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INVESTIGATIONS OF THE INNOVATION MODEL OF EXERGY EFFECTIVENESS OF AIR CONDITIONING SYSTEM FOR OPERATING CLEANROOMS

Innovation mathematical research model of the existing central straight flow air conditioning system for operating cleanrooms with the aim of computer estimation of its energy effectiveness by virtue exergetic output-input ratio depending on different factors, which have influence on its work was presented in this article. The dependence of exergetic output-input ratio $\eta_{å}$ of the existing air conditioning system on temperature difference between the inside and the supplied air $\Delta t_S = t_{in} - t_S$ was defined thanks to this model. It was noticed that at the given temperature difference between indoor and supplied air Δt_S the change in temperature of outdoor air practically don't causes the change in exergetic output-input ratio. The chosen air conditioning system should be preferably used at higher temperature difference between indoor and supplied air, for example at Δt_S = 9.0°C, that will give the opportunity to gain the highest exergetic output-input ratio, which means to gain the most advantageous economical variant of exploitation of the chosen air conditioning system.

Keywords: exergy balance, air conditioning systems, cleanrooms, exergy efficiency

1. Introduction

Nowadays during the exploitation of energy technological systems (ETS), which include air conditioning systems (ACS), to ensure carrying out a certain technology the question of economy of fuel and energy resources, is of prime importance.

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That's why the question about ETS, that can organically combine and complement the requirements of technology and power engineering, is raised now.

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 its thermodynamic analysis.

The simplest method of thermodynamic analysis is energy one based on the law of energy conservation. It allows us to estimate absolute and relative energy losses, to reveal equipment and processes with the highest losses. However, this method equates to one another values of all kinds of energy, thermal energy in particular, that is wrong from the position of the second law of thermodynamics, because any kind of energy can be completely converted into the thermal one, the reverse process at the same time is accompanied by the unavoidable losses.

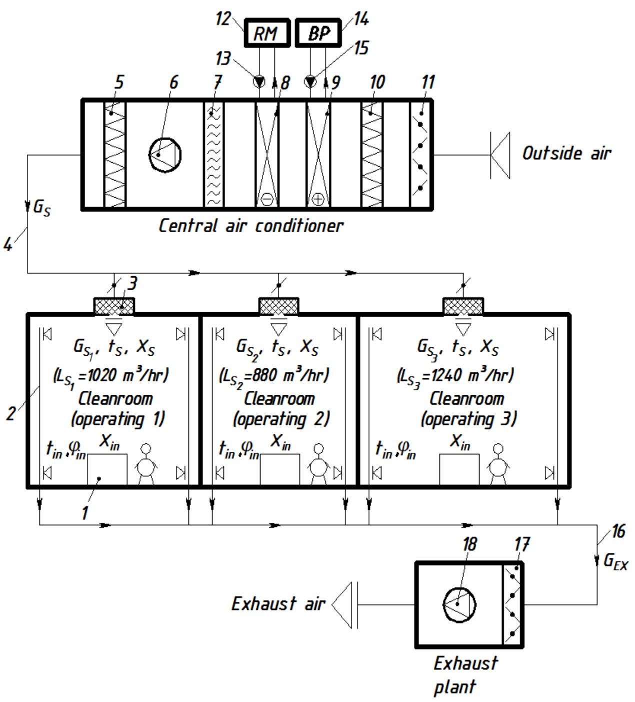
Under the influence of these requirements the exergy method of analysis has been designed in the last decades [1, 4, 5, 15, 18]. This method was described at the works of R. Clausisus, J. Gibbs, G. Guye, A. Stodola, Y. Shargut and R. Petela. Its main idea is to introduce, along with the common, fundamental concept of energy, the additional indicator – exergy, which allows considering the fact, that the energy depending on external conditions may have a different value for practical use.

The calculations of balances and different characteristics of energy technological systems, air conditioning systems in particular, taking into account the exergy enables in the easiest and clearest way to solve many scientific and technical problems. They help to remove frequent errors that can be founded and are associated with ignoring the qualitative side of transformation.

The feature of the central straight flow air conditioning system is that the starting substance, which is processed in it – is the outside air, the parameters of which may vary as the temperature and the relative humidity (moisture content and specific enthalpy, relatively). As the result of the work the air conditioning system provides required temperature and humidity of supplied and indoor air depending on parameters of outside air. Reducing the cost of energy, consumed by air conditioning systems, dictates the need of its optimization, which can be fully achieved by virtue of exergetic analysis, that takes into account not only the quantity, but also the quality of energy spent [2, 3, 5, 11-17].

2. Description of the object that is analyzed and of innovation research model

The aim of air conditioning is to keep up the certain parameters of air in some limited space (in this case, in the operating cleanrooms). Usually temperature t_{in} and relative humidity φ_{in} of air are regulated, but in cleanrooms such parameter as concentration of dust particles x_{in} in air is also regulated [6-8, 19-20].



1 – technological equipment;
2 – air exhaust channels in cleanrooms;
3 – filters of supplied air to the room;
4 – supplied air duct;
5 – filter of supplied air in a conditioner;
6 – fan unit of conditioner;
7 – drop catcher of conditioner;
8 – air cooler of conditioner;
9 – air heater of conditioner;
10 – filter of outside air in conditioner;
11 – valve of outdoor air in conditioner;
12 – refrigerating machine RM;
13 – pump of cold water of RM;
14 – boiler plant BP;
15 – pump of warm water;
16 – exhaust air duct;
17 – valve of exhaust air;
18 – fan unit of the exhaust plant

Fig. 1. Basic scheme of the implemented central straight flow air conditioning system for operating cleanrooms

Rys. 1. Podstawowy schemat zastosowania systemu klimatyzacji z centralnym bezpośrednim przepływem obsługujący czyste pomieszczenia

Let's consider the implemented by the authors' central straight flow air conditioning system for operating cleanrooms, which is shown in Fig. 1. The work of such system depends on the dominant environmental conditions, i.e. on temperature and moisture content of outdoor air. So, at a warm period of year outdoor air is taken via central conditioner through the valve 11, is cleaned in the filter 10, then passes through the air heater 9, is politropical cooled and drained in the air cooler 8, is separated in the drops catcher 7 and after that this air is supplied through the air supply filter of conditioner 5 and air supply filters 3 at the entrance of operating cleanrooms via the fan unit 6. The exhaust air of the operating cleanrooms is removed from their top and lower zones via extraction system through its valve 17 by the extractor fan 18.

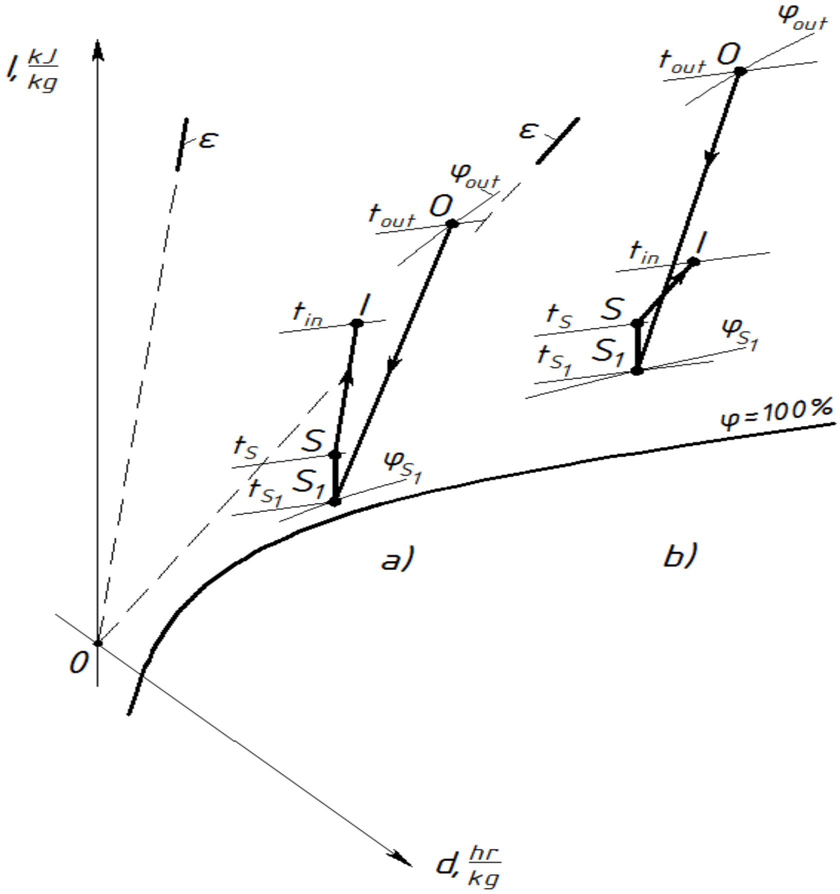


Fig. 2. The image of the process of changing the state of moist air in the implemented central straight flow air conditioning system in a warm period of year on the I-d – diagram a) for conditions of the research – t_{out} = 30; 32°C; t_{in} = 25; 26°C; Δt_S = 9; 7°C; b) for conditions of the research – t_{out} = 35; 38; 40°C; t_{in} = 27; 28; 29°C; Δt_S = 6; 5; 4°C: OS_1 – the process of polytropic treatment (cooling and drying) of air $G_{out} = G_S$ in the air cooler; S_1S – the process of supplied air G_S heating by 1°C in a fan and duct; SI – the process of excess heat and moisture assimilation in a cleanroom by supplied air via the conditioner

Rys. 2. Obraz procesu zmiany stanu wilgotności powietrza w zastosowanym systemoe klimatyzacji z centralnym bezpośrenim przepływem w ciepłym okresie roku wykres - I-d, a) dla warunków badań - t_{out}= 30; 32°C; t_{in} = 25; 26°C; Δt_S = 9; 7°C; b) dla warunków badań - t_{out} = 35; 38; 40°C; t_{in} = 27; 28; 29°C; Δt_S = 6; 5; 4°C: OS_1 - proces z polytropiczny (chłodzenie i suszenie) powietrza $G_{out} = G_S$ w chłodnicy powietrza; S_1S - proces dostarczania powietrza G_S ogrzewanego o 1°C w wiatraku i kanale; SI - proces dodatkowego ogrzewania i pochłaniania wilgoci w czystym pomieszczeniu przez dostarczanie powietrza G_S przez klimatyzator

Let's consider the work of this air conditioning system at a warm period of year, when temperature $t_{\text{in}} < t_{\text{out}}$. Figure 2 shows in a coordinate system I, d the sequence of change of air parameters, which passes through the different equipment of the implemented central straight flow air conditioning system for operating cleanrooms at a warm period of year at different parameters of outdoor air. In the researches mass productivity of the air conditioning system G = 4300 kg/hr, that was counted by the necessary multiplicity of air exchange, parameters of outdoor air varied within: temperature t_{out} = 30-40°C; relative humidity φ_{out} = 44-36% (in accordance, moisture content and specific enthalpy d_{out} = 11.7-16.8 hr/kg; I_{out} = 60.1-83.4 kJ/kg), barometric pressure p_{out} = 1010 hPA; parameters of indoor air, accordingly – t_{in} = 25-29°C; φ_{in} = 54-64% (in accordance, d_{in} = 10.8-16.3 hr/kg; I_{in} = 52.6-70.8 kJ/kg); temperature difference between inside and supplied air depending on excess heat in the cleanroom and also on temperature of outdoor air $\Delta t_{\text{S}} = t_{\text{in}} - t_{\text{S}}$ = 9.0-4.0°C; slope coefficient of excess heat and moisture assimilation in the cleanroom by the supplied air via air conditioner ε = 27058-9711 kJ/kg; initial temperature of the coolant (40% propylene glycol solution) for the air cooler t_{w_I} = 9.5-15.5°C.

The sequences of changes, that occur with the moist air, which passes through the various equipment of the implemented air conditioning system, are shown in Fig. 2. Construction on the *I-d* – diagram was made in accordance to [3]. Parameters of air in the characteristic points of the process (Fig. 2) were determined by the adopted values of parameters for outdoor air and were calculated on the proposed mathematical model by the known analytical dependences for moist air.

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

$$Q_{\text{C}} = G_{\text{S}} \cdot \left(I_{\text{O}} - I_{\text{S}_1}\right) \times 0.278 \text{ , W,} \tag{1}$$

where I_{O} and I_{S_1} – specific enthalpy of air at the corresponding points of processes of change the state of moist air in the implemented straight flow air condition system at a warm period of year on the *I-d* – diagram (Table 1), kJ/kg.

The aim of this work was to create innovation mathematical research model of the implemented central straight flow air conditioning system for operating cleanrooms with the aim of computer estimation of its energy effectiveness by virtue exergetic output-input ratio depending on different factors, which have influence on its work. Material, heat (energy) and exergy balances of the system were made up in this model, which took into account all the possible variants of its work in real conditions.

The concept of exergetic output-input ratio was used for the rational excellence assessment of the air conditioning system, 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 [1, 2, 4, 5, 9-16, 18]:

$$\eta_{\text{å}} = \frac{E_{\text{out}}}{E_{\text{in}}} . \tag{2}$$

The exergetic output-input ratio, which characterizes the effectiveness of work of the implemented central straight flow air conditioning system for operating cleanrooms at a warm period of year, was defined by the equation:

$$\eta_{\text{e}} = \frac{E_{\text{out}}}{E_{\text{in}}} = \frac{\Delta E_{\text{SI}}}{\Delta E_{\text{O}S_1} + \Delta E_{\text{S}_1\text{S}} + \Delta E_{\text{SI}} + \Delta E_{\text{exh}} + N_{\text{use}}^{\text{sup.fan}} + N_{\text{use}}^{\text{exh.fan}} + N_{\text{use}}^{\text{RM}}} , \tag{3}$$

where $E_{\text{out}} = \Delta E_{\text{SI}} = E_{\text{S}} - E_{\text{I}}$ – exergy reduction of conditioned air in the surgery operating cleanrooms (usefully used exergy), W; Å_{S} and Å_{I} – in accordance, exergy of supplied and indoor air in the cleanrooms, W; $\Delta E_{\text{O}S_1} = E_{\text{S}_1} - E_{\text{O}}$ – increasing of exergy of air in the air cooler of conditioner, W; E_{S_1} and E_{O} – in accordance, exergy of air which is processed (outdoor air) at the exit and entrance of the air cooler of conditioner, W; $\Delta E_{\text{S}_1\text{S}} = \text{Å}_{\text{S}_1} - \text{Å}_{\text{S}}$ – exergy reduction of air during its transportation in the supplied ducts and fan of air conditioning system, W; E_{S_1} and E_{S} – in accordance, exergy of air at the entrance of supplied fan of conditioner and at the exit of supplied ducts to the cleanrooms, W; $\Delta E_{\text{exh}} = E_{\text{I}} - E_{\text{O}}$ – exergy losses with exhaust conditioned air from the cleanrooms, W; $N_{\text{use}}^{\text{sup.fan}}$ – consumed power via the supplied fan of the conditioner, W; $N_{\text{use}}^{\text{ex.fan}}$ – consumed power via the fan of the exhaust plant, W; $N_{\text{use}}^{\text{RM}}$ – consumed power via the refrigerating machine for the central conditioner, W.

The values, included in the equation (3) for the determination of exergetic output-input ratio of implemented ACS at a warm period of year, were defined as follows:

$$\Delta E_{\text{SI}} = G_{\text{S}} \cdot (e_{\text{S}} - e_{\text{in}}) \times 0.278 \text{, W,} \tag{4}$$

where e_{S} i e_{in} – in accordance, specific exergy of supplied and indoor air in the cleanrooms (Table 1), kJ/kg;

$$\Delta E_{\text{O}S_1} = G_{\text{S}} \cdot (e_{\text{S1}} - e_{\text{out}}) \times 0.278 \text{, W,} \tag{5}$$

where e_{out} i e_{S_1} – in accordance, specific exergy of outdoor air and of cooled and drained air in the air cooler, (Table 1), kJ/kg;

$$\Delta E_{\mathrm{S_1}S} = G_{\mathrm{S}} \cdot \left(e_{\mathrm{S1}} - e_{\mathrm{S}}\right) \times 0.278\text{, W,} \tag{6}$$

where e_{S} – specific exergy at the exit of supplied ducts in the cleanrooms (Table 1), kJ/kg;

$$\Delta E_{\mathrm{exh}} = G_{\mathrm{S}} \cdot \left(e_{\mathrm{in}} - e_{\mathrm{out}}\right) \times 0.278\text{, W.} \tag{7}$$

By the mentioned equations the exergetic output-input ratio η_{e} for the implemented air conditioning system at a warm period of year were calculated and an appropriate conclusions were made.

Specific exergy of moist air were determined as follows.

Specific exergy of moist air at a certain points of the processes which characterize the work of the implemented central straight flow air conditioning system were determined by the following equations:

$$e = e_{\mathrm{ph}} + e_{\mathrm{ch}}\text{, kJ/kg,} \tag{8}$$

where e_{ph} and e_{ch} – in accordance, specific physical and chemical exergy in relation to parameters of outdoor air (environment);

$$e_{\mathrm{ph}} = \left(\bar{c}_{\mathrm{d.air}} + \bar{c}_{\mathrm{vap}} \cdot \frac{d}{1000}\right) \cdot \left(T - T_0 - T_0 \cdot \ln\frac{T}{T_0}\right)\text{, kJ/kg,} \tag{9}$$

$\bar{c}_{\mathrm{d.air}}$ = 1.005 kJ/(kg·K) and $\bar{c}_{\mathrm{vap}}$ = 1.86 kJ/(kg·K) – in accordance, average specific heat capacity (at constant pressure) of dry air water vapor;

T_0 i T – in accordance, absolute temperature of outdoor air (environment) and of air at a certain point of the process, K ($273 + t = T$);

$$e_{\mathrm{ch}} = T_0 \cdot \left[\left(R_{\mathrm{d.air}} + R_{\mathrm{vap}} \cdot \frac{d}{1000}\right) \cdot \ln\frac{622 + d_0}{622 + d} + R_{\mathrm{vap}} \cdot \frac{d}{1000} \cdot \ln\frac{d}{d_0}\right]\text{, kJ/kg,} \tag{10}$$

$R_{\mathrm{d.air}}$ = 0.287 kJ/(kg·K) i R_{vap} = 0.462 kJ/(kg·K) – in accordance, gas constant of dry air and of water vapor;

d_0 i d – in accordance, moisture content of outdoor air (environment) and of air at a certain points of the process, hr/kg.

The results of calculations of specific exergy of moist air at a certain points of the processes, which characterize the work of the implemented central straight flow air conditioning system, are summarized in Table 1.

It should be noted that we didn't take into account exergy losses which are related to losses of aerodynamic pressure of air flow, which are slight and could be ignored, and also losses into environment. The parameters, which characterize the state of the air at all points of the processes for the given air conditioning system, are summarized in Table 1.

Table 1. Parameters of points, which describe the state of moist air during the work of the implemented air conditioning system

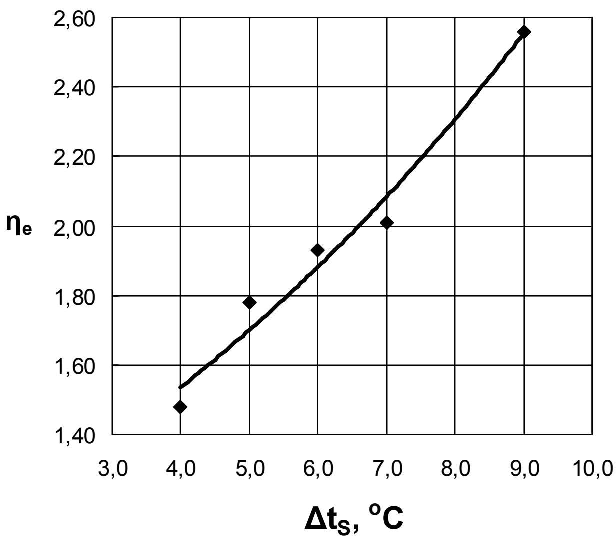
Tabela 1. Parametry punktów opisujących stan wilgotności powietrza podczas pracy zastosowanego systemu klimatyzacji

Points on the I-d – diagram	**Temperature** t, **°C**	**Specific enthalpy** I, **kJ/kg**	**Moisture content** d, **hr/kg**	**Relative humidity** φ, **%**	**Specific exergy** e, **kJ/kg**
O	30-40	60.1-83.4	11.7-16.8	44-36	0.0-0.0
S_1	15-24	42.1-64.7	10.5-15.8	98-84	0.4020-0.4421
S	16-25	42.6-65.4	10.5-15.8	92-79	0.3505-0.3882
I	25-29	52.6-70.8	10.8-16.3	54-64	0.0475-0.2059

3. Results of the research work

Substituting the received values by the research calculations in equation (3), we calculated the meaning of exergetic output-input ratio η_e for the implemented air conditioning system depending on outdoor temperature conditions, namely: temperature t_{out} and relative humidity φ_{out} of outdoor air, and showed them in the form of dependence of exergetic output-input ratio on temperature difference between indoor and supplied air Δt_S at a Fig. 3 and on this basis were making conclusions.

Analyzing the obtained research data at a Fig. 3 the following conclusions can be reached. General increase of temperature difference between indoor and supplied air Δt_S from 4.0 to 9.0°C, namely in 2.25 times more, leads to a significant growth of exergetic output-input ratio $\eta_å$ from 1.48 to 2.56, namely in 1.73 times more or at 73%. Therewith, the average speed of change of exergetic output-input ratio $\eta_å$ is $\Delta\eta_e / \Delta(\Delta t_S)$ = 0.216 1/°C. We noticed that at the given temperature difference between indoor and supplied air Δt_S the change in temperature of outdoor air t_{out} practically don't causes the change in exergetic output-input ratio $\eta_å$. It means that at a certain temperature difference between indoor and supplied air Δt_S we can ignore the effect of temperature of outdoor air t_{out}. So the chosen air conditioning system should be preferably used at higher temperature difference between indoor and supplied air, for example at Δt_S = 9.0°C, that will give the opportunity to gain the highest exergetic output-input ratio $\eta_å$, which means to gain the most advantageous economical variant of exploitation of the chosen air conditioning system.



point 1
$\Delta t_S = 4.0°C$ ($t_{out} = 40°C$; $\varphi_{out} = 36\%$; $t_{in} = 29°C$; $\varphi_{in} = 64\%$; $\varphi_S = 79\%$);
point 2
$\Delta t_S = 5.0°C$ ($t_{out} = 38°C$; $\varphi_{out} = 38\%$; $t_{in} = 28°C$; $\varphi_{in} = 64\%$; $\varphi_S = 84\%$);
point 3
$\Delta t_S = 6.0°C$ ($t_{out} = 35°C$; $\varphi_{out} = 40\%$; $t_{in} = 27°C$; $\varphi_{in} = 60\%$; $\varphi_S = 84\%$);
point 4
$\Delta t_S = 7.0°C$ ($t_{out} = 32°C$; $\varphi_{out} = 42\%$; $t_{in} = 26°C$; $\varphi_{in} = 55\%$; $\varphi_S = 82\%$);
point 5
$\Delta t_S = 9.0°C$ ($t_{out} = 30°C$; $\varphi_{out} = 44\%$; $t_{in} = 25°C$; $\varphi_{in} = 54\%$; $\varphi_S = 92\%$);

Fig. 3. The dependence of exergetic output-input ratio of the implemented central straight flow air conditioning system for operating cleanrooms at a warm period of year on temperature difference between indoor and supplied air Δt_S

Rys. 3. Zależność egzegetycznego stosunku wyjście-wejście zastosowanego centralnego systemu klimatyzacji z bezpośrednim przepływem dla obsługi czystych pomieszczeń w ciepłym okresie roku w stosunku do różnicy temperatur powietrza wewnętrznego i dostarczanego Δt_S.

The dependence (Fig. 3) has been obtained in a form of analytical equation for temperature difference between indoor and supplied air $\Delta t_S = 4.0\text{-}9.0°C$:

$$\eta_e = 1.023 \cdot \exp(0.102 \cdot \Delta t_S) \tag{11}$$

Maximum error of calculations by the equation (11) is 4.3%.

So the exergetic analysis of the implemented central straight flow air conditioning system for operating cleanrooms at a warm period of year, which was performed on created by the authors innovation mathematical research model, provided the opportunity to thoroughly estimate the dependence of exergetic output-input ratio $\eta_{å}$ of this system on temperature difference between indoor and supplied air Δt_S at different temperatures of outdoor air t_{out}.

4. Conclusions

Innovation mathematical research model of the implemented central straight flow air conditioning system for operating cleanrooms at a warm period of year was described in this article, which gives opportunity to make computer estimation of its energy efficiency by virtue of exergetic output-input ratio depending on different factors, which have influence on its work. The dependence of exergetic output-input ratio η_e of this air conditioning system on temperature difference between indoor and supplied air Δt_S at various temperatures of outdoor air t_{out} was presented. It was noticed that at the given temperature difference between indoor and supplied air Δt_S the change in temperature of outdoor air t_{out} practically don't causes the change in exergetic output-input ratio η_e. It is shown that the chosen air conditioning system should be preferably used at higher temperature difference between indoor and supplied air, for example at Δt_S = 9.0°C, that will give the opportunity to gain the highest exergetic output-input ratio η_e, which means to gain the most advantageous economical variant of exploitation of the chosen air conditioning system.

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ANALIZA SKUTECZNOŚCI EGZERGII INNOWACYJNEGO MODELU SYSTEMU KLIMATYZACJI DLA CZYSTYCH POMIESZCZEŃ

S t r e s z c z e n i e

W artykule zaprezentowano innowacyjny matematyczny model do analizy istniejącego systemu klimatyzacji z centralnym bezpośrednim przepływem wykorzystany w czystych pomieszczeniach, w celu określenie metodą numeryczną jego skuteczności energetycznej oraz zyski egzergi w stosunku wyjście-wejście zależny od różnych czynników. Na podstawie tego modelu określono zależność stosunku η_e egzergii wyjścia-wejścia istniejącego systemu klimatyzacji od różnicy temperatur $\Delta t_S = t_{in} - t_S$ powietrza wewnętrznego i dostarczonego. Zauważono, że przy danej różnicy temperatur Δt_S wewnątrz pokojowy i powietrza dostarczanego, zmiana temperatury powietrza zewnętrznego nie powoduje zmiany w egzergii stosunku wyjście-wejście. Przedstawiony system klimatyzacji daje najlepsze efekty przy zastosowaniu go w wyższych różnicach temperatur wewnętrznej i powietrza dostarczanego, na przykład przy Δt_S = 9.0 ° C, system klimatyzacji uzyskuje najwyższą egzergię stosunku wyjście - wejście, który oznacza najbardziej korzystny ekonomicznie wariant eksploatacji wybranego systemu klimatyzacji.

Słowa kluczowe: równowaga egzergii, system klimatyzacji, pomieszczenia czyste, sprawność energetyczna

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