SYSTEM SAFETY: HUMAN - TECHNICAL FACILITY – ENVIRONMENT
6(1), 2024, pp. 289-298

REDUCING THE ENERGY CONSUMPTION OF THE AIR CONDITIONING SYSTEM IN CLEAN ROOMS USING THE IDLE MODE

DOI: 10.2478/czoto-2024-0031
Received: 28/11/2024
Accepted: 20/12/2024

Alexander Čaja[1] – *orcid id: 0000-0002-5040-4471, e-mail: alexander.caja@fstroj.uniza.sk*
Andrej Kapjor[1] – *orcid id: 0000-0002-5951-9312, e-mail: andrej.kapjor@fstroj.uniza.sk*
Martin Vantúch[1] – *orcid id: 0000-0002-2610-7434, e-mail: martin.vantuch@fstroj.uniza.sk*
Nikola Čajová Kantová[2] – *orcid id: 0000-0002-7529-036X, e-mail: nikola.cajovakantova@uniza.sk*
[1]Department of Power Engineering, Faculty of Mechanical Engineering, University of Žilina, Univerzitna 8215/1, 010 26 Zilina Slovakia
[2]Research Centre, University of Žilina, Univerzitna 8215/1, 010 26 Zilina Slovakia

Abstract: Cleanrooms, essential in various industries, demand strictly controlled environmental conditions, which significantly impact building energy consumption. High-efficiency air handling systems are crucial in these environments to ensure a continuous supply of clean air and removal of contaminants. However, the substantial energy requirements of these systems pose a significant challenge to sustainable operations. This study focuses on an innovative approach to reduce the energy consumption of air handling systems in cleanrooms through the optimization of laminar airflow. Laminar flow, renowned for its high particle capture efficiency, is commonly employed in cleanrooms. Nevertheless, its operation is energy-intensive. It was proposed a novel concept involving the segmentation of the laminar flow field and the variable adjustment of airflow velocity within individual segments. This approach enables more flexible adaptation of airflow to specific operational requirements while reducing the overall volume of air that needs to be filtered and conditioned. Experimental results demonstrate that the proposed system can significantly reduce the energy consumption of the air handling system without compromising cleanliness. Compared to conventional systems, energy consumption was reduced by up to 60% in winter and 56% in summer, while simultaneously improving air quality in the critical area above the operating table.
Keywords: cleanrooms, laminar flow, energy efficiency, air handling, optimization

1. INTRODUCTION

Cleanrooms are designated environments characterized by a controlled low level of airborne particles. Maintaining their proper function is crucial for industries such as healthcare or precision manufacturing. Elevated particle concentrations can compromise product quality and performance, as well as pose risks by carrying viruses and bacteria, thereby increasing the likelihood of contamination, especially in surgical settings. The fundamental approach to maintaining low particle levels is to introduce a substantial

volume of filtered air, which effectively lowers the particle concentration in the room to meet required standards (Mičko et al., 2022).
Forced ventilation systems have stringent performance requirements, as they not only manage the transport and filtration of air into the ventilated space but also regulate the temperature and humidity of the air, leading to significant energy consumption. Research indicates that cleanrooms account for 50–70% of the electricity used by HVAC (Heating, Ventilation, and Air Conditioning) systems to maintain proper operation (Khoo et al., 2012; Kircher et al., 2010). A study conducted in Taiwan highlights the use of advanced HVAC systems that, through air recirculation, consume only around 35% of electricity, representing approximately 7.4% of total energy consumption (Hu and Chuah, 2003).
Reducing energy consumption in cleanrooms is a complex issue that requires an interdisciplinary approach. By combining appropriate technologies, optimizing operations, and ensuring regular maintenance, significant energy savings can be achieved while reducing environmental impacts.
The energy demands for cooling, dehumidification, preheating, and/or humidification of outdoor air in cleanroom HVAC systems are substantial, accounting for 30% to 65% of the total thermal energy required to maintain the clean environment within the room. Given these high energy requirements, cost-effective measures to reduce energy consumption can have a significant impact on per-unit production costs (Tsao et al., 2008).
In this work, we focused on reducing the energy load by utilizing the idle mode. In professional literature and the standard *STN EN ISO 14644*, the idle mode is characterized by a reduced volumetric flow rate of the equipment. During this mode, it is necessary to maintain pressure differences between the ventilated rooms and to maintain the state of the thermal and humidity microclimate within the set range. Both the temperature and humidity of the air in the room must be at a level such that, if necessary to use the operating room, the required thermal and humidity microclimate is ensured within a maximum of 20 minutes.
The idle mode is not precisely defined by law. Based on experience from healthcare facilities, it is recommended to reduce the volumetric flow rate to half the performance compared to the operating mode. At the same time, however, the temperature in the room should not exceed the difference from the desired temperature by more than 2°C.
This paper deals with the reduction of energy consumption in HVAC systems of cleanrooms through the implementation of an idle mode. Six variants were designed, differing in laminar flow field segmentation, internal equipment, and laboratory occupancy. The idle mode of a prototype laminar flow field was compared to that of a conventional one, evaluating the effectiveness of maintaining low particle concentration above the operating table and energy consumption.

2. METHODOLOGY OF RESEARCH
To evaluate various operating conditions during idle mode, multiple CFD simulation models were created to investigate the impact of reduced volumetric flow rate on maintaining low particle concentration above the operating table. A total of six models were developed, with the first three, A1, A2, and A3, based on a single variant and differing only in internal equipment and laboratory occupancy (Fig. 1). The patient and instrument tables were removed from the geometric model. The medical staff was reduced to a single person, assumed to be wearing regular clothing and emitting particles at a high rate. The

remaining three models, B1, B2, and B3, differed from models A only in the segmentation of the laminar flow field.

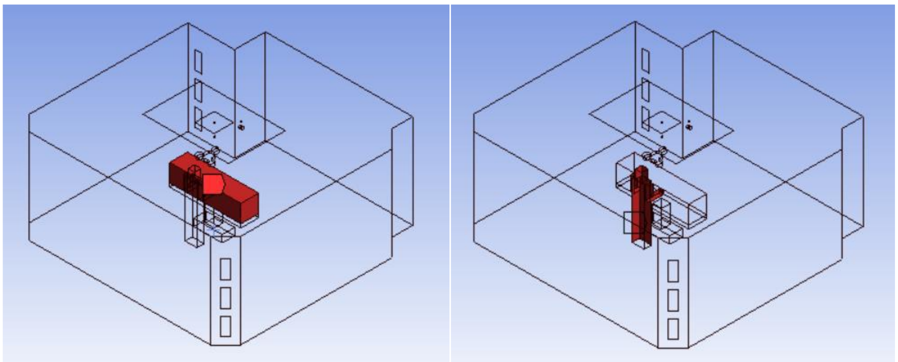

Fig. 1. CFD simulation model A_X – (left) reference volume of VOT (volume above the operating table), (right) paramedic as a source of particle emission

2.1. CFD simulation boundary conditions for the idle mode

Boundary conditions, computational and mathematical models, and surface temperatures of the structures are the same. The only differing boundary conditions relate to the volumetric flow rates of the HVAC system and the amount of particles emitted by personnel and influenced by HVAC system operation. The mass flow rate of particles from personnel was calculated according to equation (1). (Kapjor et al., 2012)

$$\dot{m}_p = \frac{\rho_p . n_h . \dot{n}_p . V_p}{60} \; [kg.s^{-1}] \qquad (1)$$

Where the $\dot{m}_p$ is mass flow of particles from personnel, [kg.s^{-1}], ρ_p is density of air particles [kg.m^{-3}], n_h is number of people in the operating room [-], $\dot{n}_p$ is number of particles emitted by one person [min^{-1}] and V_p is volume of particles [m^3].

The value of particles emitted by one person is 350,000 min^{-1}. With a particle size of 0.5 μm, the resulting mass flow rate of particles from a medical staff member is 6.87 e-16. This value is the same for all idle mode variants from A_1 to B_3. To express the amount of particles emitted by the HVAC system, it is first necessary to know the volumetric flow rates from the laminar field, which are determined by the outlet velocity. The velocity and pressure boundary conditions for variants A_1 to A_3 are expressed in Table 1. For variants B_1 to B_3, the parameters for individual segments of the laminar field are expressed in Table 2. During the idle mode, zero flow through the SUP 3 segment was considered. The cleanliness of the environment above the operating table will be ensured only by segments SUP 1 and SUP 2. The boundary conditions at the exhaust outlets are given in Table 3.

Table 1
CFD simulation boundary conditions for variants A_1 to A_3

Variant	v_{SUP} [m.s^{-1}]	V_{SUP} [m^{3}h^{-1}]	V_{ETA} [m^{3}h^{-1}]	v_{ETA} [m.s^{-1}]	p_{ETA} [Pa]
A_1	0,2	2555	639	1,7742	1,7974
A_2	0,15	1916	479	1,3307	1,0110
A_3	0,1	1277	319	0,8871	0,4493

Table 2
CFD simulation boundary conditions for variants B_1 to B_3 for air supply

Variant	$v_{SUP\ 1}$ [m.s^{-1}]	$V_{SUP\ 1}$ [m^{3}h^{-1}]	$v_{SUP\ 2}$ [m.s^{-1}]	$V_{SUP\ 2}$ [m^{3}h^{-1}]	$v_{SUP\ 3}$ [m.s^{-1}]	$V_{SUP\ 3}$ [m^{3}h^{-1}]	$V_{SUP\ \Sigma}$ [m^{3}h^{-1}]
B_1	0,1	292	0,20	193	0	0	485
B_2	0,1	292	0,15	145	0	0	436
B_3	0,1	292	0,10	96	0	0	388

Table 3
CFD simulation boundary conditions for variants B_1 to B_3 for air extraction

Variant	V_{ETA} [m^{3}h^{-1}]	v_{ETA} [m.s^{-1}]	p_{ETA} [Pa]
B_1	121	0,3365	0,0647
B_2	109	0,3030	0,0524
B_3	97	0,2695	0,0415

With a known flow rate, it is possible to express the amount of particles emitted by the HVAC system function, as shown in Table 4.

Table 4
Mass flow of particles emitted by the HVAC system and personnel for variants A_1 to B_3

Variant	V_{SUP} [m^{3}h^{-1}]	$\dot{m}_{HVAC}$ [kg.s^{-1}]	$\dot{m}_P$ [kg.s^{-1}]
A_1	2 555	1,25 e-16	6,87 e-16
A_2	1 916	9,40 e-17	6,87 e-16
A_3	1 277	6,26 e-16	6,87 e-16
B_1	485	2,37 e-17	6,87 e-16
B_2	436	2,14 e-17	6,87 e-16
B_3	388	1,90 e-17	6,87 e-16

3. RESULTS

The amount of power required to ensure the functional operation of the HVAC system during the standby mode can be determined as the sum of the power required to drive the HVAC unit fans, the power required for cooling and dehumidification in the summer months and the power required for heating and humidification in the winter months according to equation (2). (Kapjor et al., 2012)

$$P_{HVAC} = P_{FAN} + P_{ATA} + P_{AHA}\ [W] \tag{2}$$

Where the P_{HVAC} is power required for the function of the HVAC system [W], P_{FAN} is power required for the HVAC unit fans [W], P_{ATA} is power required for adjusting the supply air temperature [W] and P_{AHA} is power required for adjusting the supply air humidity [W]. The power input of the P_{FAN} fans was experimentally measured for variants A_1 to A_3. The volume flow of variants B_1 to B_3 was too low for the range of the frequency converter in the HVAC unit of the experimental clean room laboratory and was therefore estimated proportionally according to the manufacturer's performance characteristics for a lower fan series. The power input for the fan drive according to the selected variant and the frequency of the converter are expressed in Table 5.

Table 5
CFD simulation boundary conditions for variants A_1 to B_3

Variant	v_{SUP} [m.s^{-1}]	f_{SUP} [Hz]	V_{ETA} [m^{3}h^{-1}]	F_{ETA} [Hz]	P_{FUN} [W]
A_1	2 555	37	2 555	33	1 050
A_2	1 916	28	1 916	25	560
A_3	1 277	18	1 277	16	390
B_1	485	-	485	-	200
B_2	436	-	436	-	175
B_3	388	-	388	-	150

The power input value for adjusting the temperature of the supply air depends on the technology used to heat or cool the air. In this case, the air conditioning system is equipped with a condensing unit. The power input of the condensing unit is then dependent on the power required for the thermal treatment of the air supplied by the HVAC system and on the EER factor for cooling, or the COP factor for heating the air. The power input required for heating the air after recuperation is expressed by equation (3), and for cooling the air by equation (4). (Kapjor et al., 2012)

$$P_{ATA} = \frac{Q_H}{f_{COP}}\ [W] \tag{3}$$

$$P_{ATA} = \frac{Q_C}{f_{EER}}\ [W] \tag{4}$$

Where the $Q_{H/C}$ is power required for heat treatment of the supply air by the HVAC system [W], f_{COP} is heating coefficient [-] and f_{EER} is cooling coefficient [-].
The f_{COP} and f_{EER} factors are commonly given in the technical data sheets of heat pumps and condensing units. In this case, an external design temperature of -15°C is considered in winter. At this temperature, the f_{COP} factor is = 2.95. If only electric heaters are used as a heat source, the f_{COP} factor is = 1. In the summer months, the design temperature is +32°C and the f_{EER} factor is = 2.72. The power required for the thermal treatment of the air supplied by the HVAC system can be calculated using equation (5). (Kapjor et al., 2012)

$$Q_{H/C} = \frac{\dot{V}.\,|(t_{rec} - t_{dat})|.\,c.\,\rho}{3600}\ [W] \tag{5}$$

Where the $\dot{V}$ is volumetric flow rate of the HVAC system [m^3.h^{-1}], t_{rec} is air temperature after recovery [°C], t_{dat} – desired air temperature in the room [°C], c is specific heat capacity of the supply air [J.kg^{-1}.K^{-1}] and ρ is density of the supply air [kg.m^{-3}].
The power input required for adjusting the temperature of the supply air is expressed in Table 6. The power input values for adjusting the humidity of the supply air P_{AHA} for individual variants are given in Table 7.

Table 6
Power input values for supply air temperature adjustment A_1 to B_3

Variant	Heating				Cooling	
	Electric heater		Heat pump		Heat pump	
	Q_{H1} [W]	P_{ATA1} [W]	Q_{H2} [W]	P_{ATA2} [W]	Q_C [W]	P_{ATA3} [W]
A_1	5 909	5 909	5 909	2 003	2 874	1 057
A_2	4 431	4 431	4 431	1 502	2 156	793
A_3	2 953	2 953	2 953	1 001	1 436	528
B_1	1 216	1 216	1 216	412	546	201
B_2	1 008	1 008	1 008	342	491	181
B_3	897	897	897	304	437	161

Table 7
Mass flow of particles emitted by the HVAC system and personnel for variants A_1 to B_3

Variant	V_{SUP} [m^3h^{-1}]	P_{AHA} [W]
A_1	2 555	4 921
A_2	1 916	3 691
A_3	1 277	2 460
B_1	485	934
B_2	436	840
B_3	388	747

The sums of the power required for the operation of the P_{HVAC} HVAC system during the setback mode and the percentage change in the electrical power requirement compared to variant A_3 are given for the different variants in Table 8 to Table 10. The influence of the volume flow on the power requirement for heating the air with an electric heater together with humidification is shown in Fig. 2. The influence of using a heat pump for heating the air in combination with humidification is shown in Fig. 3., Fig. 4 shows the total power requirement of the HVAC system for cooling the air using a heat pump. Humidification is not necessary in the summer months.

Table 8
Total power requirement for heating the supply air with an electric heater and humidifying the air during the setback mode

Variant	P_{FAN} [W]	P_{ATA1} [W]	P_{AHA} [W]	P_{HVAC} [W]	Change in power compared to variant A_3 [%]
A_1	1 050	5 909	4 921	11 880	105%
A_2	560	4 431	3 691	8 682	50%
A_3	390	2 953	2 460	5 803	0%
B_1	200	1 216	934	2 350	-60%
B_2	175	1 008	840	2 023	-65%
B_3	150	897	747	1 794	-69%

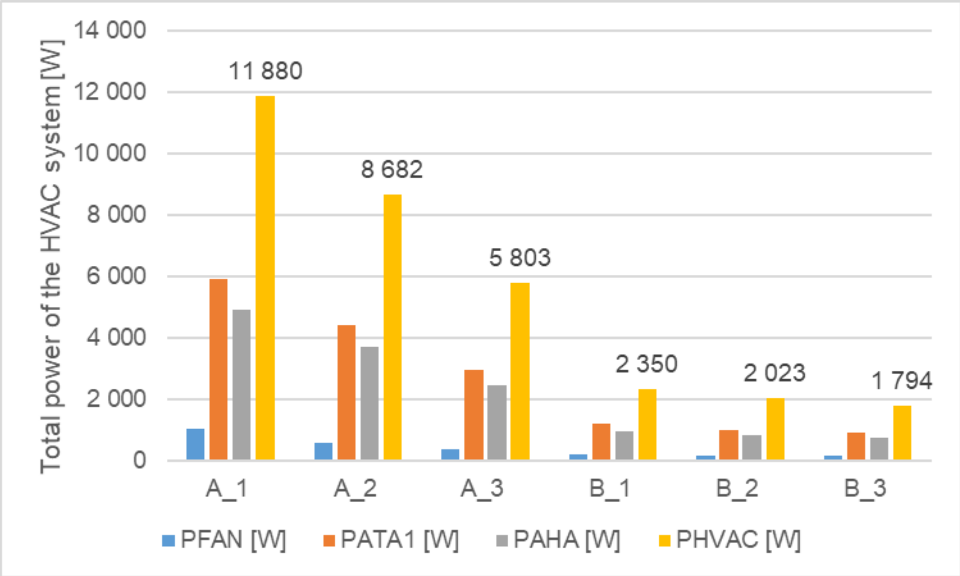

Fig. 2. Total power required for heating the supply air with an electric heater and humidifying the air during setback mode

Fig. 2 shows the total power required to heat the supply air using an electric heater and humidify the air during setback mode for different variants (A_1 - B_3).
Variants A_1, A_2 and A_3 represent different levels of airflow and fan power, while variants B_1, B_2 and B_3 represent segmented fin arrays with different airflows.
Fig. 2 shows that variants B achieve significantly lower energy consumption compared to variants A, with variant B_3 having the lowest energy consumption.

Table 9
Total power required for heating the supply air with a heat pump and humidifying the air during setback mode

Variant	P_{FAN} [W]	P_{ATA2} [W]	P_{AHA} [W]	P_{HVAC} [W]	Change in power compared to variant A_3 [%]
A_1	1 050	2 003	4 921	7 974	107%
A_2	560	1 502	3 691	5 753	49%
A_3	390	1 001	2 460	3 851	0%
B_1	200	412	934	1 546	-60%
B_2	175	342	840	1 357	-65%
B_3	150	304	747	1 201	-69%

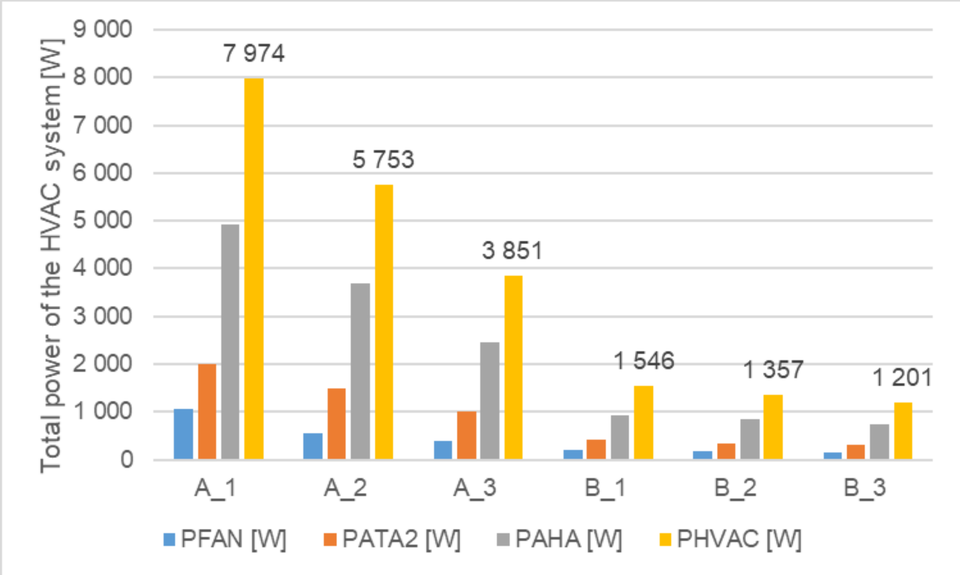

Fig. 3. Total power required for heating the supply air with a heat pump and humidifying the air during setback mode

Fig. 3 shows the total power required to heat the supply air using the heat pump and humidify the air during setback mode for different variants.
Similar to the previous graph, variants B achieve lower energy consumption compared to variants A, with variant B_3 having the lowest energy consumption.

Table 10
Total power required for cooling the supply air by the heat pump during setback mode

Variant	P_{FAN} **[W]**	P_{ATA3} **[W]**	P_{HVAC} **[W]**	**Change in power compared to variant A_3 [%]**
A_1	1 050	1 057	2 107	130%
A_2	560	793	1353	47%
A_3	390	528	918	0%
B_1	200	201	401	-56%
B_2	175	181	356	-61%
B_3	150	161	311	-66%

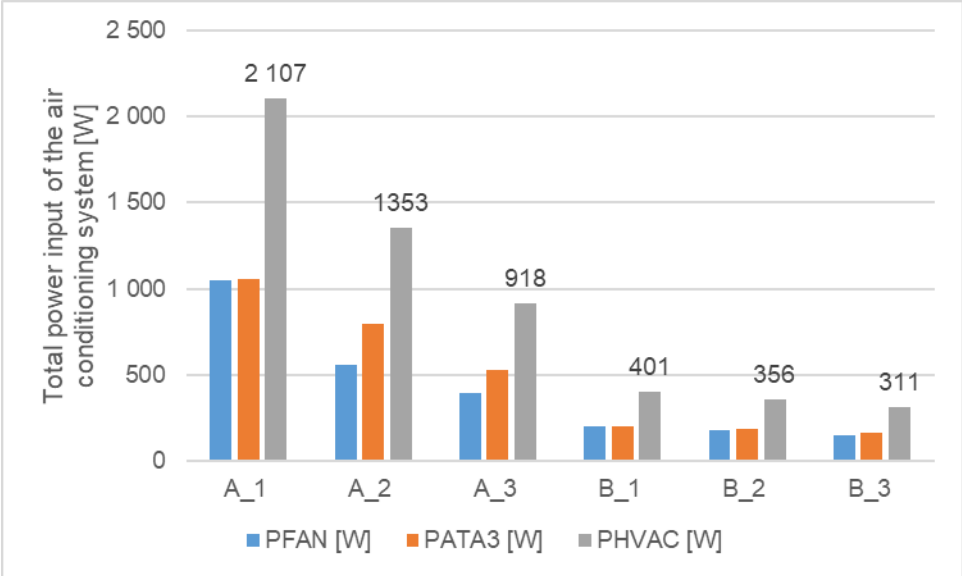

Fig. 4. Total power required for cooling the supply air by the heat pump during setback mode

Fig. 4 shows the total power required to cool the supply air using the heat pump during setback mode for different variants.
Fig. 4 shows that the B variants again achieve lower energy consumption compared to the A-variants, with variant B_3 having the lowest energy consumption.

4. CONCLUSION

This article explores innovative methods to enhance energy efficiency in cleanroom environments. Cleanrooms, essential in industries like healthcare and precision manufacturing, require stringent control over environmental conditions, which leads to high energy consumption, particularly in HVAC systems.
The study introduces a novel approach to reduce energy consumption by optimizing laminar airflow. Laminar flow, known for its high particle capture efficiency, is energy-intensive. The proposed method involves segmenting the laminar flow field and variably adjusting airflow velocity within these segments. This allows for more flexible adaptation to specific operational needs, reducing the overall volume of air that needs to be filtered and conditioned.
Experimental results from the study show significant energy savings without compromising cleanliness. The energy consumption of the air handling system was reduced by up to 60% in winter and 56% in summer compared to conventional systems. This reduction is achieved through the implementation of an idle mode, characterized by a reduced volumetric flow rate while maintaining necessary pressure differences and thermal and humidity conditions.
The research methodology involved creating multiple CFD simulation models to evaluate the impact of reduced volumetric flow rates on maintaining low particle concentration above the operating table. Six models were developed, differing in internal equipment and laboratory occupancy, as well as in the segmentation of the laminar flow field. The study found that the idle mode could effectively maintain low particle concentrations while significantly reducing energy consumption.
In conclusion, the article demonstrates that optimizing laminar airflow and implementing an idle mode can lead to substantial energy savings in cleanroom HVAC systems. By

reducing the volumetric flow rate and segmenting the laminar flow field, it is possible to achieve a more energy-efficient operation without compromising the cleanliness and functionality of the cleanroom environment. This approach not only reduces operational costs but also contributes to environmental sustainability by lowering the energy footprint of cleanroom facilities. The findings highlight the importance of innovative engineering solutions in achieving energy efficiency and sustainability in high-demand environments like cleanrooms.

ACKNOWLEDGEMENTS

This publication has been produced with the support of the project KEGA 035/ŽU-4/2024 " Ventilation and air conditioning systems for clean rooms”.

REFERENCES

Hu, S.C., Chuah, Y.K., 2003. *Power consumption of semiconductor fabs in Taiwan*. Energy 28, 895–907, DOI: 10.1016/S0360-5442(03)00008-2

Kapjor, A., Hužvár, J. Greššák, T., 2012. Vzduchotechnika 1. Žilina: EDIS. ISBN 978-80-554-05839-1.

Khoo, C.Y., Lee, C.C., Hu, S.C., 2012. *An experimental study on the influences of air change rate and free area ratio of raised-floor on cleanroom particle concentrations*. Build Environ 48, 84–88, DOI: 10.1016/J.BUILDENV.2011.09.001

Kircher, K., Shi, X., Patil, S., Zhang, K.M., 2010. *Cleanroom energy efficiency strategies: Modeling and simulation*. Energy Build 42, 282–289, DOI: 10.1016/J.ENBUILD.2009.09.004

Mičko, P., Hečko, D., Kapjor, A., Nosek, R., Kolková, Z., Hrabovský, P., Kantová, N.Č., 2022. *Impact of the Speed of Airflow in a Cleanroom on the Degree of Air Pollution*. Applied Sciences 2022, Vol. 12, Page 2466 12, 2466, DOI: 10.3390/APP12052466

STN EN ISO 14644-1 (12 5301) 1.5.2016 | Technická norma | NORMSERVIS s.r.o. [WWW Document], n.d. URL https://eshop.normservis.sk/norma/stneniso-14644-1-1.5.2016.html (accessed 11.20.24).

Tsao, J.M., Hu, S.C., Chan, D.Y.L., Hsu, R.T.C., Lee, J.C.C., 2008. *Saving energy in the make-up air unit (MAU) for semiconductor clean rooms in subtropical areas*. Energy Build 40, 1387–1393, DOI: 10.1016/J.ENBUILD.2007.12.005