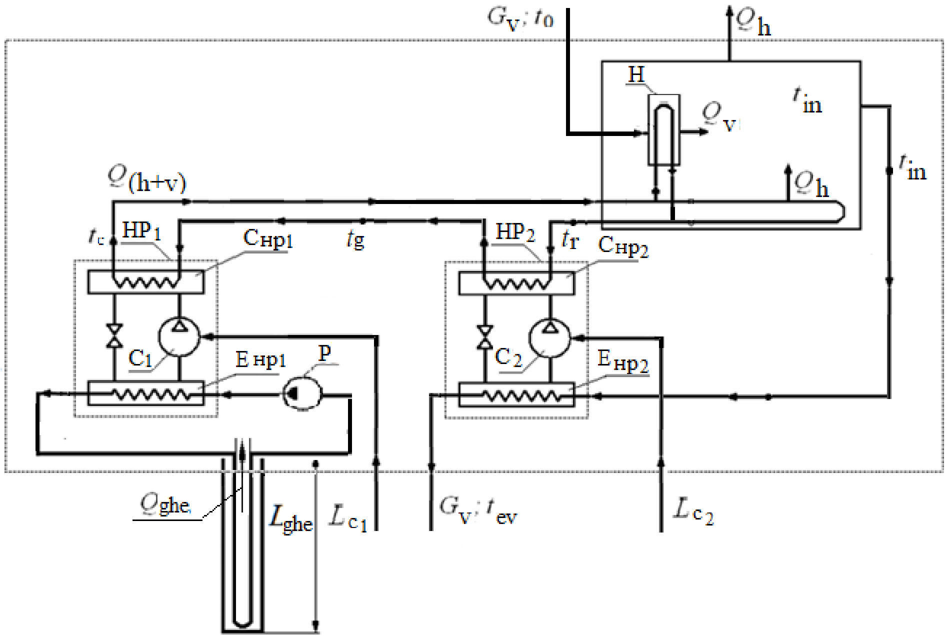

Fig.1. A general design of the ventilation and air conditioning HPS inside a premise with an excessive internal moisture generation: C_{HP} – a heat pump condenser; E_{HP} – a heat pump evaporator; C – a compressor; MC – a mixing chamber; OVAC – an object of ventilation and air conditioning.

A feature of this scheme is the sequential use of two lower heat sources (heat of ventilation emissions and heat of the soil), which, together with the energy of the HP compressor drives, provide the needs of the heating and ventilation system of the facility. In this regard, the problem arises of determining the rational depth of use of each of these sources in order to increase the efficiency of the system as a whole. At the same time, an increase in the efficiency of the system implies a decrease in both energy costs for the drive of heat pumps and a decrease in the share of the heat load covered by the use of soil heat, which is associated with a decrease in capital costs for the construction of a ground heat exchanger.

Conclusions

The dependence of the second important characteristic of the heat pump system - the specific consumption of external energy for the drive of the system - is shown in Fig. 2. It can be seen that the achievement of the COP equality condition (Fig. 2, b) has almost no effect on the amount of external work on the system drive, which does not depend on the ratio of heat consumption for ventilation and heating. At the same time, a further decrease in the temperature t_v to (Fig. 2, c) the level t_0 leads to a sharp increase in the specific work in the region of low outdoor temperatures. As a result, it can be concluded that the utilization of the heat of ventilation emissions in the considered heat pump system may be advisable when the air temperature after the evaporator of the air HP is reduced only to a level corresponding to the condition of equality of the COP of the air and ground heat pumps.

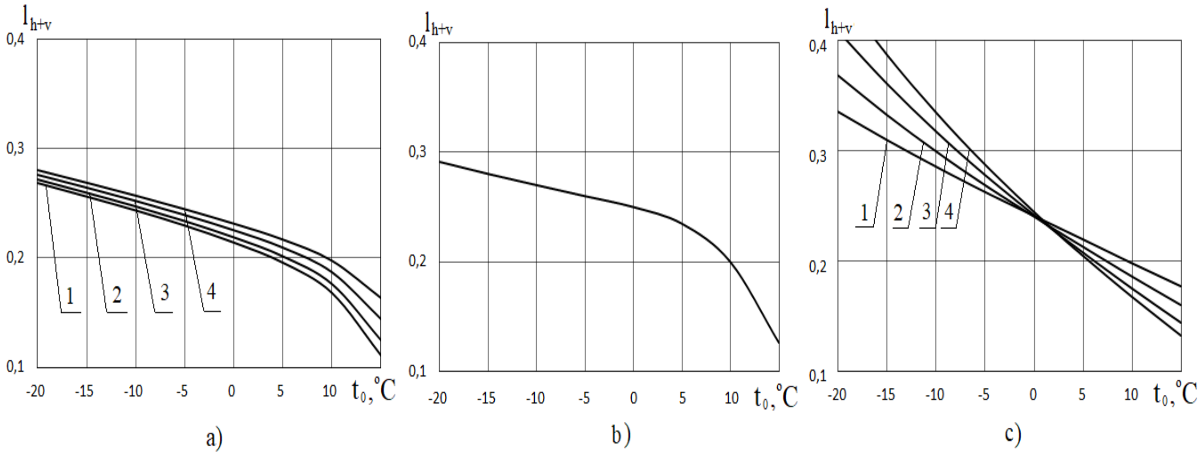

Fig. 2. Dependence of the specific consumption of external energy for the system on the outside temperature: a) condition for minimum unit energy consumption; b) COP equality condition; c) t_{ev}= t_0 condition, where 1-4 – *m*=0,5; 1; 2; 4**.**

The analysis showed that a positive both energy and investment effect can be achieved at the depth of utilization of ventilation emissions, corresponding to the condition of COP equality of air and ground heat pumps.

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PRINCIPLES OF CREATION OF GROUND HEAT ACCUMULATORS

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Investigation of the processes of long-term accumulation of heat and the development of an efficient, environmentally friendly seasonal accumulator of high thermal capacity is an urgent and rather difficult task. Practical implementation will reduce the use of traditional fuels for municipal heat supply, as well as improve environmental performance by reducing emissions into the atmosphere.

To create a seasonal heat accumulator, it is preferable to use natural accumulators. The most accessible and ubiquitous is the natural ground array. Heat accumulation in the ground array is carried out by vertical pipe heat exchangers, through which the intermediate heat carrier circulates, heating the ground array (accumulation) or cooling it (extraction). The most frequently used ground heat exchangers of the "pipe in pipe" type (coaxial), U-shaped (single-loop, multi-loop) and voluminous.

The main part of the heat is accumulated in a volume below the depth of annual changes in ground temperatures. When using U-shaped heat exchangers, due to the counter-current movement in the direct and reverse branches of the same coolant, its design temperature is substantially equalized along the height. The same will happen in a coaxial heat exchanger if the thermal resistance of the dividing wall is small.

The analysis of processes of heat conductivity showed in the unlimited ground array, that:

a) the minimum speed of heat propagation from the accumulator into its environment will be at zero temperature gradients in interacting systems, which leads to the heat drift formula:

$$R_b = \sqrt{\frac{24 a_m t}{i}} \qquad (1)$$

or a flat (i = 1) cylindrical (i = 2) and spherical (i = 3) shape of the heat propagation region;

b) a high-temperature region can be created only with counter-directed heat fluxes.
This is realized when an ordered aggregate of interacting vertical ground heat exchangers is located in the soil massif.
The accumulation area can be conditionally divided into two parts: main (internal), having a constant volume V_o and the highest temperature T_o; buffer (external) with a continuously increasing volume V_b (t) due to the continuous spread of heat in an unlimited soil mass. And since the buffer region arises as a result of the thermal interaction of the main one with the surrounding soil massif, the temperature in it changes from T_o to T_m.

For efficient accumulator operation, it is necessary to minimize the size of the buffer area. The smallest sizes of the buffer region will be in the case of a uniform temperature distribution T_o (t) over the main region V_o. This is possible with the same heat load on each heat exchanger, their uniformity, constant step L of their location, the same potential of the intermediate heat carrier and a small temperature difference at the inlet and outlet of each heat exchanger $\Delta T_в = T_{в.in.} - T_{в.out.}$ ~1 K respectively.

The minimum heat content of the buffer region is found from the geometric relationships at which $E_b / E_a \rightarrow \min$ at $V_b \rightarrow \min$ ($E_a = E_o + E_b$).

In order to reduce the heat flux into the buffer area, it is necessary to ensure its direction from the peripheral heat exchangers only to the inside of the main area and, accordingly, to reduce their heat load in comparison with the internal heat exchangers. When extracting the accumulated heat, it is desirable to make the peripheral heat exchangers of the accumulator inoperative.

If these requirements are met, the temperature distribution in the buffer region is described by the dependence:

$$\frac{T - T_m}{T_o - T_m} = (1-\psi)^3(1+3\psi), \quad \psi = \frac{u}{R_b} \qquad (2)$$

where u is the current linear size, measured along the outer normal of the closed envelope surface of the main accumulation area S_o, and R_b is determined by formula (1).

To organize a ground heat accumulator, you can use the standard shapes of the main accumulation area: a rectangular parallelepiped, a cylinder and a ball. Knowing the required energy content of the accumulator E_a, the thermophysical properties of the ground, the duration of accumulation τ_a and the final temperature $T_o(\tau_a)$, the sizes of the main V_o and buffer V_b regions of the accumulator are determined by the iteration method. According to dependencies (1) and (2), the smallest total volume will be in spherical accumulator , and the largest - in the shape of a rectangular parallelepiped. However, technically, it is difficult and impractical to implement a spherical region using a set of vertical heat exchangers.

When organizing a accumulator, it is difficult to ensure the constancy of the step between heat exchangers of the same heat load. This leads to the intersection of the cylindrical areas of action of the heat exchangers and for this reason it is necessary to abandon this form of accumulator.

When the accumulator is in the form of a rectangular parallelepiped, the implementation of the constancy of the step L is easily achievable. The greatest potential of accumulated energy will be at the minimum heat content of the buffer area. Therefore, the actual problem is to determine the ratio of the dimensions of the main area, at which this minimum is achieved. Examination of the extremum of the expression for the heat content resulted in a cubic shape of the main storage area. The study for the extremum of the expression for the heat content E_b led to the cubic shape of the main storage area. Since the value of the second derivative is always positive ($\frac{\partial^2 \tilde{E}_б}{\partial x^2} > 0$), then there is a minimum ($E_b \rightarrow E_{b,\,min}$). As R_b (t) increases, the heat content of the buffer region increases. In fig. 1 shows a graph of this important characteristic change.

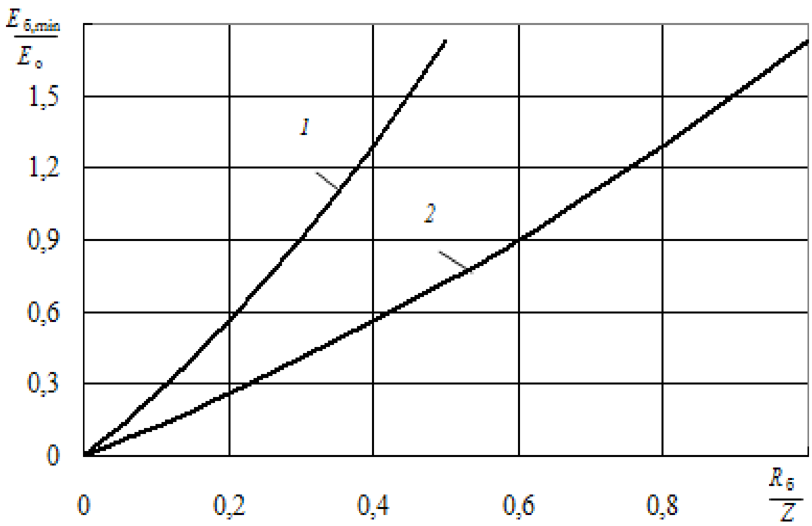

Fig. 1 Relative energy content of the buffer region,
1 - for a cubic accumulator; 2 - for rectangular accumulator

The total number of accumulator heat exchangers *k*, their "effective" numbers during the accumulation of k_a and extraction k_e can be determined according to the dependencies:

$k = m \cdot n$, $k_a = m \cdot n - m - n + 1$, $k_e = (m - 2) \cdot (n - 2)$

m, *n* - is the number of heat exchangers in the *x, y* directions.

According to requirement *a*), it is necessary to fulfill the condition T_o (x, y) = const, which is possible with a uniform heat load of the heat exchangers.

The uniformity of heat distribution can be ensured by forced control or automatic regulation of hydraulic factors. or example, the same hydraulic resistance of the supply and discharge system (SDS) of pipelines for each heat exchanger. The SDS is built according to the parallel-sequential principle with the possibility of periodically changing the direction of movement of the coolant in the entire system to the opposite (reverse). The number of heat exchangers connected in series should be such that the temperature difference of the coolant at the inlet and outlet from the circuit does not exceed 3 - 5 ºC. The resulting small temperature unevenness will change to a compensating opposite one during reverse switching.

In hydraulic calculations, a small speed of movement of the coolant in heat exchangers and SDS is selected. This is due, firstly, to the fact that the thermal resistance of the ground, and not heat transfer, is decisive in the heat transfer link "heat carrier - ground". Secondly, this reduces the energy consumption for the transportation of the intermediate heat carrier in a closed circulation loop.

The arrangement of heat exchangers on the sides of nested regular hexagons can be attributed to some modification of the cylindrical shape with a constant pitch *L*. In contrast to the rectangular scheme, where there is one “effective” heat exchanger per cell with area L^2, in the hexagonal scheme a triangular cell with area $0{,}433 \cdot L^2$ is “served” by half of the heat exchanger.

The total and "effective" number of heat exchangers are calculated according to the expressions: $k = 1 + \Sigma 6i$ $(i = 1, 2, \ldots, k)$; $k_a = k - 3k - 1$; $k_e = k - 6k$.

Since the hexagonal solution was considered as equilateral, it was compared with the corresponding rectangular ($L_m = L_n = l$). With the same areas of the main area the perimeter of the square is 7.3% larger than the perimeter of the hexagon. With the same cell areas per heat exchanger, the number of heat exchangers according to the quadrangular scheme k_4 is 9.4% more than k_6 and this figure decreases with an increase in *k* (for example, at $k = 8$, the value $k_4 / k_6 = 1.056$). The efficiency of using heat exchangers *k* is determined by the values of k_a and k_e. It turned out that the schemes differ little in this indicator.

Thus, it can be concluded that the organization of the batteries in a square or hexagonal scheme is almost equivalent. However, a simpler routing of the coolant distribution system in the case of a rectangular solution makes it preferable, especially with a large number of heat exchangers.

TRANSIENT ENERGY MODELS OF HOUSING FACILITIES OPERATION

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Abstract: The paper analyzes the energy consumption for heating a two-room apartment on the basis of dynamic modeling of energy processes in Matlab software environment and quasi-stationary model EN ISO 13790, which is the basis of the national standard of Ukraine DSTU B A.2.2-12: 2015. In this work, a comparison of the results of modeling, taking into account the thermal interaction between the zones of the building, using a dynamic model was carried out.
Keywords: energy need for heating, controller, internal air temperature, Matlab Simulink.

Introduction

Buildings are one of the main consumers of primary energy resources in the world. The climate conditions of Ukraine are characterized by a long heating period, which causes more than 85% of energy consumption for heating needs, a similar situation is typical for Western and Central European countries. For qualitative evaluation and prediction of energy consumption in short time intervals it is necessary to use dynamic calculation models [1]. There is a large number of dynamic software products, which allow to estimate energy consumption of buildings, such as EnergyPlus, TRNSYS, eQuest, Matlab / Simulink and others [1-3].

Purpose and research objectives

The purpose of the work is to analyze the energy consumption of heat energy for heating a two-room apartment using dynamic modeling, taking into account characteristics of thermal interaction between zones (rooms).

According to the set goals, the following tasks should be solved:

8. creation of dynamic models of the apartment with division into zones in Matlab software environment;
9. creation of quasi-stationary models of the apartment based on the standard DSTU B A.2.2-12: 2015 [4];
10. analysis of energy characteristics of the apartment.

Material and research results

Initial data. Existing residential housing was chosen for the study. It is a two-room apartment, located on the fourth floor of a five-story apartment building in Kyiv, built in 2016. The total area of the apartment is 49,44 m^2, the height of the walls is 2.7 m. The apartment has a window orientation to the east (E) and west (W) sides, as well as a blank outer wall oriented to the north (N). Translucent elements of the enclosures are made of metalplastic two-chamber energy-saving double-glazed windows with argon filling of the chambers. The load-bearing part of the external wall is made of 0.4 m red hollow brick, and insulated with 0.05 m mineral wool. Ventilation is natural with a multiplicity of air exchange of 0.6 $hours^{-1}$. The study used hourly climate data of a typical year of the IWEC international weather file for the Kiev city conditions [5].

Model description. The apartment room was created in Matlab software environment by specifying the enclosure area and thermal resistance, which was introduced through blocks describing the convective and thermal conductivity components, as well as the heat storage properties of the internal and external envelopes, and the air in the room. Rooms are interconnected by thermal interaction. In addition, the hourly amount of heat inputs from the sun to the area of each room and the hourly external air temperature were set. The heat source was a gas boiler in which the mass flow rate was controlled by a valve using an on/off controller, which turned on when the air temperature in the rooms fell below the specified limits and turned off when it rose above. The on/off controller has no intermediate states, either fully on or fully off. The simulation was performed under the condition of maintaining a constant air temperature of 20°C. Fig. 1 shows the thermal energy model of the apartment. The step of calculating the energy demand varied automatically depending on the magnitude of changes of external and internal fluctuations of the input parameters and was in the range of 1-200 sec.

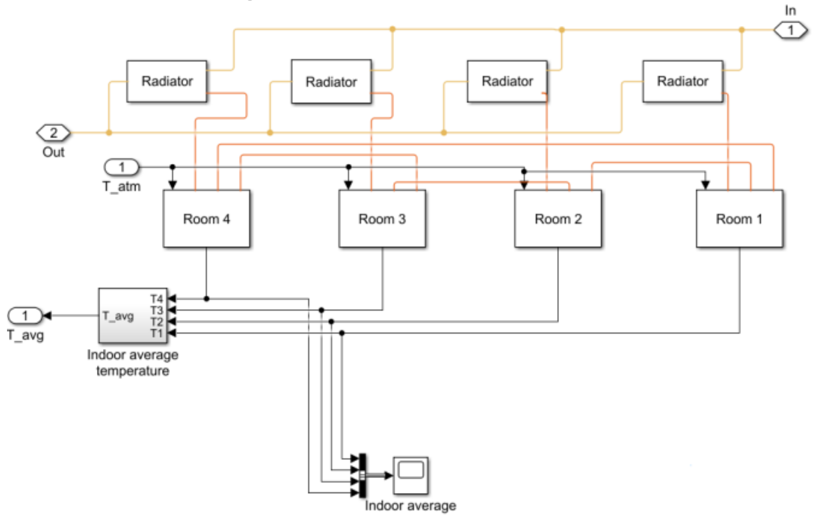


Figure 1. The thermal energy model of the apartment

The mathematical model based on the quasi-stationary method according to DSTU B A.2.2-12:2015 considers the apartment as a single zone and takes into account the thermal inertia properties of the external envelope only. The calculation is performed for monthly intervals, i.e., only seasonal weather fluctuations are taken into account, and daily fluctuations are not taken into account.

Analysis of the study results. Fig. 2 shows the results of the simulation of heat energy consumption for heating needs. In the annual section, the difference of simulation results at constant air temperature in the apartment by the two considered models is 13%, the dynamic model created in the Matlab software environment was selected for the reference model. A smaller difference in the results is observed for the off-season period (October, April) – about 30 kWh, for other colder months it is 80...140 kWh.

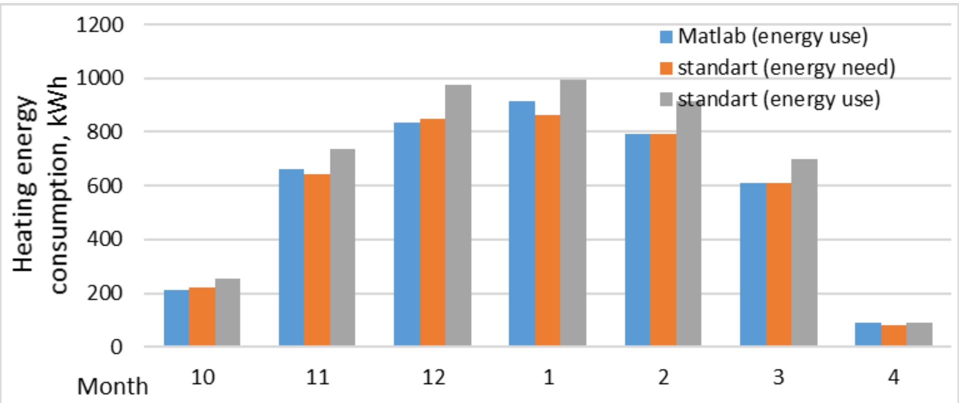
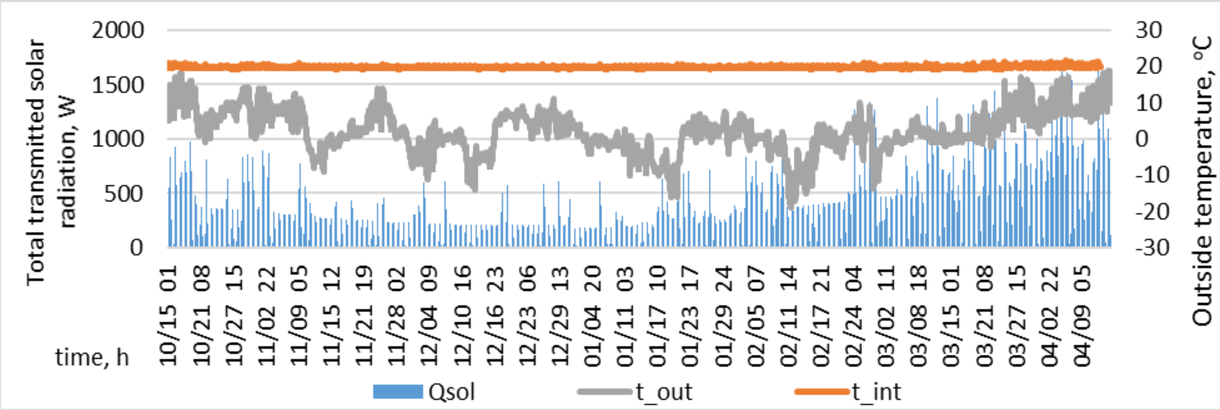


Figure 2. Heating energy consumption

From the mathematical modeling in Matlab software environment it follows that the total heat consumption of the apartment is distributed: bedroom 1 / room4 – 17%, bedroom 2 / room3 – 40%, kitchen / room2 – 31%, common areas (corridor, bath) / room1 – 12%. In bedroom 1, bedroom 2 steel radiators are installed, in the kitchen – steel radiator and water heated floor, in common areas – water heated floor and heated towel rail.

It should be noted that the boiler is controlled by the average air temperature in the rooms of the apartment, i.e. the boiler operates at a constant flow rate, and depending on the conditions, it changes the temperature of the water supply, thereby leading to a situation where the hourly air temperature in different rooms under the influence of, primarily, different heat gains varies from 18 to 24°C, while maintaining the average air temperature at 20 ± 0,5°C (Fig. 3).



t_int – internal temperature, °C; t_out – outside temperature, °C; Qsol – total transmitted solar radiation, W

Figure 3. Maintaining the average internal temperature when changing climatic data

Conclusion

In the study, dynamic modeling of energy consumption of a two-room apartment on the basis of the created model in Matlab software environment was carried out. A comparative analysis of simulation results between the dynamic and quasi-stationary model has been performed. In the future, it is planned to study different types of heating system regulation settings for intermittent and continuous heating modes and to give recommendations for the installation of indoor air temperature sensors, which transmit it to the controller.

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SCHOOL THERMOMODERNIZATION TAKING INTO ACCOUNT CHANGE IN COMFORT LEVEL

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Abstract: *Increasing the level of thermal resistance of the building envelope in combination with the choice of heat source is an urgent task. It is important to take into account changes in the cost of energy over time. Thermal modernization, in turn, allows to increase the level of thermal comfort, which is not taken into account and evaluated in practice, although the relevant standards for comfort conditions and categories of buildings to ensure comfort have been introduced in Ukraine. The paper analyzes the change in the level of comfort after thermal modernization and determines the category of the building to ensure comfortable conditions, determines the change in the average radiation temperature of the fences, as one of the main factors of PMV change in these conditions. Based on the integrated cost method, the cash flows for the model with different options of fences and heat sources are estimated, the change in the cost of energy over time is taken into account.*

Keywords: *energy need, PMV, mean radiant temperature, comfort temperature, thermomodernization.*

Introduction

The issue of improving the energy efficiency of buildings is an important and complex task in public policy. Particular attention should be paid to improving the energy efficiency of public buildings, the level of thermal comfort, which is regulated by the relevant requirements [1] and is widely covered in recent studies [2,3]. Furthermore, the human factor is taken into account by determining the level of use and the degree of occupancy of the building. Which in turn creates the need to consider economic, environmental, energy and social factors in assessing and making decisions during building thermal modernization.

Purpose and research objectives

The purpose of the work is to analyze the impact of thermal modernization on the level of thermal comfort and economic assessment of changes the heat source in combination with fences. According to the set goals, the following tasks should be solved:

1. Calculation of cash flows for the model with factual data and with improved thermal protection and new heat sources.
2. Creation of a room dynamic model to determine the change in the mean radiant temperature and thermal comfort parameters.
3. Analysis of energy and comfort characteristics of the apartment.

Material and research results

Initial data. The study object is a specialized school №64 in Kyiv, built in 1973. The average heat transfer coefficient of the walls significantly exceeds the standard heat transfer coefficient U=0,3 W/m^{2}°C and is U=1,3 W/m^{2}°C. The total area of the walls is 2814m^{2}. The average heat transfer coefficient of the building windows exceeds the calculated standard heat transfer coefficient U=1,33 W/m^{2}°C and is U=2,05 W/m^{2}°C. The total area of the windows is 1251,53 m^{2}. The average heat transfer coefficient of the roof significantly exceeds the standard heat transfer coefficient U=0,2 W/m^{2}°C and is U=0,58 W/m^{2}°C. The total roof area is 2251 m^{2}. Modeling of changes in the level of thermal comfort before and after thermal modernization is possible on the basis of the room model, which is due to the determination of the average radiation temperature. The object of the study was a typical study room located on the second floor of the building in block B measuring 10×10 m with three window openings measuring 2.05×2.1 m; thermal resistance of the window R_v =0,4m^{2} C/W, K = glazing coefficient 0,3. Outer wall area Fz = 17.4 m^{2}, thermal resistance of the outer wall R_z=0.77 m^{2} C/W. This model was created in the software product sketchUp and all engineering systems were added there, namely the simulation was performed in the software product EnergyPlus. The metabolic rate was 70 W/m^2, because in this room the main activity is sitting work, which is typical for schools and the thermal resistance of typical clothing combinations, respectively, is 0.155 m^{2}K/W. The study used hourly climate data of a typical year of the IWEC international weather file for the Kiev city conditions.

Model description. The choice of heat source in combination with building envelope is carried out using the method of cash flow, which allows to take into account the change in the cost of energy and money over time. Function of integration cost allows you to take into account the change in the cost over time of energy *l*, the discounting with E and the efficiency of the heating system ε. The discount rate is chosen according to the interest rate on bank deposits. The calculation of the level of thermal comfort is based on the Fanger method, which is presented in the standard ISO 7730 [1] and is based on the equations of human heat balance.

Analysis of the study results. As a result of complex thermal modernization of the building, the difference in % of the calculated value of specific energy consumption, from the maximum allowable value, is 20%. The energy efficiency class after thermal modernization, established by the ratio corresponds to the class - "B". Increasing the thermal resistance of the walls will be considered in conjunction with a change in the source of heat supply. At the moment, the appropriate microclimate in the room is created through the use of a centralized heating system. As an alternative, autonomous gas heating, a system with a heat pump installation was chosen. An optimistic and average forecast of changes in energy costs will be considered. The results of calculations of cash flow for the system after thermal modernization are presented in Fig.1. Cash flow graphs as integrated discounted costs will allow to determine the complex payback period of the proposed alternatives in comparison with the existing option.

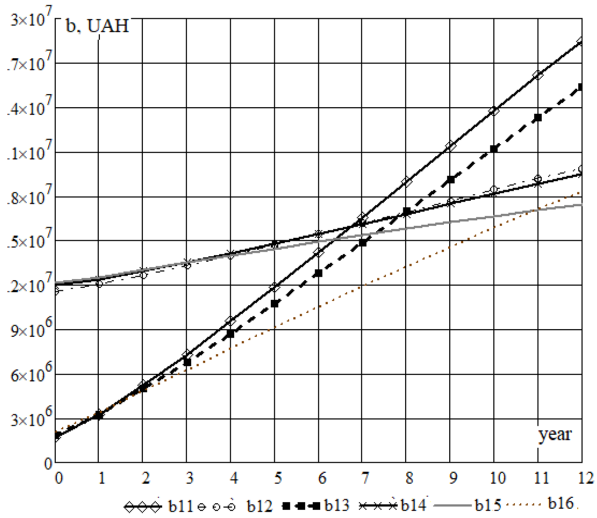

Figure 1. Cash flow for the system: b11, b12, b13, b14, b15, b16 - respectively, integrated discounted costs when using central heating before and after thermomodernization, gas boiler before and after thermal modernization and heat pump after and before thermal modernization, taking into account the change at the time of the cost of energy, UAH

Figure 1 shows the effect of accounting changes in the cost of energy. Significant changes in the discounted payback period indicate that the change in the cost of energy over time plays not to be mentioned and it should be taken into account. As a conclusion we can say that would be appropriate to change the source of heating, which in turn will lead to greater savings and faster payback measures. The calculation of thermal comfort indicators for variable mean radiant temperature during the year hourly before and after thermal modernization for North and South walls was carried out in Mathcad. The PMV values for the heating period are shown in Figure 2.

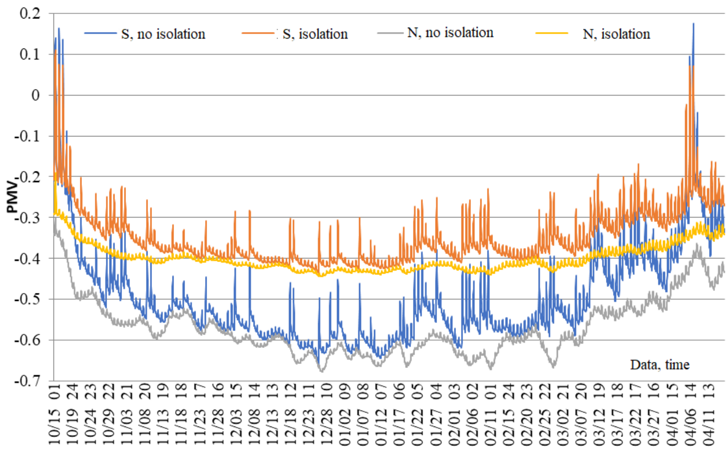

Figure 2. PMV value for the heating period

It is shown that PMV varies from -0.7 in the cold months to 0.2 in the offseason. Changing the thermal resistance of the barriers can increase the PMV, and thus improve human heat by about 0.1. South wall orientation characterized by large fluctuations PMV, due to incoming solar radiation and consequently the growth of the mean radiant temperature.

Conclusion

The paper presents a method of choosing a heat source on the example of the school, taking into account different scenarios of changes in energy costs over time. PMV has been found to vary from -0.7 in the cold months to 0.2 in the off-season. Changing the thermal resistance of the barriers can increase the PMV, and thus improve human heat by about 0.1. The wall of the South orientation is characterized by larger fluctuations of PMV, which is due to the inflow of solar radiation and as a consequence of the increase in cardiac radiation temperature of the room. Based on the calculation of PMV is found that this building has the third category of comfort - it is permissible average expectations can be used for existing buildings, but for a comfortable stay, especially for children's education is not enough. Therefore, in this paper it was proposed to conduct a comprehensive thermal modernization of the building, due to which the category of the building changed to the second category of comfort - this is a normal level of expectations to be met for new buildings and renovations.

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DISTRIBUTION OF HEATING ENERGY CONSUMPTION FOR APARTMENT BUILDING OF MASS CONSTRUCTION HEATING

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Introduction

The countries of Eastern and Central Europe are characterized by mass typical buildings of the 80s of the twentieth century, which are characterized by central regulation and accounting of energy consumption for heating, vertical single-pipe heating systems, high heating costs caused primarily by low thermal resistance, which does not meet modern requirements more than 3 times. Therefore, the current state of apartment buildings built during mass construction, require full or partial modernization and a significant increase in energy efficiency. In addition to energy saving measures, which are currently being actively implemented for the house as a whole and by individual apartment owners, there are a large number of operational problems [1, 2]. They are associated with changes in temperature schedules of heating systems, local regulation of heating conditions and so on. This, in turn, leads to different air temperatures in the areas of the house / apartments and heat flows in the middle of the building. In terms of increasing attention to energy efficiency and providing the necessary living conditions, it is important to study the impact of heat flows between adjacent areas (apartments), and how the regulation of heating by individual residents affects the comfort in neighboring apartments.

Purpose and research objectives

The aim of the work is to analyze the consumption of thermal energy for heating the block of premises on the riser of a five-storey residential building of mass construction, taking into account local regulations.

According to the set goals, the following tasks should be solved:

1. creation of a mathematical model to determine the thermal state of the premises on the riser;
2. design calculation for this building;
3. determining the impact and analysis of heat flows between apartments in the local regulation of the heating system by residents.

Material and research results

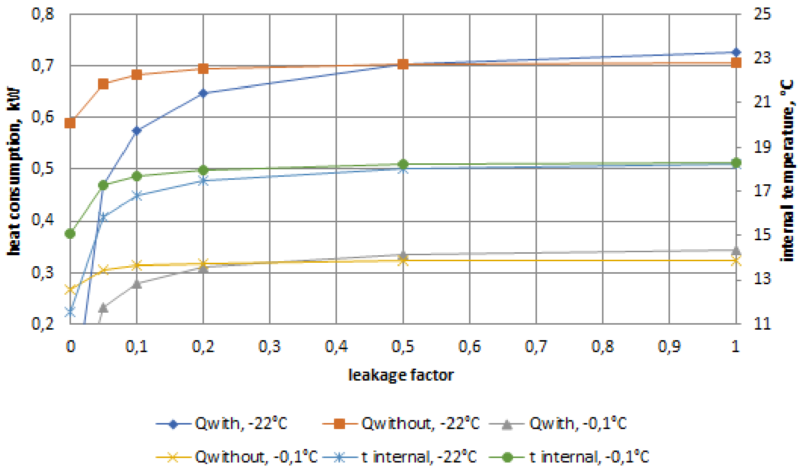
Initial data. Since most apartment buildings were built in the 60s and 80s, they mostly use a single-pipe vertical heating system with a flow-through connection of heating appliances (radiators type MS-140) with a locking area and the possibility of local regulation by bypass. The object of the study is a typical five-storey building without complex architectural and structural or structural solutions, which is located in Kiev, with the upper wiring of the heating system. The temperature schedule of the building heating system is "95/70" (for design conditions). According to the project, the leakage factor for heaters is 0.5. The load-bearing external walls are made of red hollow brick with a thermal resistance of the fence of 0.8 m^2K / W. The house has a flat roof and a floor located on the ground without recesses. The power of heating appliances, cast iron radiators, selected to provide an indoor temperature of 18°C under the calculated conditions of outdoor temperature and temperature schedule of heating.

Model description. For simulation, a model of a block of five identical rooms of different floors, which are heated, has been created. Calculation of thermophysical properties of barriers for model adjustment was performed according to the standard DSTU B A.2.2-12: 2015 [3]. In the software environment of the Mathcad model, a design calculation was performed for the design conditions of Kyiv [4], in the absence of solar heat and at an outdoor temperature of -22°C.

In addition, a series of simulations for the average conditions for the heating season - outdoor temperature -0.1°C. The influence of heating system control for example on the 4th floor with the help of regulators on heating devices for different external conditions, supply temperatures and change of heat carrier flow through the heating device was investigated.

Analysis of the study results. The mathematical model considers a riser of a single-pipe heating system with a closing section, which passes through the same living quarters of a five-storey building. Based on the equations of heat flow balances for rooms and heaters, a system of 48 equations is solved. The equation of heat balance of premises takes into account the heat flow from heating appliances, heat losses through external, internal fences, floors, floor and leakage coefficient α. Solar heat was not taken into account to distinguish the impact of each of the components and the possibility of comparing them under the calculated and average external conditions. The temperature of the coolant at the inlet and outlet of the heater, heat flow from the heater, indoor air temperature, as well as the amount of flow through internal enclosures and ceilings depending on local regulation on the heater 4th floor flow coefficients by bypass were studied.

In fig. 1 shows the dependence of the change in heat flux from the heater on the 4th floor depending on the leakage coefficient α for the calculated temperature of -22°C and the average temperature of the heating season -0.1°C, taking into account internal flows into adjacent rooms and without.



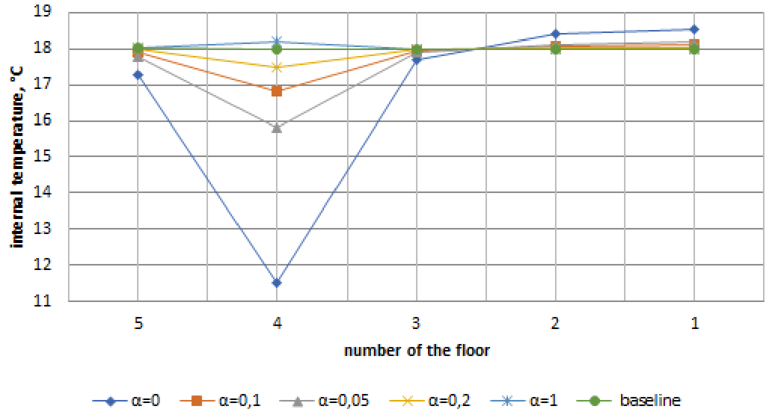
Qwith, -22°C; Qwith, -0,1°C – with flow (outside air temperature -22°C and -0,1°C respectively), kW; Qwithout, -22°C; Qwithout, -0,1°C – without overflows (outside air temperature -22°C and -0,1°C respectively), kW.

Figure 1. The magnitude of the heat flux from the heater of the 4th floor and the room temperature at different leakage factor α

From fig. 1 it follows that the dependence of heat flux on the leakage coefficient α has a typical nonlinear dependence, which is explained by the exponent of 1.3 for cast iron radiators in the equation of heat flux from heaters. The heat flux is more sensitive to small leakage factor (α = 0…0.2), with further increase α the heat flux does not change significantly. For average outdoor conditions, the water temperature in the supply pipe at the entrance to the heating device of the 5th floor is 58°C, respectively, for the temperature graph "95/70".

In fig. 2 shows the dependence of the change in air temperature for different floors under different leakage factor on the 4th floor of a 5-storey building with the upper dilution of heating devices of the heating system. Complete shutdown of the heater on the 4th floor leads to a decrease in temperature on this floor to 11.5°C, this level is maintained by the flow of heat from adjacent rooms through the interior walls, floor and ceiling. This in turn leads to a decrease in temperature in the absence of heating control in rooms 5 and 3 floors due to transmission flows to the 4th floor and increase the temperature on the lower floors to 0.5°C by increasing the temperature of the coolant at the entrance to heating appliances.

At the leakage factor of 0.05… 0.2, the internal air temperature on the 4th floor fluctuates within 16… 17.5°C, and on other floors no noticeable change in air temperature is observed. Further increase in the flow coefficient of 0.5...1 does not give a significant effect, because it leads to a change in air temperature on this floor to 0.1...0.2°C. The estimation of heat flows between the next premises in the conditions of similar regulation is carried out.



** the given coefficients α are characteristic for 4 floors, on other floors correspond to design α= 0,5*

Figure 2. Floor-by-floor distribution of internal temperature in the premises on the riser

Conclusion. The paper analyzes the local regulation of a single-pipe heating system with closing links of a 5-storey typical mass building of the 80s on the basis of a simulation model created in the Mathcad software environment for design/calculation and average conditions. The internal heat flows to the adjacent rooms, the change of the internal air temperature under the influence of the change of the flow coefficients in the heating device of the 4th floor are investigated.

In further research it is planned to conduct a similar calculation for buildings that meet the modern thermal properties of fences, as well as to investigate the impact of operational and behavioral characteristics of residents on energy consumption, taking into account indoor and solar heat in the room.

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COMPARATIVE ESTIMATES THE EFFICIENCY INDICATORS TIME SERIES OF CHP BASED ON THE THERMODYNAMIC APPROACH TO SHARING FUEL COSTS

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Introduction. The CHP provides in Ukraine about 30% of the heat demand and 10% of the total electricity production, having a significant impact on the energy, economic and environmental performance both the power system and the district heating systems. The main capacities of the combined production are concentrated at large urban and industrial CHP plants equipped with steam turbine units with an electric power of 4 to 250 MW and initial steam pressure of 3.43 to 23.5 MPa. A relatively small (up to 2%), however, a stable contribution to the total production of heat, mainly for the needs of their satellite cities, is provided by the extraction power units of large thermal and nuclear power plants. Since CHP provide certain fuel savings compared to separate production of power and heat and then reduce greenhouse gas emissions, until recently they were relieved of competition in the electricity market and operated at regulated tariffs that fully cover the costs with a normalized profit. In addition, over the past eight years, CHPPs have been able to independently distribute the costs of generating electricity and heat in order to increase profitability. Since up to 80% of costs are fuel costs, their manipulation is achieved in practice by changing the efficiency indicators of plants for the production each types of useful energy, indicated in the state statistical reporting.

As a result, efficiency indicators ceased to play their main role in reflecting the actual thermal efficiency of the CHP operation. In addition, arbitrary changes in efficiency indicators significantly complicate the comparative analysis of the technical perfection of different thermal power plants, as well as changes in the efficiency of each plant over time. In this regard, the corresponding statistical time series of performance indicators, defined by state statistics, is gradually losing its role as a source of objective information for the formation of targeted technical policy in the industry.

The purpose of this study was to develop and test a methodology for a comparative analysis of production efficiency at various plants, as well as reproduction evolution of efficiency indicators over time using retrospective statistical information about input /output energy fluxes by each plant over the years.

For cogeneration units, the experimental determination the efficiency of obtaining each product during its operation is complicated due to the uncertainty the division of primary energy costs spent in the installation between the types of useful energy. In this regard, evaluations the efficiency of such installations for the production of each useful energy flux are performed in practice by calculation methods.

The allocation fuel cost between heat and power outputs at TPP is carried out in Ukraine according to sectoral methods governing the procedure for determining the main indicators of thermal efficiency of the power plant - specific consumption of conventional fuel for electricity and for heat, accordingly.

In the period from the middle of the last century to the present day, at least three standard methods of assessing the efficiency of combined heat and power plants have been in force in Ukraine. Since 1954, the so-called "physical" method of cost sharing has been applied in Ukraine, the effect of which was extended by the industry standard of 1996 [1]. This document attributes fuel savings due to cogeneration, mainly to electricity. In 1998, it was amended to divide the savings between the two products, but with a predominance of electricity [2]. Finally, in 2003, the standard [3] came into force, according to which plants were given the right to set the desired efficiency values for a certain period of time with a restriction on the maximum efficiency of heat production. The Commission for the regulation of energy and communal services of Ukraine using [3] when regulating tariffs for heat from the CHPP, at that time the price for electrical energy is a competitive market of electrical energy. All of the above standards assume the conditional nature of performance indicators and the non-obligatory compliance with thermodynamic laws. Their calculation formulas are complex and use many technological parameters, which are not possible to reproduce.

The possibility of reproducing evolutionary changes in the efficiency of CHP is provided by the state standard of Ukraine [4], which regulates the procedure for determining the actual, physically reliable values of specific fuel consumption for heat and electricity by steam turbines of combined production. The theoretical foundations of the method used in the standards are discussed in detail in [5].

This paper presents the results of calculating the evolutionary series of efficiency indicators CHPP and extraction TPP in the period 2001-2019 according to [4] in comparison with the reported efficiency indicators.

The allocation total fuel energy consumption at the CHPP between its outputs defined as [4]:

$$B_E = B\frac{E}{E+\omega H}\,;\; B_H = B\frac{\omega H}{E+\omega H}\,;\, \omega = \frac{\eta_e^o - \eta_e}{1-\eta_e} \tag{1}$$

where E, H – power and heat net output, B – total fuel energy input, B_E, B_H – fuel consumption for the power and heat generation, ω - thermodynamic value of heat, $\eta_e = \frac{E}{B}$ – the electric efficiency, η_e^o- the thermal efficiency of the ideal condensing cycle with heat dissipation at ambient temperature.

The methodology [4] was supplemented by methodological provisions for the formation of initial data and assessment the uncertainty of determining the actual efficiency at existing power plants. The practical approbation of the methodology was carried out using the annual statistical accounting data of 7 city CHPPs and 3 coal-fired TPPs of generating companies. The objects under study differed in the types of fuel burned, the types of turbines and steam boilers, unit capacity, the initial steam parameters, the degree of load etc. The calculation of the thermodynamic heat value and objective efficiency indicators of the plants according to [4] and their comparison with the reporting indicators determined according to [3] were made.

Conclusions. It was found that the calculation results adequately reflect the changes in the operating conditions of each plant over time and make it possible to realistically compare different plants at similar points in time. The results of the study showed that the reporting indicators of considered plants after the introduction of the methodology [3] have undergone changes, not always reflecting the actual changes in operating conditions. In particular, TPPs of generating companies supplying heat only to their satellite cities have reduced the reported indicators of specific fuel consumption for heat by 35-40% to the real values determined [5], the rest left them at the initial level of 2003. The reported heat generation efficiency of CHPPs is subject to the condition of maintaining economic parity with local boiler houses and is usually below real values. The largest deviation is typical for CHP plants with high initial steam pressure. The results of the study give grounds to recommend an assessment of the existing and developed plants in accordance with [4] leaving for plants the right to use [3] for economic calculations only.

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NEURAL NETWORK MODEL FOR ENTERPRISE ENERGY CONSUMPTION FORECASTING

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Abstract. This research paper investigates the application of neural network models for forecasting in energy. The results of forecasting the weekly energy consumption of the enterprise according to the model of a multilayer perceptron at different values of neurons and training algorithms are given. The estimation and comparative analysis of models depending on model parameters is made.
Keywords: *electrical loads, daily schedule, modelling, neural network, multilayer perceptron, MLP.*

Introduction

In the presence of the electricity market, the need to obtain forecast values of energy consumption is due to economic and technological reasons. Improving the accuracy of the forecast of electric loads allows to avoid overloading of generating capacities, to improve the quality of electricity, and to minimize its losses. This points to a great relevance and importance of the tasks related to modelling and forecasting of electrical loads for planning the optimal modes of operation of electricity consumers and power supply systems.

The quality and accuracy of the forecast depends on the chosen mathematical model. There are a large number of models and methods for loads prediction, which are usually based on the retrospective dynamics of power consumption and the factors that affect it, to identify a statistical relationship between model parameters and process characteristics [1-2]. In recent decades, the mathematical tools of artificial neural networks (ANN) have been successfully used for prediction, models based on which can establish a relationship between the output characteristics of the system and input factors using the learning procedure. The use of artificial neural networks allows to achieve forecast accuracy up to 96-97%, which will have a significant impact on the management of electrical loads. The choice of network type and its configuration depends on the specific task, available data and their volume [3].

Purpose and research objectives

The purpose of the work is to develop a model for forecasting the electricity consumption of an enterprise using artificial neural networks to improve the accuracy of planning the operating mode and increase the reliability of the enterprise's estimates in making technical and economic decisions.

To achieve this goal, the following objectives are addressed in the paper:

Building a structure and developing a mathematical model of ANN for power consumption forecasting.

Investigation of neural models with different numbers of neurons to assess the effect of ANN configuration on prediction accuracy.

Object of research: graphs of electrical loads of the enterprise, neuron model of the process of power consumption at the enterprise.

Subject of research: characteristics of the neuron model, indicators of electricity consumption and factors that determine the enterprise's mode of operation and electricity consumption.

Material and research results.

Interconnected neurons form a neural network. Network configuration is determined for each separate task. To solve some individual types of problems, there are already optimal configurations described in the academic literature on the construction and operation of neural networks [4-5].

The paper predicts the schedule of active power consumption for the day ahead on the basis of data on electricity consumption for the previous days. The total sampling consists of 168 observations (24 hourly observations per day during a week) and is provided in the form of a table and graph (Fig. 1). To verify the accuracy of the forecast, the forecast will be based on 144 observations, and the data of the last day will serve as a control sequence. The accuracy of the model will be assessed by the average value of the relative error in the control sequence and the value of the relative error in determining the daily power consumption.

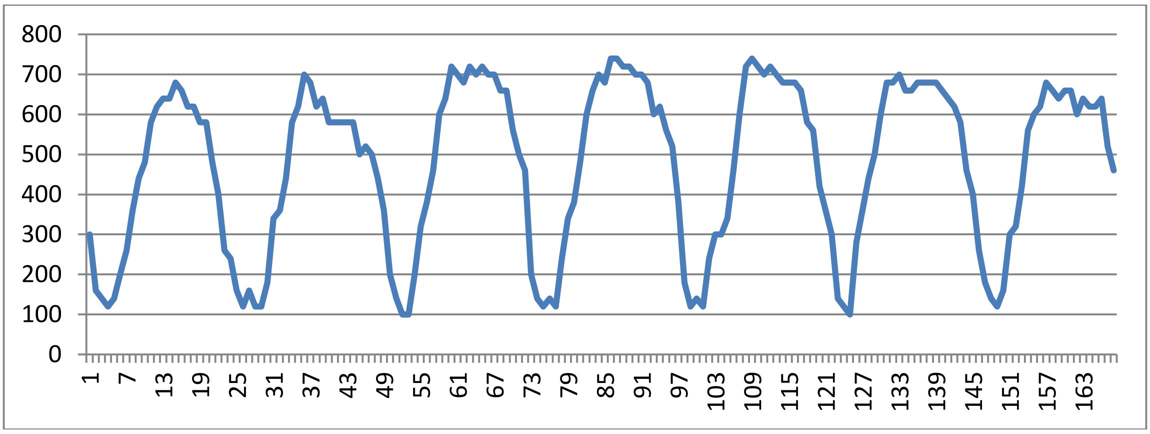

Figure 1 Weekly schedule of the enterprise's energy consumption

A multilayer perceptron was adopted as a model for prediction. The number of perceptron inputs is determined by the length of the load schedule (24 observations per day). To obtain the predicted value, one source element is sufficient.

The two networks with the best performance (the smallest absolute error in the training and control sequence) were used to build forecasts for the day ahead (Fig. 2)

The quality of the models was assessed by relative error indicators for the predicted values of electricity consumption and the total amount of electricity consumed. The average error in forecasting the current values of active power consumption for MLP 24-16-1, MLP-24-14-1 and MLP 24-1-1 networks was 9.7%, 9.9% and 9.5%, respectively. When estimating the total amount of energy consumed, the error was 1.8%, 1.6% and 4.1%.

Table 1 Electrical grids characteristics.

№	Net	Training perf.	Test perf.	Training error	Test error	Training algorithm	Hidden activation	Output activation
2	MLP 24-14-1	0,974301	0,983437	804,850	646,3876	BFGS 12	Logistic	Identity
9	MLP 24-16-1	0,975561	0,982711	752,883	740,1567	BFGS 11	Identity	Logistic

Conclusions. To build a mathematical model of the function of changing the electrical load, it is advisable to use a multilayer perceptron.

The daily load schedule with hourly data recording determines the observation period and the required number of neurons (24) in the input layer, because predicting a decrease in the number of neurons will worsen the quality of the model due to period mismatch, and increase will complicate the model.

The error of the forecast values for operation at rated mode from 10 to 17 hours (minimum value 0.4%, maximum value 5.8%) and 4.1% error of forecast daily consumption indicates sufficient accuracy of the ANN model applied for forecasting daily loads of the enterprise.

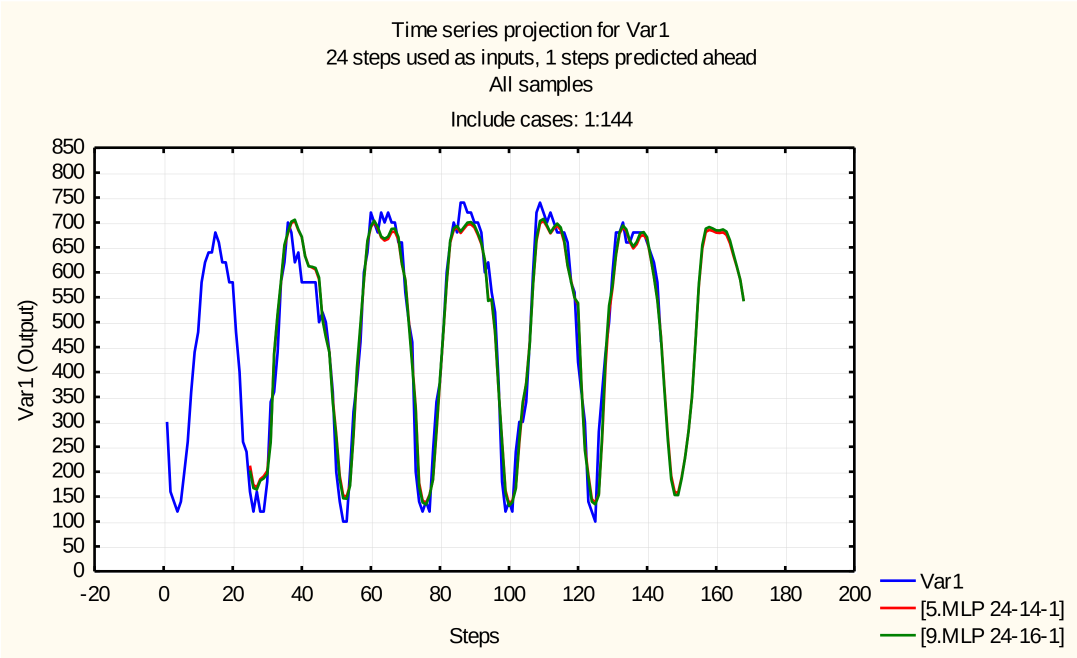

Figure 2 Energy consumption graphs by the original sequence and ANN models.

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NEURAL NETWORK MODEL OF THE MECHATRON COMPLEX "CRUSHER MILL"

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Abstract. The paper discusses the use of the technology of artificial neural networks to improve technical and economic performance of crushing and milling complex. Formulated the goal and major tasks of constructing a system of automated control and monitoring to optimize the power consumption of crushing and milling complex is analyzed modes of complex mechatronic and development of multicriteria models to provide optimal technological parameters of the equipment.
Keywords: *crushing and grinding complex, modelling, neural network,*

Introduction

Effective management of crushing and grinding is possible with the most complete mathematical model of the situation. Work of crushing and grinding complex is determined by dozens of factors, many of which are random. The increase in the number of considered factors complicates the model, so it is necessary to form the control action, finding a compromise solution that takes into account the degree of informativeness of the factor field and its complexity. It is necessary to solve a number of tasks such as: analysis of operation modes of the equipment of grinding to determine the parameters, which correspond to different operation modes; development of a multicriterial model of optimal technological parameters of equipment of the crushing and grinding complex subject of power consumption, productivity and quality of grinding; development of the model of crushing and grinding complex, which could provide optimal power consumption management of crushing and grinding complex.

The simulation of crushing and grinding complex is used to optimize its energy consumption. This particular task contains a number of subtasks such as: description of kinematics and dynamics of internal processes, the establishment of energy intensive of processes of destruction of rocks, the construction of control systems for crushing and grinding complex. The analysis of works in the field of simulation of the grinding aggregates and management highlights the following researchers [1-2].

The most famous equation that establishes a relationship between the energy expended and the fineness of the product resulting from grinding are equations of Rittinger, Kick-Kirpichev, Bond. Based on the works of Andreev, Davis, Perov, Tovarov, Olevskii, Kantorovich established that the greatest power consumption characterizes, however, and most work of grinding.

Purpose and research objectives

The purpose of this work is to develop a model of a crushing and grinding complex. To achieve this goal, the following tasks are solved in the work: analysis of factors and parameters that determine the processes of crushing and grinding in grinding units, selection of the structure and parameters of a neural network corresponding to a multifactorial task, determination and description of the factors most affecting power consumption.

Object of research: neuron model of the crushing and grinding complex.

Subject of research: power consumption of the crushing and grinding complex.

Material and research results.

Let the crushing complex consist of several grinding aggregates, included in the sequential work (Fig. 1). Each unit that is part of the complex is characterized by a certain amount of power consumption or power consumption per ton of ground material, the value of which depends on a number of factors (we take into consideration such factors as the mass of grinding bodies M, the productivity of the aggregate Q and the size of the raw material T) and size of the finished product. The size of the finished product is determined by the mode of operation of the unit and the grinding time, which affects the amount of power consumed.)

The ratio of the size of the product at the inlet to the size of the product at the outlet determines the degree of crushing of the product i. For N consecutive objects, the total degree of grinding is determined by multiplying the degrees of grinding at each stage [3].

$$i = \prod_{K=1}^{N} i_K \tag{1}$$

The value of the power consumption for the complex consists of the sum of the power consumption of individual units

$$W = \sum_{K=1}^{N} w_K \tag{2}$$

Then, the optimization problem for a crushing complex with a performance varying in a narrow range will be written in the form

$$W_{sp}(Q, i, k_N) \to \min$$
$$\begin{cases} Q = const \in [Q_{\min}, Q_{\max}] \\ i \geq i_{\partial on} \end{cases} \tag{3}$$

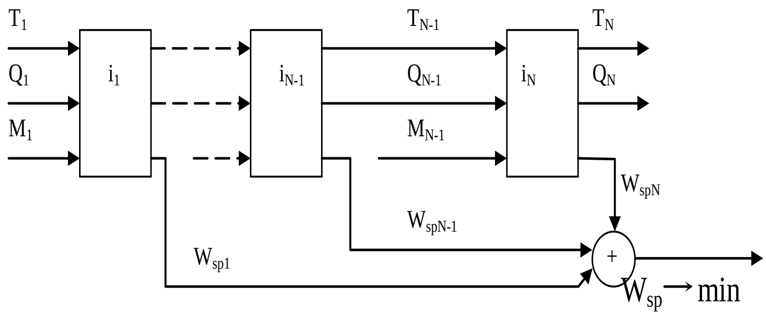

Fig. 1 Model of crushing and grinding complex.

System control using neural networks provide an alternative to the control systems, constructed according to the classical methods of management. This possibility is based on the fact that a neural network consisting of two layers and containing in the hidden layer is arbitrarily large number of nodes can approximate any function of real numbers with a given degree of accuracy [4].

To ensure monitoring system with the prediction function, it is necessary to build a neural network model of the form (4) [5].

$$y(k+d) = N\begin{bmatrix} y(k), y(k-1), ..., y(k-n+1), u(k), \\ u(k-1), ..., u(k-n+1) \end{bmatrix} \tag{4}$$

where *y (k)* is the output of the model; *d* is the number of prediction cycles; *u (k)* is the output of the model.

The result of the operation of the system with a trained controller is shown in Fig. 2 where curve 1 shows the input stimulus; and curve 2 is the output signal.

Comparison charts of the input (random) signal and output of the system shows that the use of the controller allows to achieve a more stable work area for the output product, in case of random changes of the input traffic.

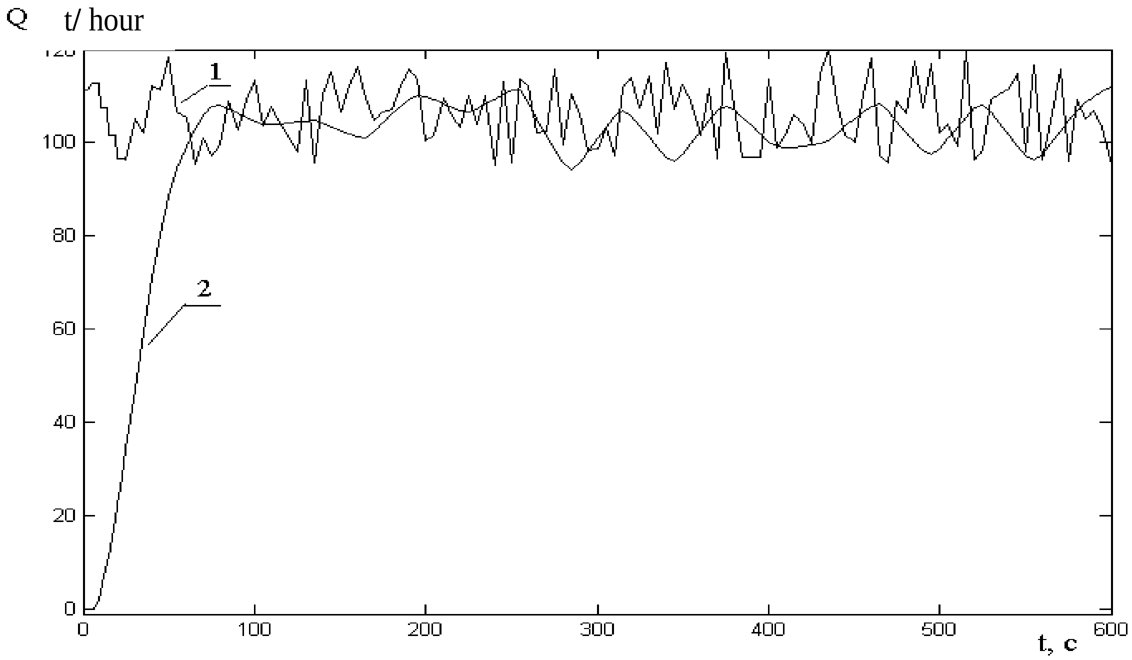

Fig. 2 Learning outcomes of the neuromodel.

Conclusions. The proposed new model of the crushing and grinding complex, which takes into account multi-factor field of the system and displays its internal links based on the mathematical apparatus of artificial neural networks lies in accounting for the formation of the objective function components that determine energy consumption and other technical and economic indicators of the complex “crusher mill” that allows you to increase the energy efficiency of the system by ensuring operation at the optimum power consumption mode.

The developed model of crushing and grinding complex, which consists of several crushing units, which operate sequentially to determine the optimal parameters for a given criterion the operation mode of the complex, which reduces the power consumption of the complex by selecting an optimal mode for reducing the substance.

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INFLUENCE OF THE COOLANT FLOW RATE ON THE TEMPERATURE CONDITIONS OF THE HYBRID PV MODULE OPERATION

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Abstract. Analytical investigations of thermal regimes of a hybrid solar collector for cooling conditions that differ in the way of the charge of heat control are given. The work purposes the method of rational thermal regimes definition. Features of observed regimes are shown and generalising method of practical calculation defining temperature characteristics in different working conditions of the photomodule is given.

Keywords: PVT, hybrid photovoltaic thermal panel, temperature regimes

Introduction

It is known that efficiency of electric energy production of the photomodule depends on its temperature. To decrease the heating of device apply natural and a forced cooling. In the last case the organised remove of warmth gives the chance to its use [1,2,3] that raises the general power efficiency of the device. In a hybrid solar collecting channel (PVT) photocells are chilled by an active cooling system of by heat-transfer fluid through channels in a back part of the module.

Brief analysis of recent publications

Producers of photomodules restrict temperature level of maintenance of photomodules in limits 50…55°C. However for PVT where the heat-transfer agent reheat temperature predetermines a direction of use of power resources, the thermal operating mode of the device should be proved. At sampling of temperature operating modes PVT, as a rule, use analytical models [1,3,4]. Boundedness of the mathematical description on a way and detail of work of installations in interfaced and essentially variable conditions thus occurs.

Goal

To develop the integrated mathematical model for definition of temperature operating modes of a hybrid solar collecting channel.

Method of investigation

For the analysis it is accepted the modelling structure of the device consisting of an absorber of solar energy which on the one hand is fenced from an outer space by a pellucid wall, with another is an element of the flat channel for chilling heat-transfer agent.

The heat transport in PVT is defined by external and internal conditions of a leakage of process. External conditions are intensity of irrradiation PVT and heat exchange with environment. Internal conditions are formed at heat exchange between an absorber and heat-transfer agent, and also between heat-transfer agent and the back wall bounding on environment. At the description of these processes the system of the equations of conservation of energy in the interfaced aspect is made. The equation of conservation of energy in an absorber is presented in a nonlinear aspect where efficiency of generation of the electric power of the photobattery is temperature function; in heat-transfer agent - in the differential form concerning axial coordinate of a stream in the channel that gives the chance to raise accuracy of the analysis and also to raise informtiveness researches.

The system of the equations added with boundary conditions, characteristic for maintenance of solar devices, dared a numerical method. As change of temperatures in system occurs along a heat-transfer agent current, rationing of observed parametres concerning a channel width is made. Thus, for the absorber square in characteristic functional parametre the length of the channel is.

By consideration of the combined power device by analysis key parametres heat processes the final temperature of heat-transfer agent and absorber temperature are. They essentially change along an absorber surface. Two refrigerating duties PVT are observed: first - at independent of sizes of an absorber the charge of heat-transfer agent, $g \neq f(A_{ab})$; second - at change of the charge of heat-transfer agent depending on the square of an absorber, $g = f(A_{ab})$. Here A_{ab} - the absorber square.

Regime $g \neq f(A_{ab})$ means that the heat-transfer agent charge is set for one, base, size PVT and with size change its value remains to constants. In the given work in the capacity of the base it is accepted typical for solar collecting channel the liquid charge g = 0,015 l/s per 1 m^2 the absorber squares. Thus, alternative sizes of the channel, proceeding from the accepted conditions on fixing of settlement width (1 m), will be defined in its length.

As absorber and heat-transfer agent temperatures are interconnected, for generalisation of the solution of system of the equations the complex defining a relationship of change of temperature of heat-transfer agent and average temperature of an absorber is injected. Such complex we name effectiveness ratio of a heat transport from an absorber to chilling heat-transfer agent and we express in an aspect

$$\eta_{he} = \frac{\delta t_f}{\bar{t}_{ab}}.$$

Here $\delta t_f = t''_f - t'_f$; t'_f and t''_f - initial and final temperatures of heat-transfer agent; $\bar{t}_{ab}$ - average temperature of an absorber.

For PVT a concrete design the heat transport effectiveness ratio depends on external conditions which are formed by external temperature and intensity of insolation. As shows the analysis, with increase in these parametres η_{he} increases. The additional factor of such dependence is the length of the channel. The increase in length raises agency of these parametres on function. Thus, value η_{he} cannot be to the accepted constants and at regime calculations it is necessary to consider agency of conditions of conducting of process.

Taking into account defining agency on heat transport effectiveness ratio at the constant charge of a liquid of intensity of intensity of irrradiation, temperatures of outdoor air and absorber sizes generalising dependence in an aspect is gained

$$\eta_{he} = 0,1\left[t_a\left(\frac{A^*}{H}\right)^{0,48} - 1\right] + (0,424A^* - 0,0253)\exp\left(-\frac{233A^* + 141}{H}\right).$$

Here H - intensity of irrradiation; t_a - ambient temperature; A^* - the square of an absorber led to width of an absorber.

As characteristic temperatures - components of effectiveness ratio of a heat transport are interconnected and defined by concrete conditions for a finding of one of them it is necessary to have a well-founded method of calculation another. Taking into account it, on the basis of resulted above model dependence for average temperature of an absorber has been gained.

In a regime $g = f(A_{ab})$ the liquid charge variable, proportionally depending on the absorber square.

As the analysis, under such circumstances absorber temperature shows does not depend on its sizes. Dependence parametre between temperature of an absorber and external temperature is magnitude of irradiance. Its growth leads to increase in temperature of an absorber. This dependence for all sizes of the channel matches to conditions of base alternative with the square 1м^2 (length, equal 1 m).

The effectiveness ratio submits to the dependences, gained for base alternative, and its magnitude does not depend on absorber sizes. Thus, at any sizes PVT of the deepest cooling it is possible to achieve in a regime $g = f(A_{ab})$.

Proceeding from the gained data, the greatest agency on heat transport effectiveness ratio under constant regime conditions is rendered by intensity of intensity of irrradiation and outdoor air temperature. Sizes do not render essential agency. Taking into account it for a refrigerating duty $g = f(A_{ab})$ generalising dependence in an aspect is gained

$$\eta_{he} = 0{,}143 \cdot \ln(H)\text{-}0{,}797\text{+}t_a(119 + 0{,}194H)^{-1}.$$

Conclusion

Developed equation can be used for definition of an absorber temperature and final temperature of heat-transfer agent of the combined solar device in problems of regime optimisation. This formula occurs at the set reference temperature t'_f equal 20°C .

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INCREASING THE EFFICIENCY OF A HYBRID PHOTOELECTRIC SYSTEM OF A LOCAL OBJECT WITH A STORAGE BATTERY BASED ON FORECAST

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Introduction. The disadvantage of photoelectric systems (PES) is the uneven generation of electrical energy. The largest generation of energy from a photoelectric battery (PV) occurs at noon hours. This creates the problem of ensuring a balance between consumption and generation in the power system as a whole, as well as at the level of local microgrids with distributed generation. The energy consumption of most local objects (LO) is a general nature with peak morning and evening loads. Excess energy for generation into the grid appears only during the hours of maximum solar activity. An urgent task is the use of energy storage devices, both at the level of the power system and local grids with PES. In this case, LO with PES becomes a participant in the energy market with a regulatory function. It becomes possible to reduce daily consumption from the distribution grid (DG) and carry out the planned generation of energy in the DG only during peak hours, which will contribute to the balance in the power system. At the same time, additional investments in a storage device (storage battery (SB)) should pay off by reducing the cost of paying for the electricity consumed by the LO from the grid. This is possible when improving the energy management of PES using the forecast of the PV generation. The ability to forecast the PV generation is now provided by open web resources [1, 2].

The aim of the work. Improvement the principles of control of PES with a storage battery to meet the own needs of local object with the possibility of planned power generation to the grid during peak hours using the forecast.

Results of research. The structure of PES with a SB with a multifunctional grid inverter, made according to the principles presented in [3], is considered. In this case, the regulation of the PV generation P_{PV} is provided.

The average value load power P_{LAV} and power P_g generation into the DG in the evening peak are determined by permissible degree of discharge $\Delta Q^*=Q^*_{SE}$ - Q^*_{EE} ($Q^*=100Q/Q_R$, Q_R – rated charge (C_B – capacity, Ah), Q^*_{SE} and Q^*_{EE} – start and end values). Energy transferred by the SB to the load is defined as $W_{BL}=0.01(Q^*_{SE} - Q^*_{EE})W_{BRL}$ ($W_{BRL}=W_{BR}\cdot\eta_C\cdot\eta_B$, $W_{BR}=U_BQ_R$ – SB energy capacity, U_B – voltage, η_C and η_B – respectively, efficiency of converter unit and SB). The possibilities of generation are increasing in the morning peak because the PV energy is added (W_{PVPm} – according to the forecast data). The power value in this case

$$P_{Lpm} = \frac{0.01(Q^*_{Sm} - Q^*_{Em})W_{BRL} + 0.9W_{PVPm}\eta_C}{3}. \quad (1)$$

Safety factor 0.9 is introduced in (1) for W_{PVPm} taking into account the possible discrepancy between the forecast P_{PVP} and the actual value P_{PVF}. Power generation into grid is $P_{gm}=P_{LPm}-P_{LP}$. An important issue is to ensure the normal functioning of the PES in the autonomous mode (AM), which is possible when the DG voltage is disconnected or when the voltage deviation exceeds the permissible value (± 10%, better ± (5 ÷ 7)%). It should be taken in account that the duration of the AM can be quite long.

The solution is possible in the presence of the recommended load schedule $P_{LREC}(t)$, which sets the maximum value of the load power during the operation of the PES. It will allow planning the LO load for this period with its current correction. At the same time, at the moment of switching t_A, $P_{LRECP}(t)$ is formed with a discreteness of 0.5 hour, then the value P_{LRECi} is issued at each subsequent interval.

The PES operation cycle is realized by the simulation model for the research of energy processes in the system without taking into account transient processes. Energy losses are taken into account through the efficiency. To assess the efficiency, the coefficient k_E=C_1/C_2 is used (C_1 is the cost of electricity consumed by the LO load at one tariff rate, C_2 is the cost of electricity consumed from the DG under the adopted tariff plan). Respectively,

$$k_E = \frac{W_{Ld}T_d + W_{Ln}T_n}{W_{gds}T_d + W_{gn}T_n - W_{gdg}T_p},$$

where W_{Ld} and W_{Ln} – the energy, consumed by LO in daytime and night, respectively; T_d and T_n – the daytime and night tariffs, respectively: T_d=T_n =1 – in the numerator, the actual values of the tariffs – in the denominator; W_{gds} and W_{gn} – the energy, consumed from DG (at P_{gp}<0) in daytime and night, respectively; W_{gdg} – energy generated into DG (at P_{gp}>0).

Parameters of the simulation model: PV power P_{PVR}=1 kW, SB with W_B=3 kWh, values of efficiency of SB and converter unit – η_B=η_C=0.94. The load $P_L(t)$ does not depend on the season of the year. The oscillograms of the daily cycle of the PES operation for a cloudy day on July 12, 2015 (W_{PV}=2.83 kWh) is shown in Fig. 1. At the same time, at the hours of the morning peak – P_{gm}/P_{LAV}=0.43, and at the evening peak – P_{ge}/P_{LAV}=0.3. On a clear day in July, P_{gm}/P_{LAV}=1.27. The k_E values for different tariff plans (T_n/T_d/T_p) are given in Table 1 (the "+" symbol means that DG pays). At W_{PV}=1.32 kWh P_{gm} = 0, which led to a decrease in k_E. Significant energy consumption occurs when SB charging at night, then presence of the night tariff has a decisive influence.

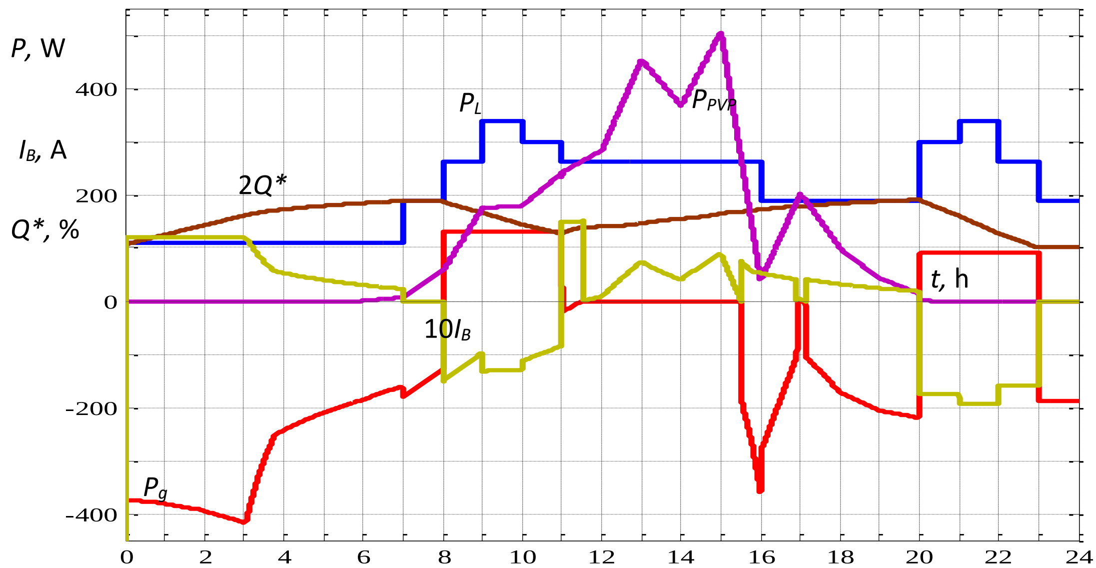

Fig.1. Oscillograms of the daily cycle of the PES operation at W_{PV}=2.83 kWh

Table 1 – The k_E values for different tariff plans $T_n/T_d/T_p$

W_{PV}, kWh	k_E			
	1/1/1	0.5/1/1	0.4/1/1.5	0.4/1/2
6	5.69	220	+	+
4.756	3.82	13.62	+	+
2.83	1.93	3.22	4.92	7.28
1.32	1.45	2.08	2.51	2.81

AM studies, when switching off the DG in the daytime, were carried out using the recommended load schedule based on the forecast $P_{PV}(t)$ with the current correction and the function of automatic regulation of the load voltage. The results confirm the possibility of increasing the degree of SB charge by the dark time of the day with an increase in the night load by 14%.

Conclusions. The possibility of meeting of own needs of the LO with the planned for the day ahead power generation to the grid during peak hours is shown. The use of the PV generation forecast for the formation of the recommended load schedule in the AM of operation with the correction of load deviations and forecast data allows to provide a SB charge for the possibility of maintaining the LO operation until the next day. Simulation modeling in a daily cycle showed that with the adopted PES parameters, a decrease in the cost of paying for electricity from the grid at one tariff rate by two or more times is possible when PV is generated over 0.5 of the maximum value on a clear day. The maximum value of the generation power at the morning peak is 1.27 times higher than the load power. If it is impossible to generate during the morning peak, the efficiency drops. The maximum efficiency is achieved with multi-zone pricing. The proposed principles of intelligent control allow increasing the efficiency of the implementation of the AM operation when the DG is disconnected.

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DEVICE FOR INTENSIFYING THE PROCESS OF BIOGAS FORMATION IN THE REACTOR

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Abstract: It is known that the disadvantage of most existing devices for intensifying the release of biogas is that as a stimulating effect on microorganisms that are in the substrate and participate in biomethanogenesis, use mixing or heating of the substance. These ways to increase the efficiency of a biogas plant have actually exhausted their potential. The proposed utility model of the device belongs to the field of processing of organ-containing waste, in particular to the means for intensifying the release of biogas from them.

Keywords: biogas, biomethanogenesis, bioenergy, biogas plant, bioreactor

Introduction. The closest to the set of features to the device under development is a stirrer, which contains installed with the ability to rotate around its axis a horizontal disk with fixed vertical rods [1]. The disadvantage of this device is its low efficiency.

To solve this problem in the installation to intensify the process of gas formation in the bioreactor on a vertical shaft installed a device containing two horizontal nozzles in the form of lattice wheels, between which are vertical rods with a metal core, externally covered with an insulating shell. Adjustable negative and positive potentials are brought to the rods through the upper lattice wheel in a checkerboard pattern.

Research tasks. The proposed device, which is shown in Figure 1, is located in the housing of the bioreactor 1, on the vertical axis of which is a shaft 2 passing through the nut 3 with a thread under the worm gear. Outside the housing, the shaft 2 rotates the drive 4 and is connected to the control unit of the electric field 5. On the shaft 2 above the biomass level in the methane tank rigidly fixed wheel-lattice 6, which is mounted rods 7 immersed in the substrate parallel to the vertical axis. The latter, as noted, have a metal (conductive) core and are covered on the outside with an insulating shell: plastic or other chemically stable non-conductive polymer. are connected by a non-conducting wheel-lattice 8. It is provided that the reactor design has a cover, a nozzle for biogas removal and a valve for substrate selection.

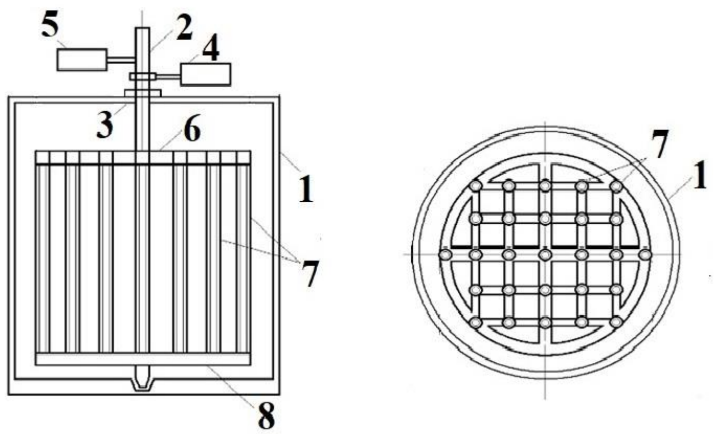

Figure 1. Device for intensifying the release of biogas in the reactor

The design features of the device allow to create an electric field in the environment of the bioreactor, which in compliance with the recommended mode, can increase the output of the biogas mixture, which will reduce its cost and increase the efficiency of the biogas system as a whole. The device can be used not only when designing new, but also when upgrading existing biogas plants of different types.

Therefore, for use in industrial conditions, a special device was developed and proposed to intensify the release of gas in the reactor, which simultaneously performs the function of efficient mixing of the substrate and the source of uniform electric field, creating optimal conditions for the development of microorganisms. Its parameters are determined based on specific application conditions, mass and size characteristics of the device, geometric dimensions of the bioreactor and electrical safety standards, etc.

Methods. The task of calculating the parameters of such a device is to determine the diameter of its vertical electrodes and the distances between them, at which the volume of the bioreactor creates the most uniform electric field of a given intensity, ie deviation from the previously established optimal voltage of 0.95 V / cm [2] should be minimal. For this purpose, the sequence of electric field calculation in the bioreactor in the form of integral equations by the method of secondary sources was used [3, 4]. This method involves the introduction of secondary additional sources in the solvable integral equations.

Research results. Obviously, the distribution of secondary sources cannot be arbitrary, but must satisfy the corresponding integral equation. The calculation of electromagnetic fields by the method of secondary sources can be reduced to the solution of integral equations of Fredholm type II, the properties of which are known and their numerical solution does not cause fundamental difficulties.

According to the method, the density distribution *σ* of secondary charges satisfies the integral equation, the surface charge density *σ (Q)* at the point *Q* located at the boundary of the regions of homogeneous conductivity is determined by the formula [4]:

$$\sigma(Q) = 2\lambda_Q \vec{n}_Q^{\,0} \vec{E}(Q), \qquad (1)$$

$\vec{n}_Q^{\,0}$ - vector of a unit normal to the boundary of the distribution at a point *Q*;

$\vec{E}(Q)$ - tension at point *Q*, created by all charges distributed at the boundaries of the regions, except the charge at point *Q*, B.

The field strength is expressed through the charge density distribution:

$$\vec{E}(Q) = -\frac{1}{4\pi\gamma_0}\int_S \sigma(M)\frac{\vec{r}_{QM}}{r_{QM}^3}dS_M, \qquad (2)$$

r_{QM} - the distance between the observation point *Q* (fixed) at which the voltage is determined and the point *M* (variable) at which the charge is located $\sigma(M)dS_M$;

$\vec{r}_{QM}$ - the vector of the distance between the points *Q* and *M*, directed from the fixed point *Q* to the variable point *M*. Integration is carried out at all boundaries of the areas where the charges are distributed with a density *σ* - secondary sources.

The final components of the electric field strength in the cylindrical coordinate system are equal to:

$$E_p(q) \approx \sum_{j=1}^{n} \frac{\rho_j - \rho_q \cos(\varphi_q - \varphi_j)}{r_{qj}^3} S_j, \tag{3}$$

$$E_\varphi(q) \approx \sum_{j=1}^{n} \sigma_j \frac{\rho_q - \rho_q \cos(\varphi_q - \varphi_j)}{r_{qj}^3} S_j, \tag{4}$$

$$E_z(q) \approx \sum_{j=1}^{n} \sigma_j \frac{Z_j - Z_q}{r_{qj}^3} S_j. \tag{5}$$

Discussion. It is expedient to determine the field strength at the control points of the bioreactor volume using an algorithm developed on the basis of the method of secondary sources, which provides for the calculation of the instantaneous distribution of the latter on the separation surface of media with different conductivities by solving integral equations.

Conclusion. Therefore, it is possible to determine the optimal design parameters of the device for intensifying the release of biogas by using the method of searching the volume distribution of the electric field strength. From a data set such combination at which deviation from the set size of tension is minimum, ie its maximum uniformity is provided is chosen.

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INFLUENCE OF CHANGING CLIMATIC CONDITIONS ON HEAT PUMP EFFICIENCY

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Reserves of traditional hydrocarbons such as gas, oil, coal are declining every year. And their use is associated with a negative impact on the environment.

Today, there is a need to move to greater use of renewable energy sources, which are inexhaustible and can guarantee energy and environmental security.

Among renewable energy sources, the use of low-potential environmental energy, converted to high-potential using heat pumps (HP), is promising.

Experience shows that HP is one of the promising types of equipment for creating heat and cold supply systems.

Currently, in some countries the level of development of this method of heat supply is such that HPs are intensively displacing traditional methods based on direct combustion of fossil fuels.

The source of heat for the heat pump used in the heat supply system can be water, soil and air, wastewater, ventilation air and other heat carriers, the temperature of which is 4-12 ^oC.

In the case of widespread introduction of heat pumps, emissions of CO_2 and carcinogenic compounds formed during the combustion of minerals are also significantly reduced.

According to [1] HPs are used for buildings of energy efficiency class C and above. When using HP in the heating system, the most common is the bivalent scheme, when HP provides most of the heat load for heating the building, and the rest is covered by an additional source: electric, gas or solid fuel boiler. To determine the power of the HP it is necessary to know the bivalent temperature at which the additional source is connected. The bivalent temperature is influenced by many factors: the type of HP, weather conditions (ambient air temperature and duration of their standing).

The value of the bivalent temperature affects the share of heat load that covers the HP, so studies aimed at determining the bivalent temperature depending on the ambient temperature are relevant.

The purpose of this work is to analyze the influence of ambient air temperature on the temperature of bivalent HP in the heating system and determine the proportion of coverage of the heat load by the heat pump.

The paper considers a building with a heating area of 1500 m^2, heating system capacity of 100 kW.

To determine the influence of ambient temperature on the value of bivalence temperature and HP power, statistical data on the duration of the standing ambient temperatures in the heating seasons during 2015 - 2020 years for the city of Kyiv (Table 1) was collected.

Table 1 - The number of hours of ambient temperatures in different heating seasons

Ambient temperature, °C	Heating power, kW	Number of hours of standing temperatures, h.					
		St. year	15-16	16-17	17-18	18-19	19-20
8	29	654	864	732	531	645	831
5	36	1480	1797	1590	1632	1323	1824
0	48	1225	711	1206	1188	1245	738
-5	60	627	105	201	171	324	48
-7	64	336	189	261	243	231	48
-10	71	130	129	159	201	81	0
-15	83	31	63	54	12	0	0
-20	95	5	0	0	0	0	0
-25	100	0	0	0	0	0	0
Average temperature heating season, °C		0,77	3,78	2,16	2,63	2,93	5,43
Number of heating hours, h		4488	3858	4203	3978	3849	3489
Heating load, MWh		230,6	175,4	201,6	176,3	169,5	145,1

The obtained data allowed to build an integrated graph of the dependence of the coverage of the heated load on the capacity of the heating system (Fig. 1).

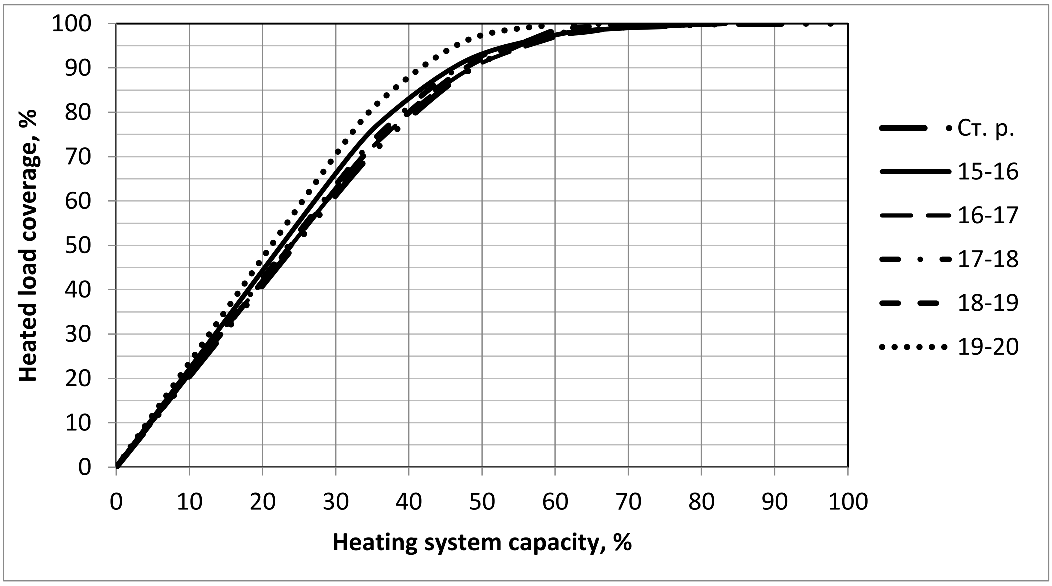

Fig 1. Dependence of the heat load coverage factor on the power of the heating system

From the graph (Fig. 2) it is seen that the increase in ambient air temperature during the heating period shifts the bivalence point towards higher temperatures (decreases the power of the HP and increases the load factor).

For a given building, it is proposed to use HP with a capacity of 60 kW to 70 kW, which will depend on the type of HP and the manufacturer. The bivalent temperature will vary from minus 5°C to minus 9.4°C with coverage of 97-99% of the heat load.

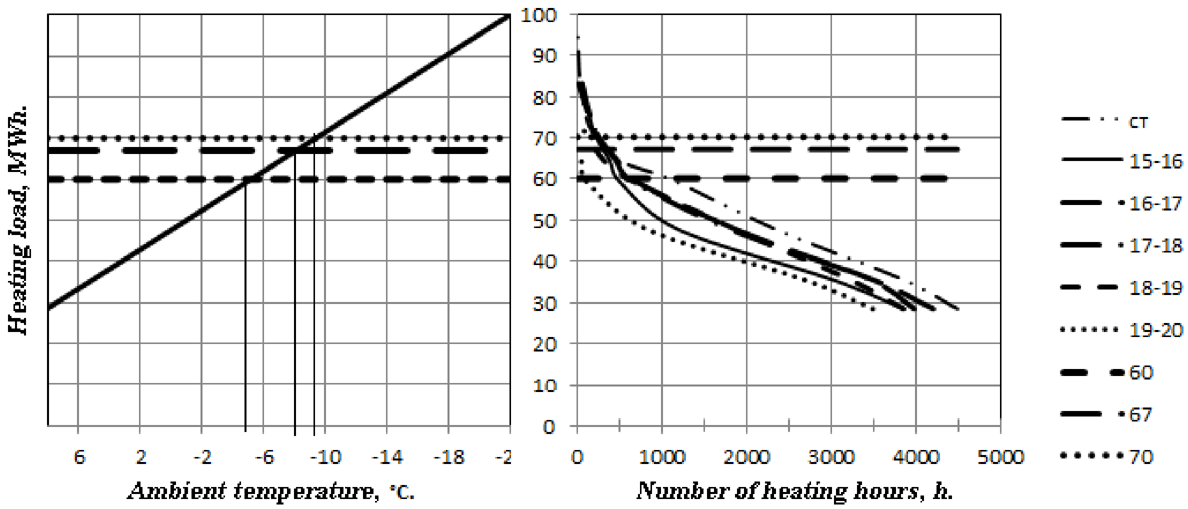

Fig 2. Determination of the bivalence point

Conclusions. For this building, it is proposed to use brine-water HP NIBE F1345 with a capacity of 67 kW, SCOP = 3,8 at a coolant temperature of 55°C [3], which will provide a bivalent temperature of minus 8°C. It will be able to cover from 95 to 100% of the heat load for heating, depending on the year. And the use of underfloor heating in a heating system with a coolant temperature of 35°C will increase SCOP to 4,7 and reduce electricity consumption.

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ANALYSIS OF AIR DISTRIBUTION IN DATA CENTERS FOR INTER-ROW PRECISION AIR CONDITIONING

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Currently, for inter-row air conditioners, it is common to use 3 main air distribution schemes in data centers: a front air distribution scheme, a side air distribution scheme with a one-way air supply, and a side air distribution scheme with a two-way chilled air supply. To analyze the air distribution during inter-row air conditioning in the data center, the method of numerical modeling is used using the STAR-CCM+ hydrodynamic software package. The k-ε turbulence model is selected for calculating the turbulent flows in the STAR CCM+ program. As the initial geometric model in the STAR-CCM+ program, a real data center machine room in St. Petersburg, Russian Federation was selected. In the real machine room of the data center, a front air distribution scheme is adopted. The model does not take into account the operation of the cooling system built into the IT equipment. To perform the calculation for the side air distribution scheme, the existing model of the real machine room of the data center in the program is modified taking into account the data on air conditioners with side air supply. To quantify the simulation results, graphs of the average values of the air velocity at the entrance to each IT equipment for the front and side (with one-way supply) air distribution schemes are constructed, and the unbiased sample variance and the standard deviation of the average speed values relative to the nominal value of the air velocity are calculated. Thus, the greatest spread of the average values of the air velocity relative to the nominal value is typical for a side scheme with a one-way air supply. This means that with the side air distribution scheme, the amount of IT equipment that does not receive enough air volume to cool is greater than with the front air distribution scheme. Insufficient cooling of the IT equipment can lead to its overheating and failure. To reduce the spread of the average speed values for the side air supply distribution scheme, an improvement method is proposed, which consists in adding slats on the rack doors. Thus, the analysis of the air distribution in the data center during inter-row precision air conditioning was performed without taking into account the cooling system built into the IT equipment. This analysis revealed that the use of precision air conditioners with a side air supply scheme in this case can lead to insufficient cooling of the racks in the machine room. The addition of slats in the rack doors reduces the risk of overheating and failure of IT equipment for the side air distribution scheme.

PERSPECTIVES OF USING MICROWAVE HEATING OF PETROLEUM PRODUCTS IN THE TANK

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The developed microwave heating technologies are characterized by high intensity and efficiency. These advantages make it possible to use microwave heating when drying food products, reducing the viscosity of oil, separating the emulsification of oil and water, etc. [1, 2]. According to the analysis of the temperature field of viscous oil, an uneven distribution of the microwave field in the oil tank will cause regional differences in the distribution of the temperature field. From the point of view of analyzing the heat transfer process [3], during microwave heating, oil molecules move rapidly and begin to quickly penetrate other areas. Macroscopically, a hot oil product transfers energy to a region with a lower temperature due to thermal conductivity. At the same time the effect of thermal conductivity is decisive in comparison with natural convection. It was determined [3-5] that microwave energy is intensively absorbed by a viscous petroleum product. Analysis of literature data shows that the method of microwave heating of oil tankers is feasible and it is advisable to study it for further application in industry. However, there is a need to conduct analytical and experimental studies of the process of heating petroleum products in a microwave field to solve certain problems, one of which is the intensification of draining high-viscosity petroleum products from railway tanks.

To describe the heating process of a cylindrical tank with petroleum products exposed to high-frequency electromagnetic radiation, the equation of thermal conductivity in cylindrical coordinates is applied in [6]. The analysis of the work allows us to conclude that the proposed method of mathematical modeling of microwave heating of petroleum products can be taken as a basis. Modeling of microwave heating of high-viscosity petroleum products should be carried out on the basis of the differential equation of thermal conductivity, taking into account internal heat sources. However, it is impossible to use the proposed results directly, since this paper considers the process of heating a large volume, which cannot be described by cylindrical coordinates.

The mathematical model of heating petroleum products in a tank from the action of a microwave source is based on the assumption that heat propagation is carried out in an unlimited array during thermal conductivity under conditions of internal energy sources. Assuming that the thermophysical properties are constant and the power of the microwave field is determined by the action of internal heat sources q_v, the differential equation of thermal conductivity takes the following form:

$$\frac{\partial t}{\partial \tau} = \mathrm{a}\,\nabla^2 t + \frac{q_v}{\rho c_p}, \tag{1}$$

a - is the coefficient of thermal conductivity,

ρ - is the density of the petroleum product,
c_p - is its heat capacity.

The conditions for unambiguity are as follows:
- petroleum product represents an unlimited array;
- the initial temperature distribution of the array is uniform.

The problem was solved in spherical coordinates, for which the Laplace operator ∇^2, provided that the temperature changes only along the radius r, has the following form:

$$\nabla^2 t = \frac{\partial^2 t}{\partial r^2} + \frac{2}{r}\frac{\partial t}{\partial r}, \tag{2}$$

The boundary condition is $\left(\frac{\partial t}{\partial r}\right)_{r=\infty} = 0$, where $r = \sqrt{x^2 + y^2 + z^2}$ (The origin is placed in the volume under consideration).
The finite difference method was used to calculate temperatures.
The following values of physical characteristics were used in the calculations: ρ = 950 kg/m^3, c_p = 3 KJ/(kg·K), L = 300 KJ/kg, λ = 0.125 V/(m·K) [13]. According to [14], for fuel oil, the relative permittivity ε'= 3,5-4,5 and penetration depth tgδ = 0,013-0,03, this is typical for dielectrics, which absorb microwave energy quite efficiently.
In Fig. 1 shows the temperature field in the oil product for different time intervals.

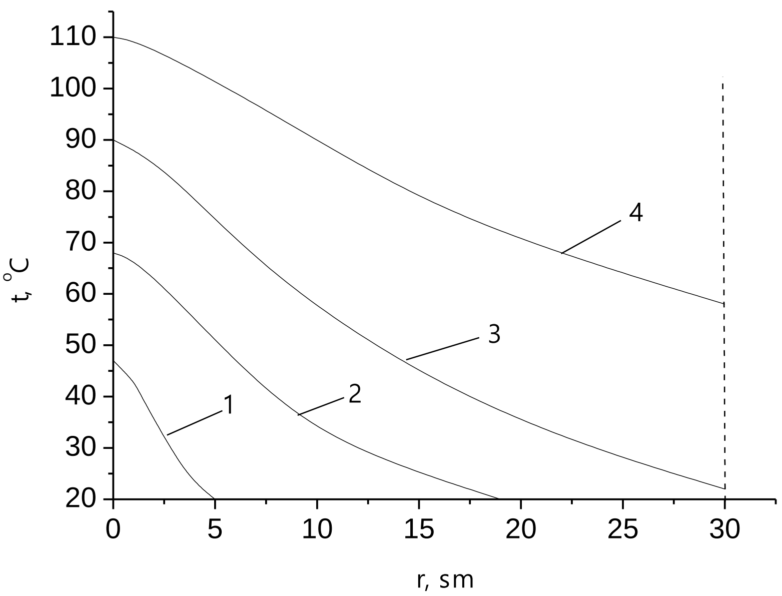

Figure 1. Calculated change in the temperature of fuel oil in the tank during microwave heating
1 - τ = 1 min, 2 - τ = 10 min, 3 - τ = 65 min, 4 - τ = 116 min.

The temperature of fuel oil increases over time, and the front of the heated area expands. After 65 minutes, this front reaches the drain hole, but the oil temperature is insufficient to start the pumping process. From fig. 1 it can be seen that the fuel oil temperature of 60 ºC at the drain hole will be reached in 116 minutes. To increase the flow rate, ensuring that the required temperature is reached at the drain hole of 60 ºC, you can install a magnetron of greater power, for example, 15 kW. Then the consumption will increase to 0.93 kg/s.
When developing a device for microwave heating, the following should be taken into account [7]: the volume of product in tanks can vary widely, and, accordingly, the load resistance (heated

volume) changes, so it becomes necessary to coordinate the latter with the microwave generator in order to avoid damage to the magnetron, which must be reliably protected from load mismatch.

Conclusions

An analytical study of the process of microwave heating of petroleum products is carried out on the example of fuel oil. It is obtained that the temperature of fuel oil increases over time, and the front of the heated region expands. For an initial temperature of 20 ºC after 65 minutes. the hot front reaches the drain hole, but the oil temperature is not sufficient to start the pumping process. Under the specified conditions (initial temperature 20 ºC, magnetron output power 3 kW, relative permittivity ε'= 4.5 and loss angle tangent tgδ = 0.03), the fuel oil temperature of 60 ° C at the drain hole will be reached in 116 minutes. To increase the flow rate, ensuring that the required temperature is reached at the drain hole of 60 ºC, you can install a magnetron of greater power, for example, 15 kW. In this case, the consumption will increase to 0.93 KG/s.

The circuit solution for a microwave device allows you to place the radiation source in close proximity to the drain hole. It is suggested to place the microwave device in a hollow pipe that can be connected to the upper hatch. The microwave energy emitter comes out of the lower base of the pipe and is located directly above the drain at a distance of 1.5 penetration depth. With this location of the source relative to the drain hole, heating, and, accordingly, a decrease in the viscosity of the oil product, will be observed in the drain zone, which can significantly intensify the process and reduce energy costs for heating.

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THE TEMPERATURE DISTRIBUTION OF THE MATERIALS IN THE CONVECTIVE HEAT TRANSFER

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With the intensification of heat transfer processes (IHTP), a significant increase in operating temperatures and increased requirements for precision of thermotechnical calculations gradually and increasingly become apparent shortcomings of well-developed theory based on linear boundary value problems of heat conduction (BPHC). Therefore, in the last few decades, increased attention is paid to the non-linear mathematical modeling (MM) of thermal processes.

Due to the mathematical difficulties exact solutions of nonlinear BPHC obtained only for some special cases. Generally, for this purpose, there are various approximate methods, which can be divided into two fundamentally different groups. The first group includes well known [1 ... 3] exact methods for solving linear BPHC. But the use of these methods must be preceded by the corresponding output linearization of nonlinear problems. The second group includes those approximation methods, which allow to solve directly nonlinear BPHC without prior linearization. This is primarily numerical and analogue methods and approximate analytical, integrated, variation, disturbance (small parameter) and others [4]. These include the method of equivalent sources (MES).

As for the research of thermal processes in the bodies of functionally dependent TFH are (non-linearity of the first kind), they are related to the difficulties caused by not only non-linear equation, but the fact that laws change TFH not reliably identified. Of course [4] the dependence of the thermal conductivity $\lambda(T)$ and volumetric specific heats $C(T)$ are approximated by polynomials

$$\lambda(T) = \lambda_0 + \sum_{i=1}^{n} \delta_{\lambda i} T^i \; ; \qquad C(T) = C_0 + \sum_{i=1}^{n} \delta_{Ci} T^i$$

For many materials coefficients T^i are so small that $i \geq 2$ members can be negligible that to some extent consistent with modern concepts of the mechanism of thermal conductivity as a superposition of streams of photons and electrons, where the fate of metals in the past for a specific temperature range is dominant.

So in future we may consider nonlinear (quasi-linear) differential equations containing linearly dependent on temperature TFH that after going to the relevant dimensionless quantities take the form of:

$$\lambda(\theta) = \lambda_0 \cdot \overline{\lambda}(\theta); \quad \overline{\lambda}(\theta) = 1 + \varepsilon_\lambda \theta;$$
$$\varepsilon_\lambda = \delta_\lambda (T_C - T_0)/\lambda_0;$$
$$C(\theta) = C_0 \cdot \overline{C}(\theta); \quad \overline{C}(\theta) = 1 + \varepsilon_C \theta;$$
$$\varepsilon_C = \delta_C (T_C - T_0)/C_0;$$

Introduced here coefficients ε_λ, ε_c will be called nonlinear parameters of the 1st kind.

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DETERMINATION OF THE TEMPERATURE CONDUCTIVITY COEFFICIENCY OF BULK BIOFUELS

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Abstract: The article presents the results of the experimental determination of the temperature conductivity coefficient of bulk fuel of plant origin. Solid bulk materials formed by solid and gas phases are widely used in human economic activities. The quality indicators of heterogeneous materials and products made from them are often determined by heat transfer modes, which depend on thermophysical properties: thermal conductivity, heat capacity, thermal diffusivity. These properties are functions of the properties of its solid and gas phases, pore volume or space between solid particles and the specific volume of the solid phase, etc. This leads to a significant effect on the thermophysical properties of mechanical, thermal and physicochemical effects that materials are exposed to during operation. Therefore, the solution of the problems of developing calculation methods and experimental determination of the properties of heterogeneous materials, taking into account their structural characteristics, is of particular relevance. Thus, the problem of controlling the thermophysical properties of heterogeneous materials which are formed by solid and gas phases, and which are subject to mechanical, thermal, physicochemical effects during operation, is of current interest. The paper presents a method for measuring thermophysical properties based on the use of a regular regime of the first kind at various stages of the measurement process.

Key words: flowability, temperature conductivity, a-calorimeter, cooling rate.

Introduction

Bulk materials can be found in nature, in industry and in everyday life. Despite its wide distribution, bulk materials are still the object of study.

While developing and designing equipment, used for pressing bulk fuel, some difficulties often arise due to the lack of reliable information on the thermodynamic properties of the bulk medium. Available in the literature information on thermodynamic properties is limited and does not allow interprete the obtained patterns for other systems. The possibilities of theoretical calculation methods are extremely limited and only provide qualitative information on the properties of bulk solids.

The process of compression of bulk material in a closed volume of the working body leads to an increase in density, thermal conductivity, and this leads to the changes in the physical and mechanical properties of the medium.

The subject of research is raw materials for the production of solid biofuels (sunflower husk, buckwheat and barley husk, sawdust.

The optimal temperature regime is one of the most important stages in a rational technology for the production of pellets and briquettes, since depending on the selected mode, one can get a product with a certain range of properties.

ENERGY EFFICIENT TECHNOLOGIES AT OIL FIELD FACILITIES

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It is known, oil is one of the most important minerals ensuring the well-being of many countries including Russia. More than 500 million tons of oil are produced annually in our country. However, a significant number of oil fields are located in the Arctic zone. The development of such fields is complicated by climatic conditions, difficulties in cargo delivery and oil transportation, as well as distance from powerful energy systems. All equipment of the well facility, complex for preparation of commercial oil and external oil transportation is a powerful consumer of electric and thermal energy. Electricity generation is provided by diesel power plants. A significant amount of diesel fuel is also spent for thermal energy generation, taking into account the low temperature for heating calculation (-39 °C) and, accordingly, the duration of a heating period of 289 days. Difficulties in diesel fuel delivery, the state policy in the field of improving the energy efficiency of production processes, the need to solve environmental problems in oil fields specify the search for alternative methods of heat and electricity generation. One of the most appropriate solutions to reduce diesel consumption is the use of associated petroleum gas (APG). In the analysis performed, the use of APG significantly reduces greenhouse gases emission into the atmosphere. In Russia, APG is used primarily for oil heating in special process furnaces at the stages of marketable product obtaining and during preparation for transportation. However, such furnaces do not provide heat recovery for a system for exhaust gases, the temperature of which reaches 580°C. The use of gas-water heat exchangers widely used at compressor stations would provide the rotational field camp and production facilities with thermal energy, thereby reducing diesel fuel costs. When using gas power plants currently the gas turbine exhaust gases with a temperature of about 500 °C are removed without recovery, increasing thermal pollution in the atmosphere. In addition, the non-regenerative cycle has a rather low efficiency (28-32%). In order to save energy, it is proposed to use a regenerative cycle to heat cyclic air, which is also widely used in gas turbine units for gas blower driving at compressor stations of gas mains. The use of the regenerative cycle according to the analysis will decrease the specific consumption of fuel gas corresponds to an increase in efficiency up to 37%. A modern energy-efficient automated induction-resistive system, which allows maintaining the necessary temperature conditions for transportation of oil (40 ... 45 °C) is implemented at the enterprise for electric heating of the main pipelines. However, in order to avoid inefficient energy consumption, the system requires modernization, which consists of installation of additional temperature sensors on each parallel pipeline upstream of a common collector. In terms of using alternative energy sources at the fields under question, wind generators can be used to generate electric energy, since the facilities are located in the way of the Bolshezemelskaya Tundra with a flat relief. The average wind speed is 4-8 m/s, which determines the feasibility of wind generators application. One of the latest trends in energy saving, in particular in the European Union, is the use of energy cogeneration. However, distributed power generation does not currently exclude the need for the delivery of diesel fuel, preparation for the respective generators of fuel gas energy (APG), "crude" oil or its highly viscous derivative products. To solve this problem, it is proposed to use the well known Stirling heat engine. The energy-saving technologies associated with the need for temperature stabilization of rocks in permafrost conditions to avoid deformation of production wells and potential emergencies have the special place in the development of oil fields. Permafrost rocks in the Bolshezemelskaya Tundra occupy almost the entire central and northeastern parts of the district. The thickness of permafrost rocks reaches 500 meters, the temperature ranges from −5 °C to −2 °C. Currently, the following technical solutions are used to prevent permafrost thawing: strapping and suspension of casing strings, thermal insulation of casing strings and construction of seasonal cooling devices (heat pipes). A more reliable method to prevent soil from thawing is to use geothermal heat pumps, while heat can be used for heating and hot water supply facilities.

INFLUENCE OF THE PARAMETERS OF THE HEATING AND HEATED AIR ON THE HUMIDITY OPERATION MODE OF THE COUNTERCURRENT PLATE THERMAL REGENERATOR

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The climate of the Russian Federation is diverse, but throughout its territory, saving heat for heating the ventilation air is relevant. One of the ways to reduce the heat demand for heating the supply air is to recover the heat of the exhaust air.

During the heating period, the temperature and humidity of the outdoor heated air vary widely.

Moreover, for a significant part of the heating period in the greater territory of the Russian Federation, the regenerator operates in the mode of the condensate precipitation from the exhaust air. Since the condensate precipitation often occurs at a negative temperature of the supply air, the risk of freezing of the thermal regenerator is high, so quite a lot of works are surveying this issue and the ways to combat this phenomenon. This paper does not address this issue. It is aimed at identifying the combinations of temperature and humidity of the exhaust heating and supply heated air, at which condensation occurs. Firstly, this will enable the assessment of the frequency of a particular humidity mode of operation of the thermal regenerator in a particular region and, secondly, to make an informed decision on how to combat its freezing up.

To perform this task, provision has been made of the heat exchange characteristics of the thermal regenerator at the exhaust air temperature from 20 ^oC to 26 ^oC in 2 ^oC increments and the relative humidity from 15 % to 65% in 5% increments. The temperature variations of the heated air varied from -25 ^oC to +15 ^oC. The moisture content of the supply air was taken from almost zero to the maximum possible at each temperature specified for the supply air.

To calculate the heat transfer of a plate thermal regenerator, the method of V. N. Bogoslovsky and M. Ya.Poz was adopted, which takes into account the influence of condensation processes on the

value of the heat transfer coefficient through the thermal regenerator wall by increasing the heat transfer coefficient from the exhaust air side by applying the conditional heat capacity of the water

vapor at 100% air humidity. Since the condensate precipitation is accompanied by the release of heat, the calculated heat transfer coefficient in the dry operation mode is slightly lower than in the case of moisture condensation. The calculation takes into account the influence of the moisture content of the moist air on its density and specific heat capacity.

The calculations performed, the specific results of which are given in the article, showed that the higher the exhaust air temperature, the higher outdoor air temperature should be to enable the dry mode maintenance. So, at the exhaust air temperature of +20^{o}C and at all the considered relative humidity values, the dry mode is possible at the outdoor air temperature of +15^{o}C and above. At the exhaust air temperature of +26^{o}C and at the supply air temperature of +15^{o}C, the condensate begins to fall out of it at a humidity of 55%. Of course, the higher the relative humidity of the exhaust air, the higher the temperature at which condensation begins. At the same time, the condensate precipitation at an outdoor temperature of -10 ° C and below occurs at all the considered values of the exhaust air relative humidity, if the moisture content of the outdoor air is minimal. If the moisture content of the outdoor air is as high as possible at a temperature of -10^{o}C, the heat exchange occurs in the dry mode at a temperature of 22 ^{o}C and above and at its relative humidity of 15%, which indicates that the humidity of the supply air affects the condensation process from the exhaust air very weakly.

It is interesting to note that the freezing of condensate occurs at an outdoor temperature below 0 ° C. Such a temperature in the average multi-year section is observed in Novorossiysk for 490 hours, in Moscow for 1250 hours, in Murmansk for 1720 hours.

NATURAL VENTILATION IN MULTI-STOREY RESIDENTIAL BUILDINGS WITH WARM ATTICS

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Abstract. Natural ventilation is usually designed for residential buildings in Russia. To save energy in multistorey buildings, the exhaust of ventilation channels enter in a warm attic. Warm attics of residential multi-storey buildings are considered as a sealed element of the natural ventilation system and represent a static pressure chamber. Its main function is to equalize and distribute the speed of the air removed from the ventilation shafts.

Advantages.

• the possibility of utilizing the heat of the exhaust air and saving energy resources for heating the building;

• reduction of the number of ventilation pipes, what simplifies the construction of the roof and increases its durability.

Problems:

• instability of natural ventilation;

• destruction of external walls due to non-stationary temperature and humidity conditions. Structurally, the walls of warm attics do not differ from other floors. However, the destruction of external walls takes place only in the zone of warm attics, as for brick buildings and as for multilayer structures with insulation.

To solve the problems of warm attics was done:

- field survey of a 25-storey residential building;
- calculation of heat and mass transfer processes in the external walls;
- mathematical modeling of heat and mass transfer in the volume of the attic;
- mathematical modeling to determine the velocity and pressure fields at the building facades and on the roof.

The registration of the temperature and humidity of the air in the volume of the attic was carried out in the autumn and winter periods:

- before turning on the heating system;
- during the operation of the heating system.

A measuring complex (iBDLR) was used to monitor the temperature and relative humidity of the air in the volume of the warm attic. To calculate the pressure fields at the building facades and in the attic volume, the STAR-CCM + program was used. Calculations performed:

- for the attic as a sealed volume;
- taking into account the influence of the normative air permeability.

The fields of velocity and temperature in the volume of the attic were obtained, the factors influencing these processes were investigated. The results and their analysis are presented

TRENDS FOR MANAGING OF COAL COMBUSTION WASTES

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Introduction. Generation of electricity in power plants fired with hard coal is associated with the generation of solid waste: fly ash and slag. The growing production of fly ash as a by-product of carbon combustion in thermal power plants has long caused an environmental problem with technological and economic effects in the worldwide context. Also, the increase in regulatory environmental laws demands not only major security controls over industrial wastes (by making them inert) or the re-use of these wastcs in other products (recycling). Hundreds of millions of tons of these materials have accumulated near powerful thermal power plants, which causes significant harm to the environment. Some of the waste is used economically, but a significant part remains in landfills. Energy waste is used in the cement industry and in building ceramics plants. There is a possibility of utilization of ashes in agriculture, e.g. for soil deacidification, in the process of composting, fertilization. The scope of use of ashes is insufficient in relation to the amounts produced. It is connected with the necessity to build and maintain landfills, which constitute a significant ecological and social problem. Storage safety depends on the physical and chemical properties of the waste itself. Deposited ashes in landfills in the atmospheric conditions they pose many environmental threats. Especially it is dangerous to wash out harmful substances through rainwater seeping through the landfill, of which the consequence may be groundwater contamination or surface over a large area. The issue of environmental pollution, the rational use of natural resources, the search for innovative solutions, the improvement of existing systems of production, processing and disposal of industrial waste are urgent problems of the modern world. At the same time, the presence of valuable components in the ash creates undoubted prerequisites for their comprehensive processing, and in this case, the storage facilities should be classified as technogenic raw material reserves of deferred demand. Ash from the TPP, which until recently was treated as waste and a source of air and water pollution, is actually a resource material and has also proven its value over a certain period. In the present paper look at ways to manage solid waste from coal-fired power plants in order to preserve our environment.

Results and discussion. In terms of pollutants, thermal power is superior to any other industry, because thermal power plants are continuous sources of emissions of combustion products and wastewater discharges, which are the cause of chemical and thermal pollution of water bodies. TPPs and CHPs are responsible for 80% of total emissions of SOx sulfur oxides and 25% of NOx nitrogen oxides in Ukraine. Therefore, solving the problem of waste disposal is a strategic task of the state. In recent years, a significant amount of research has been conducted on the use of ash in various sectors of the economy. Ash contains chemical compounds that may well serve as a substitute or source of raw materials in the construction industry, in road design technologies, in the production of ceramics, in medicine, agricultural technology and others. However fly-ash is a by-product material, its chemical constituents can vary considerably, but, all types of fly-ash include oxides of silicon, calcium, iron and aluminum. Depending on source, coal may include one or more toxic chemicals in trace amounts: arsenic, beryllium, boron, cadmium, chromium, cobalt, lead, manganese, mercury, molybdenum, selenium, strontium, thallium, and vanadium [1]. Measures to dispose of ash from coal-fired power plants can be considered from two sides. The first approach is to develop and implement measures to mitigate the effects on the environment. The second is to solve the problems of waste accumulation by disposing of them. Here are some of the potential uses of ash.

Use *of Fly Ash for Road and Embankment Works.* Numerous experimental studies have shown that fly ash can be used as a component of the asphalt mixture. One of the examples of the use of TPP ash in road design is England and America, where ash is actively used in road construction as a stabilizer or as a filler in the base layer of the upper layer of pavements [2]. On the experimental section of the highway A 52 was designed a roadway using the technology of recycling fillers and asphalt. Cement, TPP fly ash, blast furnace slag and lime were used as components of the road surface. The results of monitoring of the experimental section of the route A 52 during the year testified to the satisfactory efficiency of the operational characteristics of the pavement.

Asphalt concrete. Studies of the possibility of using TPP ash in asphalt mixtures began around the middle of the last century. TPP ash is used to prepare a bituminous solution instead of a certain amount of bitumen [3] in order to improve its properties, resistance to deformation, stiffness, viscosity at high temperatures and temperature sensitivity.

Phytomelioration. The chemical composition of the ash obtained in combustion technologies varies depending on the composition of the source fuel. Ash can have acidic or alkaline properties, which can be useful for buffering the pH of the soil [4]. Recent studies show that TPP ash has been used successfully to improve degraded soils in combination with organic additives such as cow manure and sewage. High concentration in the ash of elements such as K, Na, Zn, Ca, Mg and Fe, increases crop yields. However, the application of dry selection ash may tend to accumulate elements such as B, Mo, Se and Al, which at high concentrations reduce crop yields and, consequently, affect animal and human health. The use of volatile ash can also reduce the absorption of heavy metals, including Cd, Cu, Cr, Fe, Mn and Zn in plant tissues, which may be associated with increased pH of the ash. There are also studies on the use of ash in the technology of composting sewage sludge. This technology proposes to use TPP ash, which contains a high content of CaO, to increase the pH of compost as a substitute for lime. Increased acidity of the substrate for composting sewage sludge leads to the destruction of pathogenic microorganisms and reduces the availability of heavy metals enriched in sludge.

In agriculture, fly ash is used mainly for land reclamation. Because ash has sorption properties, it is an attractive component for the reclamation of saline soils. The main factors limiting the use of coal-fired thermal power plant ash in agricultural areas are heavy metals and radioactivity.

Fly ash Bricks. Brick is one of the most sought-after building materials in the world. A large amount of clay is used in brick production technology. Extraction of clay from soil has a negative impact on the environment. Open pit mining removes soil layers along with existing flora. The consequences of such actions are the disruption of the soil ecosystem and changes in the existing topography of the territories. Removal of the soil horizon can lead to blocking the movement of groundwater, which creates conditions for an increase in surface movement of water, and, as a consequence, ravines are formed, the relief changes, indicators of soil acidity [5]. In addition, clay is fired in energy-intensive kilns at high temperatures. One of the most effective methods of ash disposal, which is quite actively introduced in Europe and some Asian countries, is the production of bricks. Studies have shown that the quality characteristics of fly ash produced at TPPs have high pozzolanic activity, low content of unburned carbon. Therefore, ash can be a cheap substitute for clay in the composition of the raw mix for the production of bricks. One of the most effective methods of ash disposal, which is quite actively introduced in Europe and some Asian countries, is associated with the production of bricks. Studies have shown that the quality characteristics of fly ash produced at TPPs have high pozzolanic activity, low content of unburned carbon. Compared to traditional clay brick making technologies, fly ash brick production is less mercury-contaminated, energy efficient and costs about 20% less than conventional clay brick. In addition, to date, research is being conducted on the use of a mixture of ash and gypsum for brick production. When gypsum is added to fly ash and a mixture of lime, calcium aluminates are converted into calcium aluminosulfates, which increase the strength characteristics of bricks. Moreover, the curing of products occurs at ambient temperature.

Fly ash in Concrete. Waste from coal-fired power plants is used in concrete production technologies. Ash and slag have astringent properties that determine the possibility of their use in the production of concrete. When fly ash particles interact with water and free lime present in the cement matrix, additional cementitious materials are formed (pozzolanic activity of fly ash. Fly ash is used in concrete primarily due to its pozzolanic and cementing properties. The pozzolanic reaction between fly ash and lime produces less heat and therefore reduces the possibility of thermal cracking when fly ash is used to replace Portland cement. These properties increase strength and can improve performance. Adding ash to concrete reduces the cost of the product, improves its quality characteristics, increases the durability and strength of hardened concrete.

Fly ash in flowing fill. Fly ash can be used in the production of controlled low strength materials. This mixture contains 90-95% fly ash, 5-10% Portland cement and water.

The main physical characteristics of flowing fill mixes include strength development, flowability, cure time, and seepage / settling. This material is used as a structural filler and for use in confined spaces such as narrow trenches, sanitary and storm sewer pipes. This aggregate has hydrophobic properties and frost resistance, which allows it to be used in various climatic conditions in the areas of backfilling, structural filling, insulating backfill, erosion control, base pavements, underground structures, etc. power plants on a large scale will help increase fly ash use and create stronger and safer road infrastructure.

Conclusions. Coal-fired power generation technologies have the greatest impact on the environment. Emissions from thermal power plants are one of the main sources of air, water and soil pollution, and ash and slag waste require huge areas of land for storage. To preserve our environment, additional research is needed to develop measures to reduce the man-made impact of the areas adjacent to the TPP. The chemical composition, morphology, and properties of TPP waste depend on the quality of coal, combustion technology and particle size distribution of ash.

Examples of successful use of ash in various sectors of the economy show that there is an urgent need to study the possibility of using this waste in recycling technologies associated with the reuse of ash for 100% use.

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INFLUENCE OF WATER-EMULSION FUEL PARAMETERS ON ITS COMBUSTION QUALITY

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The development of the state's economy, and together with industrial enterprises, leads to an increase in demand for all types of energy. This is characterized by the constant search for and research of new advanced types of energy and sources of raw materials, especially for thermal power plants and the utilities sector, which use liquid fuel - usually residual fuel oil. For such enterprises it is important to have a stable and timely supply of fuel, improve the methods of its combustion, reduce the cost of energy, and so on. Existing technologies for combustion of heavy flooded hydrocarbons are of poor quality and, accordingly, it is advisable to develop new or optimize existing ones. This will reduce energy consumption for technological preparatory operations for fuel dehydration, reduce the number of oil-contaminated water bodies, and minimize their harmful effects on the environment. One of the promising areas for solving the combustion of heavy flooded hydrocarbons is the use of water-emulsion fuels (WEF). The stability and combustion efficiency of such a fuel emulsion will significantly depend on the available amount and dispersion of water in the WEF. Today, such emulsification technologies and WEF parameters are still insufficiently studied and therefore have great scientific and practical significance.

The goal of the work

To improve the combustion quality of WEF it is necessary to determine:

- the optimal area of change of water dispersion in the fuel;
- limits of water content in fuel;
- the relationship between dispersion and water content.

Presentation of the main material

According to [1], the maximum water content of the fuel oil burned was 63.8%. However, the use of such highly watered fuel oil was inefficient. The author [1] argues that even with any forcing of the furnace, increasing the water content in the fuel by more than 25-30% is impractical, because there is a significant decrease in the theoretical temperature due to the dilution of combustion products with water vapor. The author also emphasizes that the fuel content of 25-30% of water is proportional to the increase in the concentration of water vapor in the combustion products of "dry" fuel oil by only 2-3% (taking into account the partial pressures). Therefore, the limit value in fuel oil should be considered 25-30% of water. At higher water content, the effect of the catalytic effect of water will be completely exhausted due to the decrease in temperature in the combustion zone. The positive effect of water (water vapor) is compensated and even overlapped by its negative impact as an inert solvent of the combustible mixture.

Adamov V.A. [2] claims that for high-quality grinding of fuel sometimes water is added to it, preferably not more than 10%. When the moisture evaporates, the vapor bubbles burst, causing its secondary grinding.

According to [3], the optimal amount of water in the emulsion is 6-14%, and the dispersion - from 1 to 35-45 μm.

Scientists [4] studying the dependence of the influence of the size of the dispersed phase (the size of water droplets) on the combustion rates came to the conclusion that the optimal dispersion of WEF with a water content of 12% is 10 μm.

Scientists Akimov A.V. and Ageev M.S. [5] emphasize the correct dosage of moisture in the fuel. Because at its reduced quantity the effect of neutralization is incomplete, and at excessive concentration of NO and CO remain practically constant as before its giving. The authors believe that the optimal humidity of the fuel emulsion (FE) is 12%. As for the dispersion, at its large size there is a decrease in the temperature of the torch due to ballasting the water vapor of the active combustion zone. Also, the optimal size of the dispersed phase will depend on the dispersion of the fuel spray. At the size of the dispersed phase of 35… 43 μm, the NO_X concentration was the maximum (the minimum concentration of soot parts corresponds to the maximum value of the NO_X concentration).

In [6], to reduce nitrogen oxide emissions by 30%, it is proposed to burn WEF with a water content of 15%. This amount of water as optimal is proposed in [7]. In particular, the authors [7] investigated that the optimal water content in the water-oil emulsion is 10-15%. At the same time increase of luminosity of a torch is observed, process of combustion of fuel oil intensifies, coking of nozzles decreases and concentration of soot particles decreases. The use of such WEF increases the efficiency of heating units by 3-5%, reduces emissions of carbon monoxide and soot by 50-80%, nitrogen oxides by 30-35%, benzo (a) pyrene by 50-90%.

The optimal part size [8] of the dispersed phase is 3-10 μm during oil combustion. Based on the conducted experiments, its combustion is not allowed in the furnaces of steam boilers without their prior modernization. Otherwise, the fuel will not burn completely, the combustion tract will be polluted and the products of incomplete combustion will be deposited, which are flammable and explosive, the toxicity of flue gases will increase. Instead, when preparing crude oil for combustion as a water-fuel emulsion, complete fuel combustion is achieved (with almost zero chemical and mechanical underburning), which also eliminates the occurrence of emergencies. At the same time there is no need to modernize the furnace-burner devices of the boiler. In [8] it was possible to prepare a fuel oil-oil emulsion from crude oil, the combustion of which in the steam boiler was observed reduction of chemical annealing by 4 times and maintaining the efficient operation of power equipment.

High dispersion is the main requirement for the preparation of stable WEF [9]. The stability of the emulsion mainly depends on the deposition rate of large water droplets larger than 30 μm. It is believed that drops of this size in the volume of the emulsion should not exceed 10%. Therefore, when formulating requirements for fuel emulsions, the following requirements apply [9]:

- the size of water particles in the fuel should be within 5… 10 μm;
- providing the possibility of preparing an emulsion of any required composition with a humidity within 5… 50%.

From the above it follows that with a decrease in the diameter of the water droplets improves the quality of combustion of WEF. However, according to [8], extremely small water particles do not provide the desired effect of "micro-explosion", due to which additional grinding of fuel droplets after their spraying in the combustion zone. The maximum effect of "micro-explosions" for water-fuel oil emulsion is achieved at an average value of the diameter of the dispersed water particles of 5-15 μm (0.1 of the diameter of the drop of emulsion sprayed by the nozzle in the furnace). The authors [8] point out that the phenomenon of "micro-explosion" for each water-fuel emulsion has an optimal value of dispersion, which achieves the maximum effect of combustion.

Conclusions

The analysis of known researches shows contradiction of theories which describe influence of dispersion and quantity of water in fuel emulsion on indicators of its combustion. This is due to the influence of different physical characteristics of the fuel: composition, viscosity, temperature, etc.

It is established that the optimal value of the amount of water in WEF is in the range of 3 - 30%. The value of water dispersion in WEF has a wide range - from fractions of a micrometer to several millimeters, but its optimal values are in the range of 1-35 μm.

Since the studies were conducted under different conditions and modes of operation of the equipment, with different fuel emulsions, it can be assumed that there is an optimal relationship between the dispersion and the amount of water in the WEF. Determining the quantitative characteristics of this ratio requires additional experimental studies.

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SIMULATION OF BIOPELLET COMBUSTION PROCESS IN LOW POWER BOILERS

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Bioenergy is the main source of renewable energy, providing 86% of renewable heating and cooling in the EU (58,6% of total renewable energy consumption) [1].

The total consumption of biofuels in Ukraine in 2050 may be 23,1 million toe/year. About 50% of these resources will be used for heat production and will replace natural gas (equivalent to replacing 13.7 billion m^3/year of natural gas). The other part will replace coal and nuclear generation for electricity generation, as well as petroleum products in transport.

In the period 2020-2050 in Ukraine the use of wood biomass will remain at the same level, but the share of use of straw, stems, sunflower husks, agricultural residues, energy crops, liquid biofuels, solid waste for energy production is growing. This projection is based on calculations that show that the potential of wood biomass and sunflower grasslands in Ukraine in 2020 has already been used more than 90% [2].

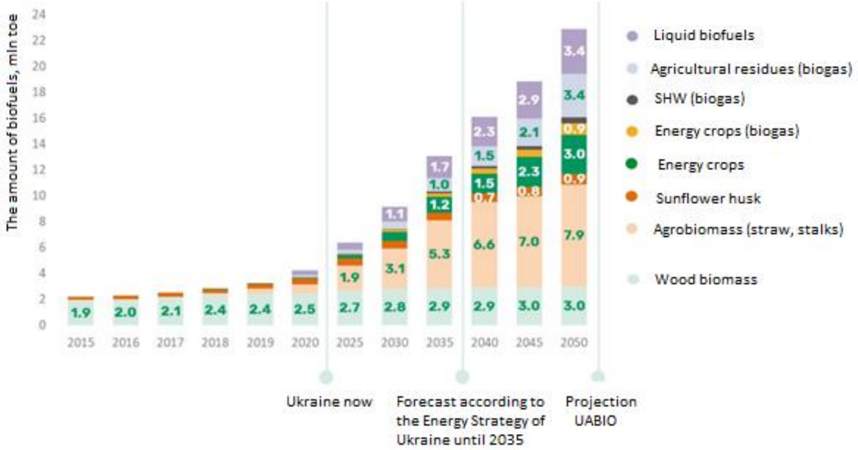


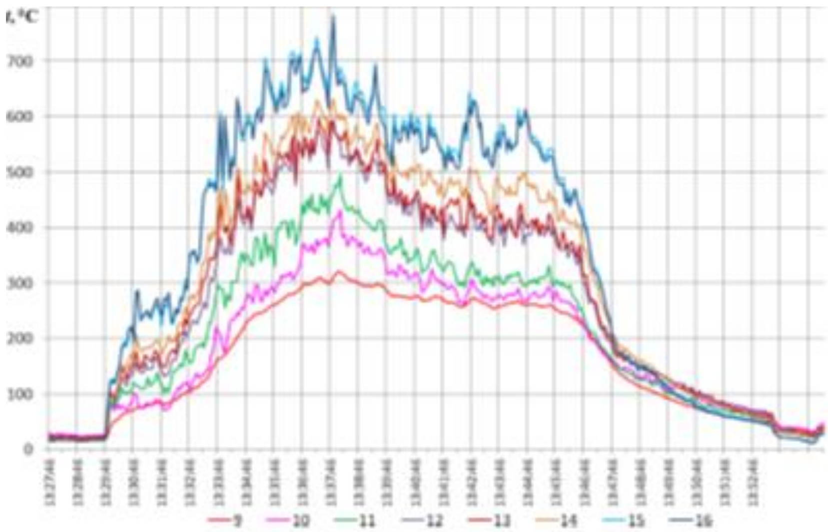
Fig 1. *Biofuel potential in Ukraine*

Biomass combustion requires special boiler design. The basic requirements for biofuel boilers relate to the thermal capacity of the bioboiler; the need for reliable boiler equipment to store and supply of biomass of different humidity, fire-extinguishing system and fuel preparation for combustion; provision of highly efficient systems to clean the gas emissions from boilers from ash and dispersed particles to concentrations determined by the environmental impact assessment project; possibilities of periodic (preferably automated) removal of ashes from the heating surfaces of the bioboiler; ensuring the completeness of fuel combustion; provision of a range of fire safety measures in the boiler room and the composition of biofuels, etc.

The burning of biomass is one of the oldest methods of generating thermal energy. However, there are a number of problems in its practical use, the main one being to achieve deep fuel combustion. Technical devices used for direct biomass combustion include furnaces, furnaces and combustion chambers. Biomass can be used by direct combustion in power plants in a flare, boiling or compacted layer, with subsequent thermal and electrical energy. The main industrial technology in this direction is direct combustion in the boiler and generation of electricity in the steam turbine unit.

Given the relevance of the study of the combustion of biofuels at the Institute of Engineering Thermophysics of the National Academy of Sciences of Ukraine developed and tested an experimental installation (Fig 2) solid fuel boiler with pellet burner for heating a passive type house (Kyiv, Bulakhovskogo Street, 2) [3].

a) *Experimental plant for burning agropelet*



b) *Change in temperature of thermocouple from wheat straw burning*

Fig 2. *Experimental research of the process of burning agropellets*

The developed experimental installation on the basis of a solid fuel boiler with a pellet burner allows to investigate successfully features of burning of vegetable pellets for heating of the house of passive type. In particular, the burning of straw pellets (barley, wheat) and pellets from corn cobs was studied. Based on the technical characteristics of the plant, the aerodynamic combustion processes in biofuel boilers are simulated.

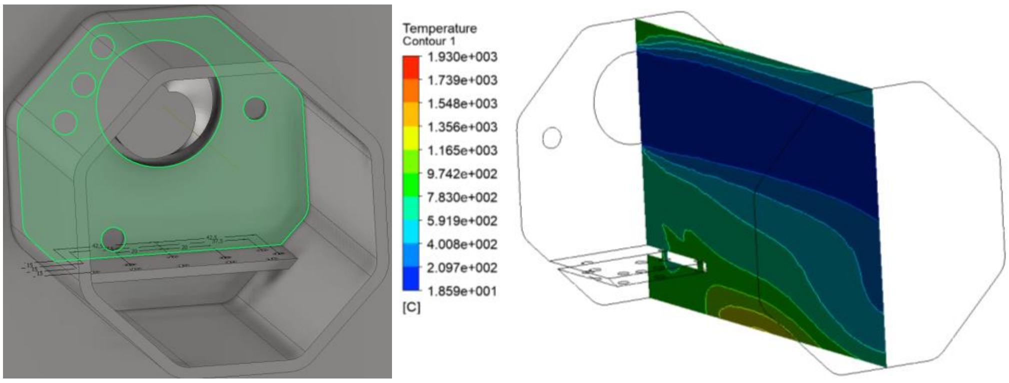
In order to deepen the understanding of the processes occurring during the combustion of biomass in the above-mentioned (Fig. 2) pellet burner, the authors made an attempt to build a CFD-model which is implemented by ANSYS-Fluent.

Modern CFD modelling techniques are known to predict fairly accurately the distribution of mean speeds and temperatures of the mixture in burners of any design, intended for the combustion of homogeneous gaseous and solid fuels. The use of available visualization of the propagation of three-dimensional trajectories of particles in the burner and will perform an in-depth analysis of the aerodynamic structure of the flow, which significantly depends on the distribution of components involved in the combustion reaction. However, such models are not able to predict the processes occurring in the pellet burner during biomass combustion.

To date, it is possible to simulate the combustion of pulverized solid fuel only by modeling the flow of a continuous gas phase when it interacts with the discrete phase of solid fuel particles.

Gaseous combustible substances will be released from the solid fuel particles as they pass through the gas, which will be the source for the combustion reaction. The reaction can be modelled either using a species transport model or a non-premixed combustion model. The question then arises as to which existing techniques would be more suitable for modelling biomass combustion.

Thus, the authors, using the standard method of CFD-modeling, have developed a geometric model of the pellet burner, is a computer copy of the experimental stand, created a finite element model with boundary conditions and a mathematical description of the calculation process as a system of differential equations of motion, continuity and transfer of the i-th component of the mixture and is solved by numerical methods in Ansys-Fluent.



a) б)

Fig 3. *Geometric burner model (a) and the approximate temperature distribution in the cross section of the burner (b)*

Of course, element grid of the model takes into account the development of the boundary layer on the surfaces of the solid-state burner walls. The area outside the boundary layer was modeled using a non-uniform tetrahedral grid. In this case (Skewness) did not exceed 0.8, and the aspect ratio (AspectRatio) of finite elements did not exceed 40.

The calculation was performed in a stationary setting (Steady). An implicit installation algorithm (Pressure Based Implicit) was chosen as the solution algorithm.

The heat properties of the reacting components (in this case air and methane) were given as a polynomial dependence, which depends on the temperature.

As a computational authors used Realisable k-□ model of turbulence, which is known for the presented class of problems has a high computational efficiency.

Conclusions.

The results of the study can be further used to increase the efficiency of the combustion process in the combustion of biofuels and modernization of fuel combustion systems of low-power boilers of municipal and industrial heat, social and budgetary sphere, individual household sector and others.

The computer model is sensitive to the thermophysical properties of the reactants and reaction kinetics, but their correct task will allow a fairly accurate assessment of the aerodynamic structure of the flow in the burners and boiler furnace in which the burner is installed, as well as emission characteristics from biomass combustion.

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STATE AND PERSPECTIVES OF RENEWABLE ENERGY DEVELOPMENT IN UKRAINE

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Introduction. Nowadays, lowering of power consumption of traditional energy sources is a global development priority in energy policy in the world. Currently, Ukraine covers its needs for electricity through using nuclear, thermal and hydro power plants.

Politic and economic situation in Ukraine has meant that majority of thermal power plant units has exceeded the limit of physical wear and tear and need thorough modernization or replacement. Most nuclear power plant units are approaching the design life limit, which caused a sharp decline in their efficiency parameters. The power balance of the power system of Ukraine is characterized by a shortage of regulatory capacity, which leads to irrational use of existing capacity and a high level of losses. There is a growing engagement to reduce the use of coal and nuclear, and instead choose a green product with impact.

That's why particularly important is solving the problem of implementation energy of renewable sources in Ukraine aimed at improving energy efficiency parameters which in turn requires developing of renewable energy based on a systematic approach. At the same time is due to a reduction in the cost of renewable energy, as well as growing require-ments for combating climate change.

During 2010-2020 there is a steady growth trend in the share of renewable energy sources in the world [1]. Based on data published in the material [2] the installed capacity of the unified energy system of Ukraine as of March 2021 is 55069.60 MW and the capacity of renewable energy sources (RES) equipment is 6971.10 MW. Alternative energy in Ukraine is generated by using renewable energy resources such as solar, wind and biofuel energy.

The installed capacity of renewable energy sources for 2015 - 2021 is illustrated in the material [2]. It showed a significant increase in electricity generation by solar power plants. The increase in the number of solar power plants in the power system of Ukraine leads to consideration of the maneuverability and productivity of such generation sources. In particular, the study of methods to increase the efficiency of photovoltaic systems is given in the material [3]. Literature sources imply that modern automated control systems of photovoltaic stations and models for forecasting the amount of electricity is a priority at present [4-5].

However, the above works examine the effectiveness of individual solar power plant. Unified energy system of Ukraine is a complex object that requires a comprehensive study to introduction of highly maneuverable capacities.

According to the Global Wind Report 2021 [6], today, there is now 743 GW of wind power capacity worldwide, helping to avoid over 1.1 billion tonnes of CO_2 globally – equivalent to the annual carbon emissions of South America. The world needs to be installing an average of 180 GW of new wind energy every year to limit global warming to well below 2°C above pre-industrial levels [6].

A comparative analysis of the efficiency of wind energy in the leading countries of the world and in Ukraine is presented in the material [7]. It is established that wind energy in Ukraine has good prospects for further improving the efficiency due to the available wind potential. However, it requires the use of more modern models of equipment with a larger unit capacity.

Modern bioenergy provided 5.1% of total global final energy demand in 2018 [8], accounting for around half of all renewable energy in final energy consumption. In the electricity sector, bioenergy's contribution rose 9% in 2019, to 501 terawatt-hours (TWh). China extended its lead as the largest country producer, and bio-electricity growth also was strong in the EU, Japan and the Republic of Korea. In the energy sector of Ukraine is increased capacity for bioenergy for 2010 - 2020 [1].

Therefore, there is a need to apply a comprehensive approach to study of the prospects of development of renewable energy sources in the energy system of Ukraine.

To analyze the state of development of renewable energy sources of Ukraine was also used a qualitative parameter - the capacity factor (CF), which defined as the ratio of the net electricity generated, for the time considered to the energy that could have been generated at continuous full-power operation during the same period:

$$CF = \frac{E}{P \cdot t} \tag{1}$$

where E is the amount of produced energy during the monitored period (kWh/year (month)), P is installed power (kW), t is the length of the period (in hours per the calendar year (month)).

Capacity factor is defined for electricity from renewable energy sources of Ukraine for 2021 is shown in Figure 1.

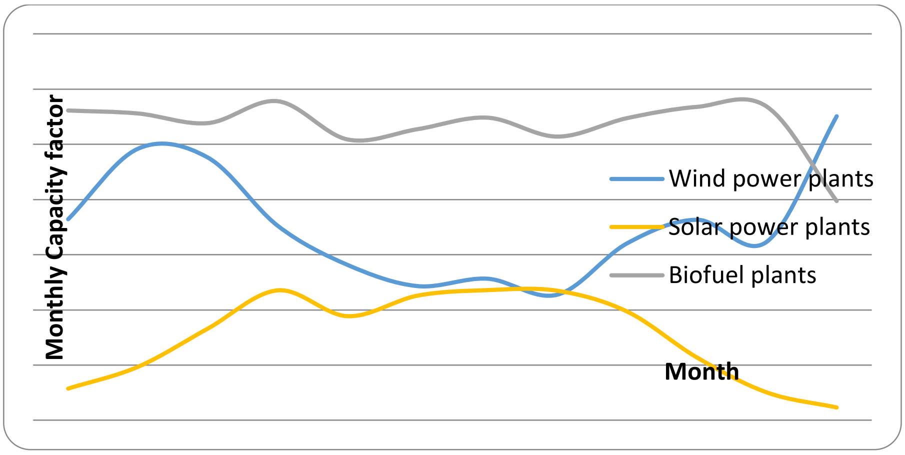

Fig. 1. *Capacity factor of renewable energy power plants for 2021 year*

The value of the capacity factor used to compare different types of electricity production. Values of capacity factor greatly vary depending on parameters of the energy source (fuel), operating conditions, the nature of power output, time, geographic location, and other factors. For individual fuels used in standard power plant in electricity sector, approximate values of capacity factors in Ukraine according to formula 1 are: coal - 0.5, nuclear - 0.63, wind - 0.35, solar - 0.15, biofuel - 0.43. The conducted calculations are showed that the value of the capacity factor of renewable energy sources is insignificant and differ significantly from the factor determined for nuclear and thermal energy.

Based on the calculation of the value of the capacity factor it is established that its value for renewable sources is insignificant. This is due to weather conditions, terrain, etc. Therefore, there is a need to improve maneuverability of power plants in order to ensure the balance of power capacity in the power system of Ukraine.

The power system of Ukraine can accept 3,000 MW of solar and wind power plant capacities without serious deviations in operation [2]. Green energy needs to be balanced.

Maneuvering capabilities in Ukraine are presented by hydro storage, hydro and thermal power plants. Each gigawatt requires about 300 MW of balancing capacity. National power company of Ukraine "Ukrenergo" issued technical conditions for the accession of power value of

RES 7,500 MW for the period up to 2025. If the state does not increase the maneuvering capabilities to help balance the energy system, this means that it will have to limit the production of cheap nuclear energy and increase the use of more expensive and dirty electricity from thermal power plants, which today are also used as maneuvering capacities. This situation necessitates the construction of 2,500 MW of highly maneuverable capacity in Ukraine. It is possible to increase the maneuverability of the power system through consumers. It is necessary to install energy storage facilities in Ukraine. The optimal installed capacity of energy storage system will provide the amount of reserves needed to balance the growing share of renewable energy sources (RES) for the integrated power system (IPS) of Ukraine.

Conclusions.

Based on the analysis, it is established that there is a growing trend to increase the share of renewable sources in Ukraine. The introduction of renewable energy sources in Ukraine will reduce carbon emissions and reduce the cost of electricity, as solar and wind power plants had become very competitive in the market due to low operating costs. The value of the capacity factor as a qualitative indicator is calculated and it is established that its value for renewable sources is insignificant. Therefore, there is a need to increase the maneuverability in order to ensure the balance of power in the united power system of Ukraine.

Thus, there are a number of shortcomings of the current policy of development of electricity generation from RES in Ukraine. Primarily in relation to the reliability of the country's electricity system, economic burden on electricity consumers and imposing an excessive burden of economic costs on end users.

According to world experience, the most significant growth of these indicators for the same period was only in those countries where the strategic priority of energy development has been the active development of RES and an extremely high level of subsidies to the industry.

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RESEARCH OF CYCLONE CHARACTERISTICS FOR DRY CLEANING OF GASES FROM DUST

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The development and application of new, more efficient dust collection units that will help reduce emissions and conserve some very valuable resources for production is an important area of research. With the growth of innovation in technological enterprises, the number of harmful emissions into the atmosphere is growing. Thus, the ecological condition of the environment deteriorates.

The basic analytical dependences which are necessary for construction of a technique of carrying out experiments and calculations of dust catching for concrete working conditions are developed. Methods of calculating cyclones as vortex devices and research of cyclone operation for air purification from dust were investigated.

On the basis of the used basic theoretical positions of heat and mass transfer and thermodynamics at carrying out analytical researches the mathematical model was offered. Calculations of new designs of modern cyclones to obtain their geometric dimensions, resistance and dust capture efficiency were presented. Modern cyclones are designed to more effectively remove dust from the air during various types of work.

Industrial enterprises purify the air that is supplied not only to the shops, departments, but also removed from them into the atmosphere to prevent air pollution in the enterprise and the residential areas attached to it.

Dedusting devices are divided into dust collectors and filters.

Dust collectors include devices in which dust particles are deposited under the action of gravity and inertial forces with a change in speed and direction of air flow. Such devices are dust chambers, cyclones and other devices operating on the basis of centrifugal forces.

Filters are devices in which dusty air is purified by passing through mesh or porous materials (glass wool, gravel, coke, porous paper, fabric, metal mesh).

Dedusting devices can be not only dry but also wet. When using wet dedusting devices, the efficiency of air purification from dust increases.

To date, both the theoretical basis for capturing dust and gas components and methods for calculating various equipment for these purposes have been developed.

Particular attention was paid to cyclone cleaning. The most reliable results can be obtained through experimental experiments conducted on physical models. But for each specific design you need to conduct a separate experiment. More general results can be obtained using a mathematical model of hydromechanical processes of cyclones. Creating a mathematical model of the movement of dust particles in a swirling flow will assess the impact of various factors on the efficiency of dust control in cyclones.

In louvered dust collectors, dust is released from the gas stream under the action of inertial forces when changing the direction of gas flow. With the help of louver plates installed in the flue, the gas flow is divided into two parts.

One stream is 80 - 90% of the total amount of gas and is largely free of dust, the other is 10 - 20% and it concentrates the bulk of the dust, which is then captured in a cyclone or other, quite efficient dust collector. The movement of gas through the cyclone is due to the pressure drop on the louver.

Ultrasonic dust collectors use the ability of dust particles under the action of a powerful sound stream to coagulate, ie to coagulate in a flake, which is very important for the capture of aerosols from the air. These flakes fall into the hopper. The sound effect is created by a siren. The sirens which are issued can be applied in dust-cleaning installations with a productivity up to 15000 m3 / h.

The described devices for air purification, shops and departments of industrial enterprises, which are removed to the atmosphere by exhaust ventilation, do not exhaust all types of dust collectors and filters used to prevent air pollution in cities.

Despite the existing variety of cleaning devices, cyclones are now the most common for cleaning gases from dust due to their low cost, simplicity and ease of operation. In this regard, the development of perforated cyclone requires research aimed at increasing the degree of dust capture from gases, which is considered in this paper.

The calculation of cyclones is reduced to obtaining their geometric dimensions, resistance and dust collection efficiency.

Currently, the most common method of calculating cyclones is the method of generalization and use of indicators obtained by testing cyclones in industrial conditions or on stands.

The method of calculating cyclones using experimental data is based on determining the diameter of the cyclone by the formula

$$D_c = \sqrt{\frac{Q_g}{900 \cdot \pi \cdot W_{con}}},$$

where Q_g - volumetric gas flow through the cyclone, m^3/h;

W_{con} - conditional flow rate of gas in the cyclone, m/s.

Cyclone resistance is determined by the following equation

$$\Delta p = \xi_0 \cdot \frac{W_{con}^2 \cdot \rho}{2}$$

or

$$\Delta p = \xi_0 \cdot \frac{W_{con}^2 \cdot \gamma}{2g}.$$

The speed of the gas in the inlet of the cyclone is determined by the formula

$$W_{in} = \sqrt{\frac{2 \cdot \Delta p}{\xi_{in} \cdot \rho}}.$$

High emissions and low efficiency of cyclones led to significant residual dust in the atmosphere, which required both the development of a new design of cyclones and new theoretical solutions and systems for dedusting of gases. Thus, the generalizing design parameter of cyclones is found in the work and its optimal value is determined, which provides the maximum efficiency of the cyclone, and the analytical dependence characterizing the length of the vortex chamber required to capture the minimum dust particles dmin is obtained.

$$L_{max} = \frac{9D_c}{\varepsilon^2} \cdot \frac{\mu}{\rho_m} \cdot \frac{\sum f}{\pi \cdot R_0 \cdot R_p \cdot \cos\beta} \cdot \frac{R_c}{R_0} \cdot \frac{1}{W_0} \cdot \frac{\left(\frac{R_p}{R_c}\right)^4 \cdot \frac{1}{\cos\beta}}{\left[1 + \frac{R_p}{R_c} + \left(\frac{R_p}{R_c}\right)^3 + \left(\frac{R_p}{R_c}\right)^4\right] \cdot d_{min}^2},$$

where D_c is the inner diameter of the cyclone;

μ - the coefficient of dynamic viscosity of the cleaned medium;

ρ_m - the actual density of the powder;

$\sum f$ - the total area of the pipes of the supply of the cleaned medium in the cyclone;

R_0 - twisting arm (distance from the axis of the supply pipe to the axis of rotation of the cyclone);

R_p - the inner radius of the cyclone pipe;

W_0 is the speed of the cleaning medium at the outlet of the supply pipe.

In the development of large cyclones, their ability to capture dust can be determined on a model of smaller size, but this requires dependencies, which could be converted from model to nature.

Using the dependence determine the size of the dust, which will be caught by the cyclone at the length L of its chamber

$$d_{min} = \frac{3D_c}{\varepsilon} \cdot \sqrt{\frac{\mu}{\rho_m} \cdot \frac{1}{W_0} \cdot \frac{\frac{\sum f}{\pi \cdot R_0 \cdot R_p \cdot \cos\beta} \cdot \frac{R_c}{R_0} \cdot \left(\frac{R_p}{R_c}\right)^4}{\left[1 + \frac{R_p}{R_c} + \left(\frac{R_p}{R_c}\right)^3 + \left(\frac{R_p}{R_c}\right)^4\right] \cdot L \cdot \cos\beta}}.$$

If under index 2 to denote a model cyclone, and under index 1 - a natural variant, the test results of a model cyclone for dust capture with a diameter of d_2 can be converted into dust capture by a natural cyclone d_1.

Cyclones are widely used in industry. There is a great variety of works devoted to the study of the parameters of cyclones and are private, episodic. The great variety of the offered dependences for calculation of their parameters testifies to complexity of the solved scientific problem which does not have the unambiguous answer yet. Despite the adequacy of aerodynamic processes occurring in cyclones, there is currently no single method for calculating the characteristics of the most common in the industry cyclone-vortex devices. The lack of scientifically sound theoretical developments summarizing the large accumulated material for the study of various cyclones prevents the development of scientific and technological progress in the field of creation and improvement of existing cyclones.

As a result of the theoretical researches carried out in work analytical dependences of the basic parameters of a dust collector are defined. These dependencies make it possible to conduct experimental studies of the dust collector and build a streamlined method of calculating dust collectors of new design, which will allow you to design dust collectors with maximum efficiency for specific operating conditions.

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ACTUAL PROBLEMS OF RENEWABLE ENERGY, CONSTRUCTION AND ENVIRONMENTAL ENGINEERING

The time and place of the meeting: 3 – 5 June 2021
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