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Innovation model for energy effective investigations of air conditioning systems for cleanrooms

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Abstract. Innovation mathematical research model of the central straight flow air conditioning system for cleanroom in order to computer's estimation its energy effective by virtue of 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 for chosen air conditioning system on parameters of external air was defined thanks to this model.

Key words: exergy balance, air conditioning systems, cleanrooms, energy efficiency.

INTRODUCTION

In modern technologies, which are related to energy transformation, namely in the air conditioning systems, important place are occupied by objects creation and improvement of which requires the use of innovation thermodynamics. Classic apparatus of this science is often insufficient to solve new tasks; it is necessary not only to its further development, but its combination with the elements of a systematic approach and economy.

Under the influence of these requirements the exergy method has been designed in the last decades [6-10]. 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 technical systems, air conditioning systems in particular, taking into account the exergy enables the easiest and clearest way to solve many scientific and technical problems. They help to remove frequent errors that are found and associated with ignoring the qualitative side of change.

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). Supplied and indoor air reached necessary temperature and humidity as a result of the air conditioning systems work. Outside air is the external environment for air conditioning systems, the parameters of which may vary depending on location and time. That's why outside air is taken as external environment – as its dry part and its water vapor, which is in the air. Reducing the cost of energy, consumed by air conditioning systems (ACS), 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 the energy spent [7, 8, 11-14, 16, 18, 19].

DESCRIPTION OF OBJECT THAT IS ANALYZED AND 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 cleanroom). Usually temperature t_{in} and relative humidity φ_{in} of air are regulated, but in cleanrooms a concentration of dust particles x_{in} in air is also regulated [1-5].

As an example, we can consider central straight flow air conditioning system for cleanroom, which is shown in Fig. 1. The work of such system depends on the dominant environmental (external) conditions, i.e. on temperature and moisture content in 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 in the air washer 9, is heated in the air heater of second heat 10 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.

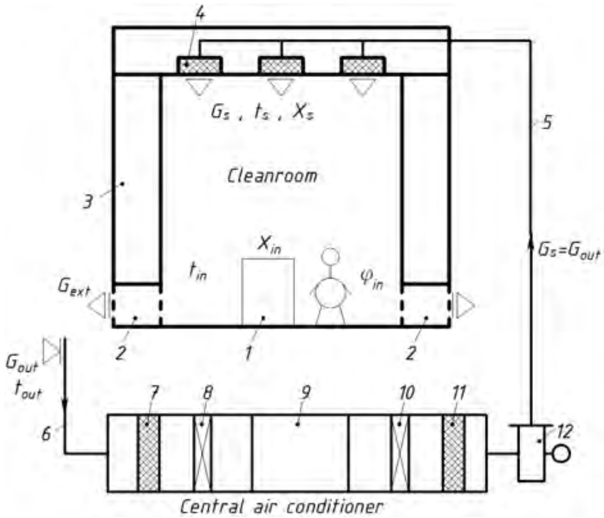

Fig. 1. Basic scheme of the central straight flow air conditioning system for cleanroom: 1 – technological equipment, 2 – air exhaust channels, 3 – gateway premises, 4 – filters of air supply to the room, 5 – air supply duct, 6 – air-intake shaft, 7 – filter of external air, 8 – air heater of first heat,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

Lets consider the work of this air conditioning system in the warm period of year, when moisture content $d_{in} < d_{out}$ and temperature $t_{in} < t_{out}$. 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 = 10000 kg/hr is accepted, that was counted by the number of dust particles, parameters of outdoor air varied within: t_{out} = 26-42°C; d_{out} = 9,3-15,9 gr/kg (specific enthalpy: I_{out} = 49,8-83,2 kJ/kg; relative humidity: φ_{out} = 27-54 %, accordingly), barometric pressure: p_{out} = 1010 hPa, parameters of indoor air, respectively: t_{in} = 23-30°C; φ_{in} = 50 % (d_{in} = 8,8-13,4 gr/kg; I_{in} = 45,5-64,4 kJ/kg, accordingly); temperature difference between the inside and the supplied air depending on excess heat in the cleanroom: $\Delta t_S = t_{in} - t_S$ = 1,5-6,0°C; slope coefficient of excess heat and moisture assimilation in the cleanroom by the supplied air via air conditioner: ε = 9946-16858 kJ/kg; water temperature (coolant temperature) for the air washer initial: t_{w_I} = 7,0-14,9°C; final: t_{w_F} = 9,8-17,3°C; heat transfer agent (water) temperature for the heater of second heat initial: t_{heat} = 70°C; final: t_{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 year was defined by the equation:

$$Q_C = G_S \cdot \left(I_O - I_K\right) \times 0,278 \text{, W,} \tag{1}$$

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

$$Q_2 = G_S \cdot \left(I_{S_1} - I_K\right) \times 0,278 \text{, W,} \tag{2}$$

where: $I_O = I_{out}$, I_K and 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 (Table 1), kJ/kg.

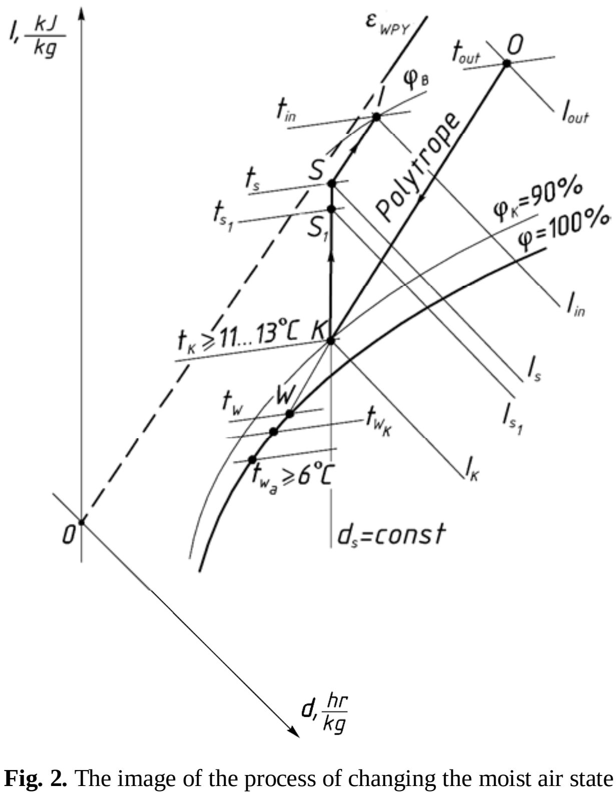

Fig. 2. The image of the process of changing the moist air state in the straight flow air conditioning system in a warm period of year on the *I-d* – diagram: *OK* – the process of polytropic air treatment (cooling and drying) $G_{a.w} = G_S$ in the air washer; *KS*$_1$ – the air G_S heating process in the air heater of second heat; *S*$_1$*S* – 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 aim of this work was to create the innovation mathematical research model of the central straight flow air conditioning system for cleanroom in order to computer's estimation its energy effective by virtue of exergetic output-input ratio in dependence from different factors, which have influence on its work. 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 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 [6-10, 12, 13, 15-20]:

$$\eta_e = \frac{E_{out}}{E_{in}}. \quad (3)$$

The exergetic output-input ratio, which characterizes the efficiency of the central straight flow air conditioning system work for cleanroom in the warm period of year, was defined by the equation:

$$\eta_e = \frac{\Delta E_{SI}}{\Delta E_{a.w} + \Delta E_{heat} + N_{use}^{c.w} + N_{use}^{h.w} + N_{use}^{fan} + N_{use}^{RM}}, \quad (4)$$

where: $\Delta E_{SI} = E_S - E_I$ – exergy reduction of conditioned air in the cleanroom, W; E_S and $E_I = E_{in}$ – exergy of supplied and indoor air in the cleanroom, in accordance, W; $\Delta E_{a.w} = E_{w_I} - E_{w_F}$ – exergy change of water in the air washer (exergy growth of air in the air washer, relatively), W; E_{w_I} and E_{w_F} – exergy of water in the air washer with it initial and final temperatures, relatively, W; $\Delta E_{heat} = E_{heat} - E_{rev}$ – exergy change of heat transfer agent (hot water) in the air heater of second heat (exergy reduction of air in the air heater of second heat, relatively), W; E_{heat} and E_{rev} – heat carrier exergy in a feeding and reverse jets of the air heater of second heat, relatively, W; $N_{use}^{c.w}$ – consumed power via the pump of cold water for the air washer, W; $N_{use}^{h.w}$ – consumed power via the pump of how water for the air heater of second heat, W; N_{use}^{fan} – consumed power via the fans engine of the accepted central air conditioner, W; N_{use}^{RM} – consumed power via refrigeration machine for the central conditioner, W.

The values, included in the equation (4), were defined as follows:

$$\Delta E_{SI} = G_S \cdot (e_S - e_I) \times 0,278, \text{ W}, \quad (5)$$

where: e_S and $e_I = e_{in}$ – specific exergy of supplied and indoor air in the cleanroom (Table 1), relatively, kJ/kg;

$$\Delta E_{a.w} = G_{a.w} \cdot (e_K - e_O) \times 0,278, \text{ W}, \quad (6)$$

where: $e_O = e_{out}$ and e_K – specific exergy of outdoor air and cooled and drained air in the air washer (Table 1), relatively, kJ/kg;

$$\Delta E_{heat} = G_S \cdot (e_K - e_{S_1}) \times 0,278, \text{ W}, \quad (7)$$

where: e_{S_1} – specific exergy of air, that was heated in the air heater of second heat (Table 1), kJ/kg.

If it is necessary to find a specific exergy of water under a certain absolute temperature T_w, then it can be defined as follows:

$$e_w = c_w \cdot \left(T_w - T_0 - T_0 \cdot \ln \frac{T_w}{T_0} \right), \text{ kJ/kg}, \quad (8)$$

where: $c_w = 4.19$ kJ/(kg·K) – water specific heat (at constant pressure).

If it is necessary, the exergy change in a thermal process can be determined by the equation:

$$\Delta E = Q \cdot \left(1 - \frac{T_0}{\overline{T}} \right), \text{ W}, \quad (9)$$

where: Q – heat flow, which takes place in the thermal process, W; $\overline{T} = 273 + 0.5(t_1 + t_2)$ – absolute average temperature in a thermal process, K; t_1 and t_2 – initial and final temperature in the thermal process, °C.

The power consumption of fans electromotor for air transport was determined by the equation:

$$N_{use}^{fan} = N_{set}^{fan} \cdot \eta_{fan}, \text{ W}, \quad (10)$$

where: N_{set}^{fan} – installed fans motor capacity of the adopted central air conditioner, W; η_{fan} – fans output-input ratio. $N_{use}^{c.w}$ and $N_{use}^{h.w}$ were similarly determined.

Taking the coefficient of the energy class of refrigerator E.E.R. = 2,8, its power consumption were determined by the equation:

$$N_{use}^{RM} = \frac{Q_C}{E.E.R.}, \text{ W}. \quad (11)$$

The specific enthalpies of moist air were calculated as follows:

Specific exergy of moist air at certain points of the processes, that characterize the work of the central straight flow air conditioning system, are determined by the following equations:

$$e = e_{ph} + e_{ch}, \text{ kJ/kg}, \quad (12)$$

where: e_{ph} and e_{ch} – specific physical and chemical exergy in relation to outdoor air (environment), respectively:

$$e_{ph} = \left(\overline{c}_{d.air} + \overline{c}_{vap} \cdot \frac{d}{1000} \right) \cdot \left(T - T_0 - T_0 \cdot \ln \frac{T}{T_0} \right), \text{ kJ/kg}, \quad (13)$$

where: $\overline{c}_{d.air}$ = 1,005 kJ/(kg·K) and $\overline{c}_{vap}$ = 1,86 kJ/(kg·K) – average specific heat (at constant pressure) of dry air and water vapor, respectively;

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

$$e_{ch} = T_0 \cdot \left[\left(R_{d.air} + R_{vap} \cdot \frac{d}{1000} \right) \cdot \ln \frac{622 + d_0}{622 + d} + R_{vap} \cdot \frac{d}{1000} \cdot \ln \frac{d}{d_0} \right], \text{ kJ/kg}, \quad (14)$$

$R_{d.air}$ = 0,287 kJ/(кг·К) and R_{vap} = 0,462 kJ/(кг·К) – the gas constant of dry air and water vapor, respectively; d_0 and d – the moisture content of the outdoor air (environment) and air in a certain point in the process, respectively, hr/kg.

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

It should be noted that we have not been taken into account exergy loss, associated with the loss of aerodynamic pressure of air stream, which are relatively small and can be neglected, losses in the environment also are not accounted, besides it is accepted that the process of humidification in the air washer occurs polytropic. The parameters, which characterize the state of the air at all points of the processes for the given central air conditioning system, are summarized in Table 1.

RESULTS OF THE RESEARCH WORK

Substituting the obtained estimated values in a equation (4), the values of exergetic output-input ratio η_e for the specified air conditioning system depending on external temperature conditions were calculated, namely: temperature t_{out} and moisture content d_{out} of outdoor air served them in the form of appropriate dependencies on the Fig. 3 and on this basis analysis was made.

Analyzing the obtained research data on a Fig. 3 the following conclusions can be reached.

The general increase of outdoor air t_{out} from 26 to 42°C, namely in 1.62 times more, according to the general increase of moisture content of outdoor air d_{out} in a row 3 from 10.3 to 14.9 hr/kg, namely in 1.45 times more, leads to a significant growth of exergetic output-input ratio value η_e from 0.26 to 2.30, namely in 8.85 times more or at 785 %. It should also be noted (Fig. 3), that the increase of moisture content of outdoor air d_{out} from 13.9 to 15.9 hr/kg, namely in 1.14 times more, when the temperature of outdoor air t_{out} = 42 °C, leads to a slight decrease of exergetic output-input ratio η_e from 2.44 to 2.18, namely in 1.12 times less or at 12 %, which can be ignored if necessary. So the chosen air conditioning system should be preferably used at higher temperatures of outdoor air, that is, for example t_{out} = 42 °C, 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.

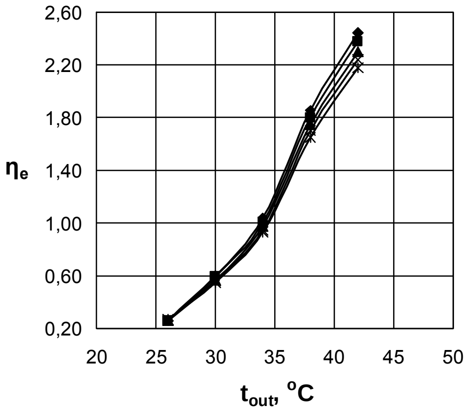

Fig. 3. The dependence of exergetic output-input ratio η_e of the central straight flow air conditioning system for cleanroom on temperature t_{out} and moisture content of outdoor air: row 1 ♦ – d_{out} = 9,3-13,9 hr/kg; row 2 ■ – 9,8-14,4; row 3 ▲ – 10,3-14,9; row 4 ✕ – 10,8-15,4; row 5 ✱ – 11,3-15,9

It should be noted (Fig. 3), that there is one law of change the exergetic output-input ratio, when the temperature of outdoor air t_{out} is from 26 to 34°C, and when the temperature of outdoor air t_{out} is from 34 to 42°C – another. This is because the indoor temperature of the air in the clean room is taken differently to outside air temperature t_{out} = 30°C and above it. That's why lets consider these changes separately.

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

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	26-42	49,8-83,2	9,3-15,9	27-54	0,0-0,0
I	23-30	45,5-64,4	8,8-13,4	50-50	0,0173-0,2707
S	21,5-24,0	43,5-57,3	8,6-13,0	54-69	0,0385-0,5904
S_1	20,5-23,0	42,4-56,2	8,6-13,0	57-74	0,0559-0,6542
K	13,3-19,7	35,1-52,8	8,6-13,0	90-90	0,2870-0,8925

So, the general increase of outdoor air t_{out} from 26 to 34°C, namely in 1.31 times more, according to the general increase of moisture content of outdoor air d_{out} in a row 3 from 10.3 to 12.7 hr/kg, namely in 1.23 times more, leads to a significant growth of exergetic output-input ratio value η_e from 0.26 to 0.98, namely in 3.77 times more or at 277 %. It should also be noted (Fig. 3) that the increase of moisture content of outdoor air d_{out} from 11.7 to 13.7 hr/kg, namely in 1.17 times more, when the temperature of outdoor air t_{out} = 34 °C, leads to a slight decrease of exergetic output-input ratio η_e from 1.04 to 0.93, namely in 1.12 times less or at 12 %, which can be ignored if necessary. At the same time the average rate of change of exergetic output-input ratio η_e at this initial section $\Delta\eta_e / \Delta t_{out}$ is 0.09 1/°C.

Respectively the general increase of outdoor air t_{out} from 34 to 42°C, namely in 1.24 times more, according to the general increase of moisture content of outdoor air d_{out} in a row 3 from 12.7 to 14.9 hr/kg, namely in 1.17 times more, leads to a significant growth of exergetic output-input ratio value η_e from 0.98 to 2.30, namely in 2.35 times more or at 135 %. At the same time, the average rate of change of exergetic output-input ratio η_e at this section $\Delta\eta_e / \Delta t_{out}$ is 0.165 1/°C, that is in 1.24 times more, namely in 84 %, than at initial section.

The dependencies (Fig. 3) we have obtained in the form of analytical equations for temperatures of outdoor air t_{out} = 26-34°C:

$$\eta_e = 0,1841 \cdot t_{out} + 0,1990 \cdot d_{out} - 0,0075 \cdot t_{out} \cdot d_{out} - 4,5645$$

$$\eta_e = 0,1841 \cdot t_{out} + 0,1990 \cdot d_{out} - 0,0075 \cdot t_{out} \cdot d_{out} - 4,5645, \qquad (15)$$

and for t_{out} = 34-42°C:

$$\eta_e = 0,3187 \cdot t_{out} + 0,2585 \cdot d_{out} - 0,0093 \cdot t_{out} \cdot d_{out} - 9,1135. \qquad (16)$$

The maximum error of calculations by the equation (15) is 16.4 % and by the equation (16) – 1.7 %.

So the exergetic analysis of the central straight flow air conditioning system for cleanroom, which was performed on innovation mathematical research model, which was created by authors, provided the opportunity to thoroughly estimate the dependence of exergetic output-input ratio η_e this system on temperature t_{out} and moisture content d_{out} of outdoor air.

CONCLUSIONS

Innovation mathematical research model of the central straight flow air conditioning system for cleanroom in order to computer's estimation its energy effective by virtue of exergetic output-input ratio depending on different factors, which have influence on its work, was described in this article. The dependence of exergetic output-input ratio for chosen air conditioning system η_e on temperature t_{out} and moisture content d_{out} of outdoor air was presented. It is shown that the chosen air conditioning system should be preferably used at higher outdoor air temperatures, namely, for example t_{out} = 42°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 chosen air conditioning system.

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