ENERGY ENGINEERING AND CONTROL SYSTEMS

Volume 1, Number 2, 2015

Analysis of Exergy Efficiency and Ways of Energy Saving in Air Conditioning System for a Cleanroom

Dmytro Harasym, Volodymyr Labay*

Lviv Polytechnic National University, 12, S. Bandery St., Lviv, 79013, Ukraine

Received: September 30, 2015. Revised: October 16, 2015. Accepted: November 02, 2015.

Abstract

In modern technologies, which are related to energy transformation, namely in air conditioning systems, important places are occupied by equipment and processes, the objective estimation of value of its energy perfection can be defined only on the basis of analysis of its exergy efficiency. So, reducing the cost of energy consumed by air conditioning systems preconditions the need for its optimization, which can be fully achieved by virtue of exergy analysis that takes into account not only the quantity but also the quality of energy spent. Nowadays cost estimates can't be the only measure of effectiveness of energy equipment that is using energy resources. Exergy is a physical not an economical criterion and defines its independence from conjectural fluctuations of prices. At the same time, cost parameters don't allow making long term predicting. The minimum should be defined not by financial costs but exergy losses per unit of received heat. Unfitness of just the financial costs is obvious. Analysis of exergy efficiency of the central straight flow air conditioning system (ACS) for cleanroom, that was gained on its innovation mathematical research model depending on different factors, which have influence on its work was presented in this article, and the ways of energy saving for this ACS was proposed. It was found that temperature difference between inside and supplied air in a room, temperature of inside air which depends on temperature of outside air and coefficient of transformation EER of refrigerator machine of ACS have the biggest impact on exergetic output-input ration of chosen air conditioning system.

Keywords: air conditioning systems; cleanrooms; exergy balance; exergy efficiency; energy saving.

1. Introduction

Nowadays for Ukraine the issue of saving energy resources is especially topical under the conditions of market economy with limited resources of the main energy carriers – oil and gas.

That's why in the last decades the fundamental researches of activity of some branches, productions and technologies from the positions of exergy methodology are carried out abroad [6–10, 12, 13, 15, 16]. This methodology was described in the works of R. Clausius, J. Gibbs, G. Guye, A. Stodola, Y. Shargut and R. Petela. The dimension that defines suitability for action (ability to work) of resources of substance and energy was called exergy, and functions that define its value – exergetic. The term "exergy", that defines suitability for action (ability to work) of resources of substance and energy was introduced in 1956.

Energy and exergy flows are always in coexistence. They can be equal to each other in case of flows of mechanical or electrical energy, and can vary greatly from each other in the flows of heat. Exergy does not only quantitatively characterize energy of any kind, but allows estimating its qualitative state. It defines transformability, suitability of energy for technical applying in any given conditions.

As exergy is the only measure of capacity, suitability of energy resources, its applying allows giving an objective assessment of energy resources of any kind. So, exergy is some universal measure of energy resources. Exergy balance, by which the size of using energy resources can be set, indicates the possibility to increase the output-input ratio of the process.

* Corresponding author. Email address: wlabay@i.ua

There are many cases of successful application of exergetic methodology by technical and economic optimization in industry, especially in energy intensive branches. Irreversible loss of exergy can be reduced thanks to combination of heat processes with technological and heat processes.

Nowadays cost estimates can't be the only measure of effectiveness of energy equipment that is using energy resources. Exergy is a physical, not an economical criterion and defines its independence from conjectural fluctuations of prices. At the same time cost parameters don't allow making a long term predictions. The minimum should be defined not by financial costs, but exergy losses per unit of received heat. Unfitness of just the financial costs is obvious.

In some leading European countries and in the USA, exergetic analysis was implemented as a necessary component of engineer projects and modernization plans of productions.

In modern technologies, which are related to energy transformation, namely in air conditioning systems, important places are occupied by equipment and processes, the objective estimation of value of its energy perfection can be defined only on the basis of analysis of its exergy effectiveness.

So, reducing the cost of energy, consumed by air conditioning systems, preconditions the need for its optimization, which can be fully achieved by virtue of exergy analysis, that takes into account not only the quantity but also the quality of energy spent [7, 8, 11–20]

2. Description of the object that is analyzes and of its work

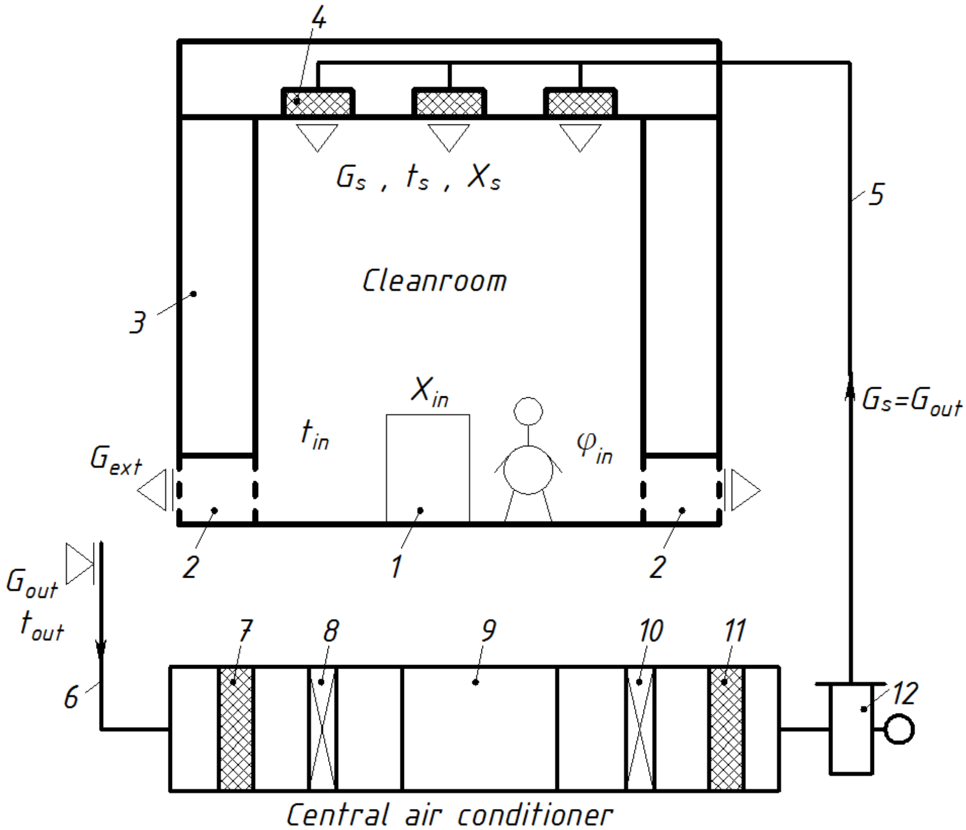

Fig. 1. Basic scheme of the central straight flow air conditioning system for cleanroom:
1 – technological equipment, 2 – air exhaust channels, 3 – gateway premises, 4 – filters of air supply to the room, 5 – air supply duct, 6 – air-intake shaft, 7 – filter of external air, 8 – air heater of first heat, 9 – air washer, 10 – air heater of second heat, 11 – air supply filter in the air conditioner, 12 – fan unit

The aim of air conditioning is to keep up the certain parameters of air in some limited space (in this case, in cleanroom). Usually temperature t_{in} , relative humidity φ_{in} of air are regulated, but in cleanrooms a concentration of dust particles x_{in} in air is also regulated.

Temperature of inside air in a room t_{in} was limited depending on temperature of outdoor air:

when $t_{out} \leq 30$ °C, $t_{in} = 20 \ldots 25$ °C

$$t_{in} = 20 + 0.63 \cdot (t_{out} - 22), \text{°C}, \tag{1}$$

when $t_{out} > 30$ °C, $t_{in} \geq 26$ °C

$$t_{in} = 25 + 0.4\,(t_{out} - 30), \text{°C}. \tag{2}$$

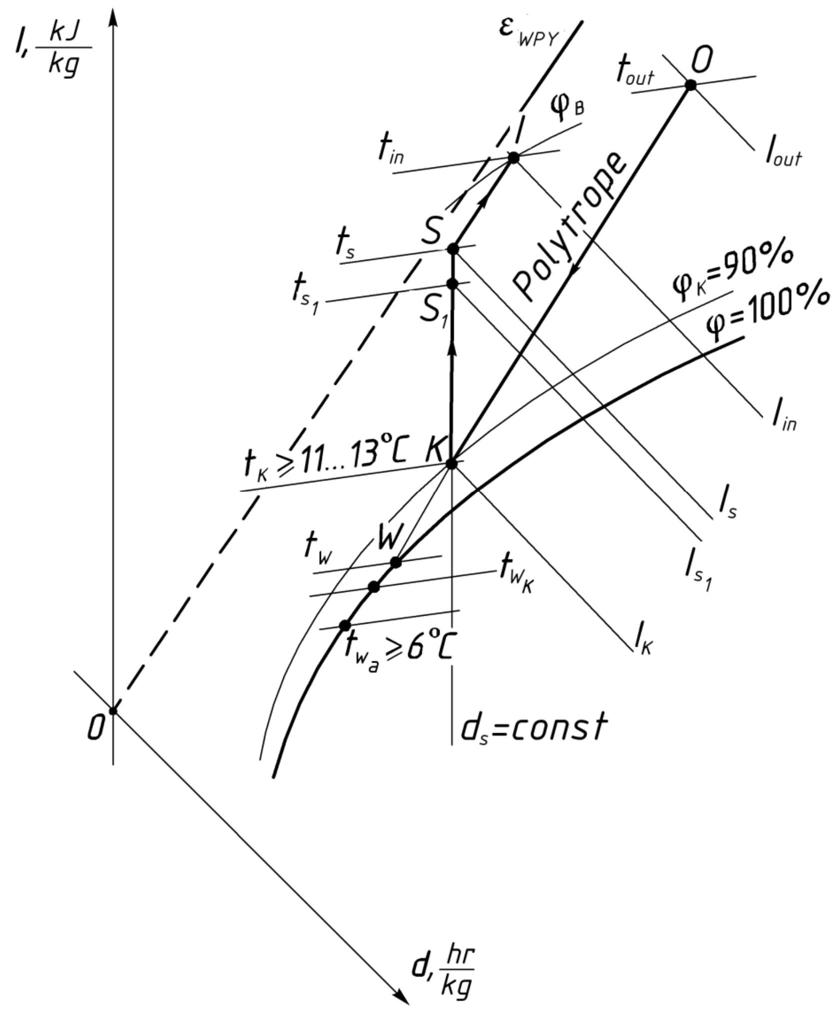

Fig. 2. The image of the process of changing the moist air state in the straight flow air conditioning system in a warm period of year on the I-d – diagram: OK – the process of polytropic air treatment (cooling and drying) $G_{a.w}$=G_S in the air washer; KS1 – the air G_S heating process in the air heater of second heat; S_1S – the air G_S heating by 1 °C in the fan and the supply air duct; SI – the process of excess heat and moisture assimilation in the cleanroom by the supplied air G_S via the air conditioner

The central straight flow air conditioning system, which is shown in Fig. 1, can be applied for this. The work of such system, as we know, depends on the parameters of environmental (external) conditions, i.e. on temperature and moisture content in the air of external environment. Environmental air is taken via central conditioner through the air-intake shaft 6, is cleaned in the filter of outside air 7, then passes through the air heater of first heat 8, is politropical cooled and drained by sprinkled cold water from refrigerator machine in the air washer 9, is heated in the air heater of second heat 10 by warm water from boiler plant, and after that this air is supplied in a cleanroom through the air supply filter 11 in the central air conditioner via the fan unit 12 and air ducts 5 and also via air supply filters 4 at the entrance to the room.

Let's consider the work of this ACS in the warm period of year when moisture content $d_{\text{in}} < d_{\text{out}}$ and temperature $t_{\text{in}} < t_{\text{out}}$. Fig. 2 shows in the coordinate system I, d the sequence of air parameters change, which is passing through the different equipment of central straight flow air conditioning system for cleanroom in the warm period of year. In researches mass productivity of air conditioning system $G_{\text{S}} = 10000$ kg/hr is accepted, which was counted by the number of dust particles, parameters of outdoor air varied within $t_{\text{out}} = 26$–42 °C; $d_{\text{out}} = 9.3$–15.9 gr/kg (specific enthalpy $I_{\text{out}} = 49.8$–83.2 kJ/kg; relative humidity $\varphi_{\text{out}} = 27$–$54$ %, accordingly), barometric pressure $p_{\text{out}} = 1010$ hPa; parameters of indoor air, respectively – $t_{\text{in}} = 23$–30 °C; $\varphi_{\text{in}} = 50$ % ($d_{\text{in}} = 8.8$–13.4 hr/kg; $I_{\text{in}} = 45.5$–64.4 kJ/kg, accordingly); temperature difference between the inside and the supplied air depending on excess heat in the cleanroom $\Delta t_{\text{S}} = t_{\text{in}} - t_{\text{S}} = 1.5$–$6.0$ °C; slope coefficient of excess heat and moisture assimilation in the cleanroom by the supplied air via air conditioner $\varepsilon = 9946$–16858 kJ/kg; water temperature (coolant temperature) for the air washer: initial $t_{\text{W}_{\text{S}}} = 7.0$-$14.9$ °C; final $t_{\text{W}_{\text{F}}} = 9.8$–$17.3$ °C; coefficient of transformation of refrigerator machine $EER = 2.8$–4.4; temperature for the heater of second heat: initial $t_{\text{heat}} = 70$ °C; final $t_{\text{rev}} = 42$ °C.

The sequences of changes that occur with moist air, which passes through the various equipment of air conditioning system, are shown in Fig. 2. Construction on the I-d – diagram was made in accordance to [11]. Air parameters in the characteristic points of the process (Fig. 2) were determined by the adopted values of the parameters for outdoor air and were calculated on the proposed mathematical model by the known analytical dependency for moist air.

Amount of the cold for air treatment in the air washer (cooling capacity of air washer) in the warm period of the year was defined by the equation:

$$Q_C = G_S \cdot (I_O - I_K) \times 0.278, \text{ W}, \tag{3}$$

and amount of the heat for the second air heating in the warm period of year was defined by the next equation:

$$Q_2 = G_S \cdot (I_{S_I} - I_K) \times 0.278, \text{ W}, \tag{4}$$

where I_{O}, I_{K}, i I_{S_1} – specific air enthalpy at the corresponding points of processes that cause the change of moist air state in the central straight flow air conditioning system in the warm period of year on the I-d – diagram, kJ/kg.

The aim of this work was to make analysis of exergy effectiveness of the central straight flow air conditioning system for cleanroom that was gained on its innovation mathematical research model [18–20] depending on different factors, which have influence on its work and to propose the ways of energy saving for this ACS. Material, heat (energy) and exergy balances of the system were made up in this model, which take into account all the possible variants of its work in real conditions.

The concept of exergetic output-input ratio was used for assessment of exergy effectiveness of this ACS which was defined as the ratio of air exergy increase in air conditioned premises E_{out} to the exergy of air conditioning system transmission E_{in}, which was spent on maintaining the process [18–20]:

$$\eta_e = \frac{E_{out}}{E_{in}}. \tag{5}$$

The method for determining the exergetic output-input ratio, which characterizes effectiveness of work of central straight flow air conditioning system for cleanroom in the warm period of year, was precisely described in our articles [18–20].

3. Results of the research work

The values of exergetic output-input ratio η_e was found thanks to settlement and quantitative experiments that was conducted on innovation mathematical research model of the central straight flow air conditioning system for cleanroom depending on temperature difference between indoor and supplied air Δt_S, temperature t_{in} of indoor air and coefficient of transformation EER of refrigeration machine, and was shown as dependencies of exergetic output-input ratio η_e (Fig. 3, 4), and on its base the analysis was made. Therewith little impact of relative humidity of outdoor φ_{out}, indoor φ_{in} and supplied φ_S air (maximum within 10 %) on exergetic output-input ratio η_e was found by us.

It was decided to show results of the research work as such general exponent dependence:

$$\eta_e = C \cdot \left(\Delta t_S / t_{in}\right)^{\alpha} \cdot EER^{\beta} . \tag{6}$$

To get the exponents α and β in the equation (6), results of the research are shown as dependencies in Fig. 3 and 4.

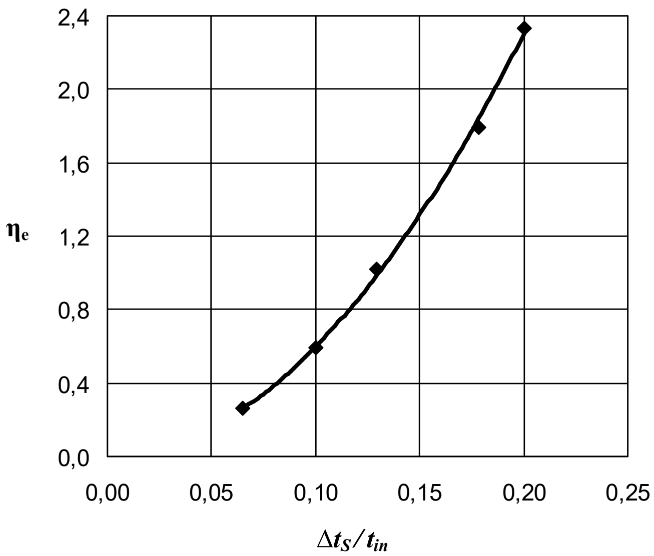

Fig. 3. The dependence of exergetic output-input ratio η_e of the central straight flow air conditioning system for cleanroom on the ratio of temperature difference between indoor and supplied air Δt_S to the temperature t_{in} of indoor air $\Delta t_S / t_{in}$:

G_S = 10000 kg/hr; Δt_S = 1.5–6.0 °C; t_{in} = 23–30 °C; EER = 2.8; t_{out} = 26–42 °C; φ_{in} = 50 %; φ_{out} = 27–44 %; φ_S = 69–54 %

The dependence of exergetic output-input ratio η_e of the central straight flow air conditioning system for cleanroom on the ratio of the temperature difference between indoor and supplied air to the temperature of indoor air $\Delta t_S / t_{in}$ can be presented as such an exponent dependence:

$$\eta_e = 52.6 \cdot \left(\Delta t_S / t_{in}\right)^{1.94} . \tag{7}$$

Maximum error of calculations by the equation (7) is 3.9 %.

So, on the basis of dependence (Fig. 3, equation (7)) the value of exponent α, which is in our case 1.94, was found.

Analyzing the obtained research data on Fig. 3 the following conclusions can be reached. General increase of the ratio of temperature difference between indoor and supplied air to the temperature of indoor air $\Delta t_S / t_{in}$ from 0.07 to

0.20, namely at 2.86 more, leads to a significant growth of exergetic output-input ratio η_e from 0.26 to 2.33, namely at 8.96 more or at 796 %. It should also be noted (Fig. 3) that the highest increase in exergetic output-input ratio is by the temperature difference between supplied and indoor air Δt_S = 6.0 °C. So, with the aim of energy saving, the chosen air conditioning system should be preferably used at higher temperature difference between supplied and indoor air, that is, for example Δt_S = 6.0 °C (at any temperature of indoor air) that will make possible to gain the highest exergetic output-input ratio η_e, which means to gain the most advantageous economical variant of exploitation of chosen air conditioning system.

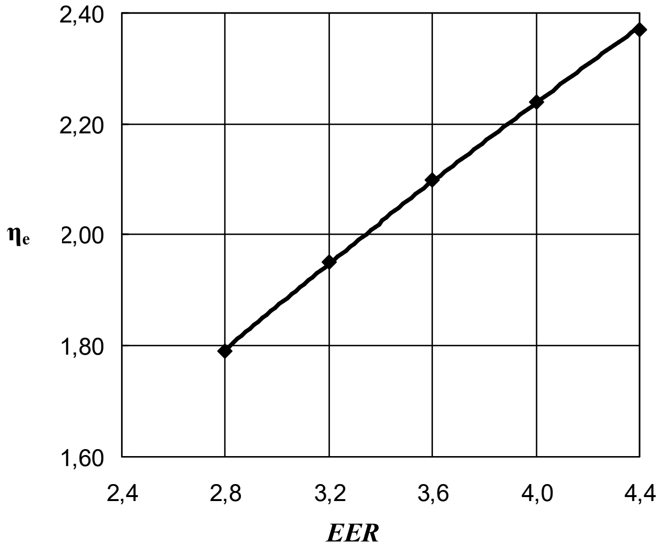

Fig. 4. The dependence of exergetic output-input ratio η_e
of the central straight flow air conditioning system for cleanroom on coefficient
of transformation *EER* of refrigerating machine:
G_S = 10000 kg/hr; Δt_S = 5.0 °C; t_{in} = 28 °C; Δt_S / t_{in} = 0.18; t_{out} = 38 °C;
φ_{in} = 50 %; φ_{out} = 30%; φ_S = 66 %

The dependence of exergetic output-input ratio η_e of the central straight flow air conditioning system for cleanroom on coefficient of transformation *EER* of refrigerating machine can be presented as such an exponent dependence:

$$\eta_e = 0.945 \cdot EER^{0.62} . \quad (8)$$

Maximum error of calculations by the equation (8) is 0.4 %.

So, on the basis of dependence (Fig. 4, equation (8)) the value of exponent β, which is in our case 0.62, was found.

Analyzing the obtained research data on Fig. 4, the following conclusions can be reached. The general increase of coefficient of transformation *EER* of refrigerating machine from 2.8 to 4.4, namely at 1.57 more, leads to a significant growth of exergetic output-input ratio η_e from 1.79 to 2.37, namely at 1.32 more or at 32 %. It should also be noted (Fig. 4) that the highest increase in exergetic output-input ratio is by coefficient of transformation of refrigerating machine *EER* = 4.4. So, with the aim of energy saving, the chosen air conditioning system should be preferably used by higher coefficient of transformation of refrigerating machine, that is, for example *EER* = 4.4, that will make possible to gain the highest exergetic output-input ratio $\eta_{å}$, which means to gain the most advantageous economical variant of exploitation of chosen air conditioning system.

Finally, we determine that coefficient C = 27.1, and get a general exponent dependence for the chosen central straight flow air conditioning system for cleanroom:

$$\eta_e = 27.1 \cdot (\Delta t_S / t_{in})^{1.94} \cdot EER^{0.62} . \tag{9}$$

Maximum error of calculations by the equation (9) is 4.4 %.

So the exergetic analysis of the central straight flow air conditioning system for cleanroom was performed on the innovation mathematical research model created by the authors which provided the opportunity to thoroughly estimate the dependence of exergetic output-input ratio of this ACS depending on different factors, which have influence on its work.

4. Conclusions

Innovation research model of the central straight flow air conditioning system for cleanroom was used, which gave opportunity to estimate its energy effectiveness by the virtue of exergetic output-input ratio depending on different factors, which have influence on its work. General exponent dependence of exergetic output-input ratio η_e of central straight flow air conditioning system for cleanroom on the ratio of temperature difference between indoor and supplied air to the temperature of indoor air $\Delta t_S / t_{in}$ and coefficient of transformation EER of refrigeration machine was found. It was shown that with the aim of energy saving the chosen air conditioning system should be preferably used by higher temperature difference between indoor and supplied air, for example Δt_S =6.0 ºC (at any temperature of indoor air) and by higher coefficient of transformation of refrigeration machine, for example EER = 4.4, that will give opportunity to gain the highest exergetic output-input ratio η_e, which means we will gain energy saving regime of exploitation of the chosen air conditioning system.

References

[1] Fedotov A. E. 2003. Cleanrooms. Second edd. – Moscow : ASINKOM, 2003. – 576 p. (in Russian)

[2] Khaiakava I. 1990. Cleanrooms. Transl. from Jap. – Moscow: Mir. – 456 p. (in Russian)

[3] Whyte W. 2002. Cleanrooms technology. Basics of designing, testing and maintenance. Transl. from Engl. – Moscow : Cleanroom. – 304 p. (in Russian)

[4] Whyte W. 2004. Cleanrooms designing. Transl. from Engl. – Moscow : Cleanroom. – 360 p. (in Russian)

[5] GOST ISO 14644-1. Cleanrooms and Associated Controlled Environments. Part 1. Classification of air cleanliness by particle concentration (in Russian)

[6] Sokolov E. Y. 1981. Energy bases of heat and cooling processes transformations: Tutorial for higher educational institutions. – 2nd edd. / E. Y. Sokolov, V. M. Brodianskii. – Moscow : Energoizdat. – 320 p. (in Russian)

[7] Shargut Y. 1968. Exergy / Y. Shargut, R. Petela. – Moscow : Energiia. – 280 p. (in Russian)

[8] Exergetic design of engineering systems. 1991 : handbook / [V. M. Brodianskii, G. P. Verhivker, Y. Y. Karchev et ali]; under editorship of A. A. Dolinskii, V. M. Brodianskii; Inst. Of engineering thermal physics AS USSR. – Kyiv : Nauk. dumka. – 360 p. (in Russian)

[9] Brodianskii V. M. 1973. Exergetic method of thermo-dynamical analysis / V. M. Brodianskii. – Moscow : Energiia. – 296 p. (in Russian)

[10] Ber G. D. 1977. Engineering thermodynamics / G.D. Ber. – Moscow : Mir. – 518 p. (in Russian)

[11] Bogoslovskii V.N. 1985. Air conditioning and refrigeration: Tutorial for higher educational institutions / V. N. Bogoslovskii, O. Y. Kokorin, L. V. Petrov. – Stroiizdat. – 367 p. (in Russian)

[12] Prokhorov V. I. 1981. Method for calculation of exergy of humid air flow / V. I. Prokhorov, S. M. Shilkloper // Kholodil'naia tekhnika. – No. 9. – P. 37–41 (in Russian)

[13] Shilkloper S. M. 1982. Exergetic analysis of systems for microclimate control and energy supply / S. M. Shilkloper, S. I. Zhadin // Stroitel'stvo i arkhitektura. Ser. 9. – Iss. 4. – P. 18–27 (in Russian)

[14] SNiP 2.04.05–86. 1987. Heating, ventilation and conditioning. – Moscow: TsITP Gosstroia USSR. – 64 p. (in Russian)

[15] Iantovskii E. I. 1988. Flows of energy and exergy / E. I. Iantovskii. – Moscow: Nauka. – 144 p. (in Russian)

[16] Bes T. 1962. Exergy in processes of heating, climate control and drying / T. Bes // Industrial Power Engineering. – 10, No. 11. – P. 388–392 (in Polish)

[17] Labay Volodymyr. 2000. Exergetic efficiency of central conditioners / Volodymyr Labay, Taras Ivanukh // V Rzeszow-Lviv-Koszyce Scientific Conference "Up-to-date problems of civil and environmental engineering". Proceedings of Rzeszow University of Technology "Civil and environmental engineering". – Vol. 32, Part 2: Environmental engineering. – Rzeszów: Rzeszow University of Technology. – pp. 229–235 (in Ukrainian)

[18] Labay V. 2014. Innovation model for energy effective investigations of air conditioning systems for cleanrooms / V. Labay, D. Harasym // ECONTECHMOD – Lublin-Rzeszow. – Vol. 3, No. 1. – P. 47–52.

[19] Labay V.Y. 2014. Investigation of exergetic efficiency of cleanrooms air conditioning systems / V.Y. Labay, D.I. Harasym // Scientific and Engineering Journal “Refrigeration engineering and technology”, No. 4 (150). – Odessa: ONAKHT. – P. 47–53. (in Ukrainian)

[20] Harasym D. I. 2014. Assessment of cleanrooms air conditioning systems by the Grassmann diagram / D. I. Harasym, V. Y. Labay // Civil engineering, mateial science, mechanical engineering. Proc. Iss. 76. – Dnipropetrovsk: PHASA. – P. 95–100 (in Ukrainian)

Аналіз ексергоефективності та шляхів енергозбереження системи кондиціювання повітря чистого приміщення

Дмитро Гарасим, Володимир Лабай

Національний університет “Львівська політехніка”, вул. С. Бандери, 12, Львів, 79013, Україна

Анотація

У сучасних технологіях, пов’язаних з перетворенням енергії, а саме у системах кондиціювання повітря, важливе місце займають обладнання і процеси, об’єктивну оцінку ступеня енергетичної досконалості яких можна встановити тільки на основі аналізу їх ексергоефективності. Отже, зменшення витрат енергії, споживаної системами кондиціювання повітря, диктує необхідність їх оптимізації, що найповніше може бути досягнуто на основі ексергетичного аналізу, який враховує не тільки кількість, але й якість витраченої енергії. Нині вартісні оцінки не можуть слугувати єдиною мірою ефективності енергетичного обладнання, які переробляють енергоресурси. Ексергія є фізичним, а не економічним критерієм і визначає незалежність цього параметра від кон’юнктурних коливань цін. Водночас вартісні показники не дають змоги здійснити довгострокове прогнозування. Визначати мінімум необхідно не грошовими витратами, а витратами ексергії на одиницю виданої теплоти. Непридатність тільки грошових критеріїв очевидна. У статті наведено аналіз ексергоефективності центральної прямотечійної системи кондиціювання повітря (СКП) чистого приміщення, отримано на її інноваційній математичній дослідницькій моделі залежно від різних факторів, що впливають на її роботу, та запропоновані шляхи енергозбереження для цієї СКП. Встановлено, що найбільший вплив на ексергетичний ККД вибраної системи кондиціювання мають різниця температур між внутрішнім і припливним повітрям у приміщенні, температура внутрішнього повітря, залежна від температури зовнішнього повітря, та коефіцієнт трансформації EER прийнятої холодильної машини СКП.

Ключові слова: системи кондиціювання повітря; чисті приміщення; ексергетичний баланс; ексергетична ефективність; енергозбереження.