EDUCATION AND SCIENCE MINISTRY OF UKRAINE

NATIONAL AVIATION UNIVERSITY

Faculty of Aeronavigation, Electronics and Telecommunications

Department of computer integrated complexes

ADMIT TO DEFENSE

Head of the graduate department

_______ Viktor M. Sineglazov

“____” ______________ 2023y.

QUALIFICATION WORK

(EXPLANATORY NOTE)

GRADUATE OF EDUCATION AND QUALIFICATION LEVEL

“BACHELOR”

Specialty 151 "Automation and computer-integrated technologies"

Educational and professional program "Computer-integrated technological processes and production"

Theme: System for optimization of energy efficiency parameters of ventilation and air conditioning in industrial buildings

Performer: student of group FAET-404 Moskalichuk Artem Oleksandrovich

Supervisor: Senior Lecturer Dolgorukov Serhii Olehovych

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(sign)

Kyiv 2023

МІНІСТЕРСТВО ОСВІТИ І НАУКИ УКРАЇНИ

НАЦІОНАЛЬНИЙ АВІАЦІЙНИЙ УНІВЕРСИТЕТ

Факультет аеронавігації, електроніки та телекомунікацій

Кафедра авіаційних комп’ютерно-інтегрованих комплексів

ДОПУСТИТИ ДО ЗАХИСТУ

Завідувач випускової кафедри

_________ Віктор СИНЄГЛАЗОВ

“ ____ ” ____________2023 р.

КВАЛІФІКАЦІЙНА РОБОТА

(ПОЯСНЮВАЛЬНА ЗАПИСКА)

ВИПУСНИКА ОСВІТНЬОГО СТУПЕНЯ

“Бакалавр”

Спеціальність 151 «Автоматизація та комп'ютерно-інтегровані технології»

Освітньо-професійна програма «Комп’ютерно-інтегровані технологічні процеси і виробництва»

Тема: Система оптимізації параметрів енергоефективності вентиляції та кондиціонування повітря в промислових будівлях.

Виконавець: студент групи КВ-404Ба Москальчук Артем Олександрович

Керівник: старший викладач Долгоруков Сергій Олегович

Нормоконтролер: ________________Філяшкін М.К

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Київ - 2023

NATIONAL AVIATION UNIVERSITY

Faculty of aeronavigation, electronics and telecommunications

Department of Aviation Computer Integrated Complexes

Educational level: bachelor

Specialty: 151 "Automation and computer-integrated technologies"

Educational and professional programme "Computer-integrated technological processes and production"

APPROVED

Head of Department

Sineglazov V. M.

"____" ___________2023

TASK

For the student's thesis

Moskalichuk Artem Oleksandrovich

1. **Theme of project:** “System for optimization of energy efficiency parameters of ventilation and air conditioning in industrial buildings”.
2. **The term of the project:** from May 22, 2023, until June 14, 2023
3. **Output data to the project:** measuring the optimum performance of the fan and ventilation systems.
4. **Contents of the explanatory note:** 1.Preface and an overview of the literature ; 2.systems analysis of the subtle area systems for ventilation and air conditioning; 3.hvac model development and modelling approaches; 4.modelling the optimal control system of hvac operating modes based on feedback.
5. **List of required illustrative material:** tables, figures, diagrams, graphs.

6. **Planned schedule**.

№	Task	Execution term	Execution mark
1.	Getting the task	22.05.2023 – 23.05.2023	Done
2.	Formation of the purpose and main objectives of the study	23.05.2023 – 26.05.2023	Done
3.	Analysis of existing methods	26.05.2023 – 31.05.2023	Done
4.	Theoretical consideration of problem solving	28.05.2023 – 01.06.2023	Done
5.	Consideration of system analysis for building an energy consumption model of an HVAC installation,	30.05.2023 – 08.06.2023	Done
6.	Preparation of an explanatory note	10.06.2023 – 14.06.2023	Done
7.	Preparation of presentation and handouts	13.06.2023 – 14.06.2023	Done

7. **Date of task receiving:** 22 “May” 2023.

Diploma thesis supervisor ______________ Dolgorukov S. O.

(sign)

Issued task accepted ______________ Moskalichuk A. O.

(sign)

НАЦІОНАЛЬНИЙ АВІАЦІЙНИЙ УНІВЕРСИТЕТ

Факультет аеронавігації, електроніки та телекомунікацій

Кафедра авіаційних комп’ютерно-інтегрованих комплексів

Освітній ступінь: бакалавр

Спеціальність 151 «Автоматизація та комп'ютерно-інтегровані технології»

Освітньо-професійна програма «Комп’ютерно-інтегровані технологічні процеси і виробництва»

ЗАТВЕРДЖУЮ

Завідувач кафедри

_________ Віктор СИНЄГЛАЗОВ

“ ____ ” __________2023 р.

ЗАВДАННЯ

на виконання кваліфікаційної роботи студента

Москальчук Артем Олександрович

1. **Тема роботи:** «Система оптимізації параметрів енергоефективності вентиляції та кондиціонування повітря в промислових будівлях».
2. **Термін виконання роботи:** з 22 травня 2023 року по 14 червня 2023 року
3. **Вихідні дані до роботи:** вимірювання оптимальної продуктивності вентилятора та систем вентиляції.
4. **Зміст пояснювальної записки (перелік питань, що підлягають розробці):** 1.Передмова та огляд літератури; 2.Системний аналіз систем вентиляції та кондиціонування повітря тонких зон; 3.Розробка моделі вентиляція та кондиціонування та підходи до моделювання; 4.Моделювання системи оптимального керування режимами роботи вентиляція та кондиціонування на основі зворотного зв'язку.
5. **Перелік обов’язкового графічного матеріалу:** таблиці, зображення, діаграми, графіки.

6. Календарний план-графік

№ п/п	Завдання	Термін виконання	Відмітка про виконання
1	Отримання завдання	22.05.2023 – 23.05.2023	Виконано
2	Формування мети та основних завдань дослідження	23.05.2023 – 26.05.2023	Виконано
3	Аналіз існуючих методів	26.05.2023 – 31.05.2023	Виконано
4	Теоретичний розгляд рішення задач	28.05.2023 – 01.06.2023	Виконано
5	Розгляд системного аналізу для побудови моделі енергоспоживання установки ОВіК,	30.05.2023 – 08.06.2023	Виконано
6	Оформлення пояснювальної записки	10.06.2023 – 14.06.2023	Виконано
7	Підготовка презентації та роздаткового матеріалу	13.06.2023 – 14.06.2023	Виконано

7. Дата видачі завдання___“22” травня 2023 р.

Керівник: __________________Долгоруков С.О

(підпис)

Завдання прийняв до виконання ________________ Москальчук А.О

(підпис)

ABSTRACT

With the growing number of ventilation systems and increased air regulation, it is crucial to optimize the operational modes of ventilation equipment. The electricity consumption of these units is substantial, particularly because many of them are outdated and not operating in energy-efficient modes. By simulating the installation, it becomes possible to identify the most efficient modes for different types of loads. This can be achieved through methods such as forecasting power consumption and adjusting it using automation mechanisms.

The objective of the work is to create a model of the ventilation system that accurately represents its operating modes under various types of loads. This necessitates employing different methods to investigate power consumption. The research task involves identifying optimal operational modes for the system by utilizing automation (controllers) and effective management techniques.

The employed methods rely on correlation analysis, specifically utilizing the MS Excel "Data Analysis Package." This enables the identification of critical factors. Additionally, simulation approaches based on physical laws were employed, with the model's empirical data being adjusted accordingly. The software tools employed for the research include CoolPack, OpenModelica, and MATLAB.

The developed model and the corresponding recommendations for controlling the operational mode of the ventilation system can be further utilized for conducting a detailed analysis of individual HVAC components.

РЕФЕРАТ

Зі зростанням кількості вентиляційних систем та збільшенням обсягів регулювання повітрообміну дуже важливо оптимізувати режими роботи вентиляційного обладнання. Споживання електроенергії цими установками є значним, особливо тому, що багато з них є застарілими і не працюють в енергоефективних режимах. Моделювання установки дозволяє визначити найбільш ефективні режими для різних типів навантажень. Цього можна досягти за допомогою таких методів, як прогнозування енергоспоживання та його регулювання за допомогою механізмів автоматизації.

Метою роботи є створення моделі системи вентиляції, яка точно відображає режими її роботи при різних типах навантажень. Це зумовлює необхідність застосування різних методів дослідження енергоспоживання. Завдання дослідження полягає у визначенні оптимальних режимів роботи системи за допомогою засобів автоматизації (контролерів) та ефективних методів управління.

Застосовані методи базуються на кореляційному аналізі, зокрема, з використанням "Пакету аналізу даних" MS Excel. Це дозволило виявити критичні фактори. Крім того, були використані підходи моделювання, засновані на фізичних законах, з відповідним коригуванням емпіричних даних моделі. Програмні інструменти, використані для дослідження, включають CoolPack, OpenModelica та MATLAB.

Розроблена модель та відповідні рекомендації щодо керування режимом роботи вентиляційної системи можуть бути використані для проведення детального аналізу окремих компонентів системи опалення, вентиляції та кондиціонування повітря.

Table of Contents,

Basic concepts and terms

1. **Production premises** are enclosed spaces in specially designed buildings and structures in which people work constantly (in shifts) or periodically (during part of the working day).

2. **Work area** - a space in which workplaces are located permanent or non-permanent (temporary) stay of employees.

3. **Workplace** - a place of permanent or temporary of the employee in the course of labour activity.

4. **Permanent workplace** - a place where an employee is more than 50% of the working time or more than 2 hours continuously. If the work is carried out in different points of the of the working area, the entire area is considered a permanent workplace.

5. **A non-permanent workplace** is a place where an employee is less than 50% of the working time or less than 2 hours continuously.

6. The **warm season** is a time of year when the average daily ambient temperature is greater than +10 degrees Celsius.

7. The term "**cold season**" refers to the time of year when the average daily ambient temperature is +10 degrees Celsius or lower.

PROBLEM STATEMENT

Increasing demands on the quality of life are leading to higher standards for ventilation systems used in modern construction. These standards affect not only people's comfort but also their health. Many studies have shown a correlation between the amount of fresh air supplied and the number of cases of so-called "building-related illnesses". There is also an increase in employee productivity when the quality of the indoor environment improves. Therefore, improving the health, comfort and safety of people in buildings can bring economic and social benefits through increased productivity, reduced sickness and healthcare costs.

The use of more modern ventilation systems is advisable and has an economic justification both in new buildings and in retrofits. However, this leads to higher capital and operating costs, as more outside air needs to be handled and maintained at the supply and indoor air requirements. Today, there is a wide range of technical solutions available to reduce operating costs, based on passive heat recovery, thermodynamic cycles and various combinations of these methods.

After a review of modern technical solutions in the field of energy efficiency of life support systems and taking into account the technical and economic indicators of heat recovery equipment, a number of problems were identified related to the development and application of devices for active and passive heat recovery. Of all the available technical solutions that allow the use of thermal energy from exhaust air, rotary regenerators are the leaders in terms of price and quality. However, their design limits their use for retrofitting existing systems.

Understanding these disadvantages of existing passive and combined heat recovery systems, such as unstable operation and reduced efficiency at low outdoor temperatures, increased number of heat exchangers in the supply line and limited use of certain types of recuperators, leads to the need for a universal heat recovery system that can adapt to new and modernised ventilation systems. The implementation of such a system will have a significant environmental and economic impact.

CHAPTER 1
PREFACE AND AN OVERVIEW OF THE LITERATURE

1.1. Energy consumption by buildings

The ever-increasing number of people on planet Earth, coupled with the growing trend towards concentration of a significant part of the population of developed countries in large cities, leads to a steady increase in energy consumption by buildings and structures.

Increased energy consumption in the current environment of non-renewable energy sources is consistently leading to significant changes in the planet's ecosystem. According to the International Energy Agency, in 2018, the demand for primary energy grew by 2.3%, almost twice the average growth recorded in 2010 [1]. Moreover, since 2015, there has been a steady increase in the annual growth rate (Figure 1.1).

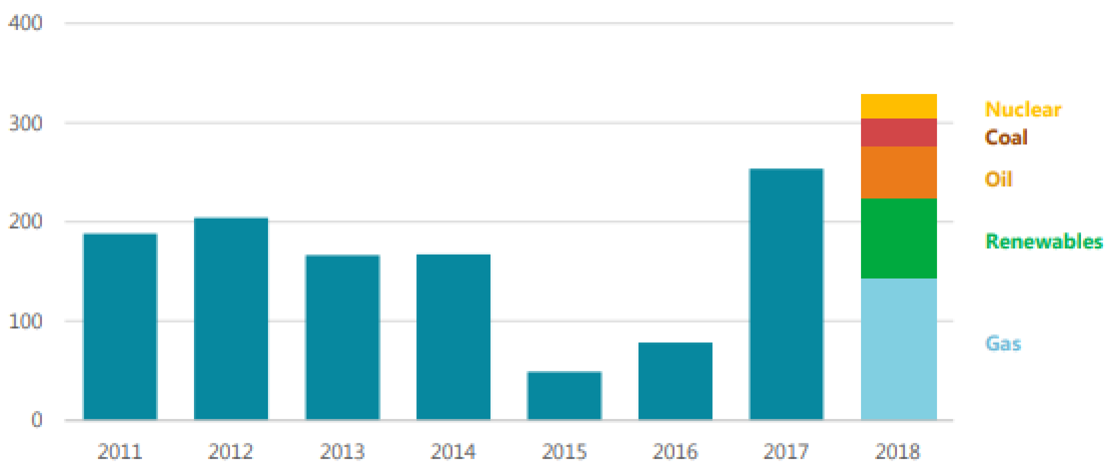

Figure 1.1 - Growth rate of non-renewable energy sources

The limited amount of fossil resources, as well as the impact of their use on global warming and ozone depletion, along with technological barriers, make it impossible to effectively increase energy production from renewable sources. To ensure the sustainable development of humanity, it is necessary to focus on reducing energy consumption in all areas of life.

When analyzing the components of global energy consumption growth, the growth of energy consumption in the non-industrial sector, in particular in residential and commercial buildings, stands out. Today, with industrial energy consumption projected to decline from

4.2% in 2019 to 2.2% in 2050, the share of energy consumption in residential and commercial buildings is growing steadily by 1.7% annually. In developed countries, energy consumption by buildings is currently estimated at between 20% and 40% of total energy consumption [2].

For many years, among the main engineering systems in buildings, the heating, ventilation and air conditioning system has been a leader in terms of energy consumption. According to statistical surveys, the share of energy consumption of heating, ventilation and air conditioning systems is at least 55% of the total energy consumption of buildings, and in some regions this number can reach 70% [3].

The development of building technologies can significantly reduce energy losses in modern buildings by using efficient insulation materials and energy-saving glazing to maintain the microclimate in enclosed spaces [4].

However, it is impossible to create a completely airtight building envelope, as any modern building requires a general exchange ventilation system. Growing demands on the quality of life set higher performance standards for ventilation systems used in modern construction. These characteristics affect not only the comfort of visitors, but also their health. The link between the frequency of air exchange and the number of cases of work-related illnesses in a building has been scientifically proven [5].

Studies have also shown an increase in employee productivity by improving the quality of the indoor environment [6].

Thus, improving the health, comfort and safety of people in buildings can bring economic and social benefits by increasing productivity, reducing sickness and healthcare costs. The rational and economically feasible use of advanced ventilation systems is advisable both in the construction of new buildings and in the modernization of existing buildings [7].

1.2. Analysis of research and publications

Improving the efficiency of ventilation systems has been an urgent task over the past decades due to the wide range of technical solutions offered by the ventilation industry today [8].

Assessment of the technical and economic efficiency of measures to reduce energy consumption of ventilation and air conditioning systems is a complex task with many factors. Firstly, it requires a complete analysis of several air handling processes, such as transport, cleaning, cooling, heating, dehumidification, humidification and heat recovery. Secondly, there is a variety of technical solutions for these processes, which differ in their technical implementation and physical principles of operation. Thirdly, the devices in question interact with each other, which affects their performance. For example, assessing the efficiency of a rotary regenerator and a heat pump separately will not reflect their actual performance when they are connected in series in a ventilation unit, as their overall aerodynamic drag will affect the operation of the fans and change their energy consumption characteristics.

The role of temperature treatment of air in ventilation systems is key in the context of the above-mentioned energy consumption. Most modern technical means that increase the efficiency of heating, ventilation and air conditioning systems are responsible for the energy efficiency of this process [9].

Depending on the region and climatic conditions, the average annual temperature difference between the internal environment of a building and the outside air can be positive or negative. Thus, methods to improve the efficiency of the air temperature treatment process can be divided into improving cooling and heating methods responsible for both processes.

Using the properties of exhaust air to reduce energy consumption is a major area of technology in this field. Creating a match between the thermophysical parameters of the exhaust air from the premises and the air coming in from the outside through natural heat exchange processes is an effective method of reducing overall energy consumption and has a good scientific basis.

There are two main groups of heat recovery devices used in ventilation and air conditioning systems. These groups include passive and active devices. Passive devices are characterized by very low or zero energy consumption compared to the heat energy transferred. Passive devices include plate heat exchangers, rotary regenerators, heat tubes and heat exchangers with intermediate heat transfer fluid. The efficiency of these devices depends on their design and can vary.

Today, the most efficient regenerators are rotary regenerators, which, according to certain estimates, can reduce the energy consumption for processing the air entering the room by 85% in energy terms through the heat recovery process [10]. However, the design features of such regenerators place certain limitations on their use, especially from a technological and regulatory perspective [11]. As rotary regenerators have their own design features, in some cases, mixing of inlet and outlet air is not allowed, which limits their use in some facilities, such as catering, healthcare facilities, and some industrial premises.

The use of rotary units is limited by their large dimensions, as in most cases it is necessary to increase the cross-section of other elements of the ventilation system to accommodate the rotary regenerator. This leads to an increase in the size and cost of ventilation equipment, and creates additional technological and economic difficulties when modernizing existing ventilation systems.

Improving the methods of rotary heat exchangers is considered as a transition from the analytical determination of the effective geometric parameters of rotor filling [12] to research methods based on mathematical modelling. More modern research methods make it possible to take into account such factors as aerodynamic drag, the level of air pollution with particulate matter, condensation and ice formation on heat transfer surfaces [13], along with the thermophysical parameters of the treated air.

Works [14], [15] consider methods for improving the efficiency of plate heat exchangers and identify independent variables for their optimization. The advantages of the structural-modular approach in the development of optimization algorithms and programmes are studied. The OPTO-2010 programme, which allows solving several tasks, is investigated, in particular:

- selection of the optimal heat exchanger by calculating all possible options;

- sorting the obtained results by ascending or descending of one of the efficiency criteria, as well as assessing the impact of various parameters on the overall efficiency of the heat exchanger;

- Calculation of the optimal arrangement of devices in the heat exchanger, not just the most optimal heat exchanger device;

- optimization of both design and operating parameters.

The application of similar simulation research methods to plate heat exchangers, as well as the use of special materials, can help to involve the phase transition in the heat exchange process and increase their design efficiency by up to 65% under certain conditions [16]. In particular, the results of experimental studies conducted by Rasouli and co-authors on four 10-storey office buildings showed actual annual energy savings from the use of plate heat exchangers at the level of 20% for cooling and 40% for heating [17]. However, Belonogov [18] provides an explanation for the significant deviations between the predicted and actual figures, which reflect the dependence of the thermal efficiency of plate heat exchangers on the temperature difference between the supply and exhaust air in humid conditions.

Calculations with some assumptions show a significant decrease in thermal efficiency at a temperature difference of less than 15 K on heat exchangers.

Heat recovery units with intermediate heat transfer fluid or RAC (Run Around Coil) offer a wide range of retrofit possibilities that differ from the above examples. In addition to the heat exchangers integrated into the ventilation system, these devices have a heat transfer fluid circuit in their design, which allows for more flexible management of the recovered energy and increased efficiency of the heat exchange processes.

The inclusion of heat accumulators in the coolant circuit allows the actual amount of recovered heat energy to be equalised with the actual demand of the ventilation system and to use alternative sources of heat energy to provide energy supply [19]. This ensures optimal use of thermal energy and increases system efficiency.

RAC systems have a wide range of applications. They allow for complete separation of supply and exhaust air flows, and the dimensions of heat exchangers can be adjusted to match other components of ventilation systems, as well as their location depending on the required equipment configuration. However, these systems are of low interest to the scientific community: out of 100 published papers on passive heat recovery in 2018, only 3 papers were devoted to RAC systems [20]. In terms of energy efficiency, they are also inferior to other types of passive devices, showing estimated energy savings of no more than 45% [16].

The RAMEE (Run Around Membrane Energy Exchanger) technology, which is being actively researched in the field of heat exchangers with intermediate heat transfer fluid, has brought new opportunities for using latent heat of the phase transition. According to various sources, this technology can achieve energy savings of 55% to 65% [21].

Passive heat recovery systems have their own general characteristics, including a limited ability to provide full temperature treatment of the air. In order to equalise the temperatures between the inlet and outlet air, it is necessary that the area of the heat exchanger that transfers heat energy is infinite, and that there are no energy losses during the transfer, for example, in RAC systems. However, the increase in heat transfer surface area is limited by the increase in aerodynamic drag of the heat exchanger, which leads to an increase in energy consumption by the ventilation system fans. Since it is not practical to fully heat or cool the air using passive recovery means alone, heaters or coolers must be installed, which also increases the power requirements of the fans.

The study of the influence of the design of some components of the ventilation system, such as filters, on the power characteristics of fans has been considered in detail in both Ukrainian and foreign studies [22]. However, the literature does not sufficiently address the issue of the negative impact of increasing the aerodynamic drag of heat exchangers on their efficiency.

Passive-type devices used to heat air at low ambient temperatures are characterised by the risk of icing. In Ukraine and abroad, research on icing processes and methods to reduce its impact on the efficiency of ventilation systems is being actively conducted [23]. The risk of icing, which can lead to complete or partial blockage of the heat exchanger channels, limits the use of the full potential of these devices during their maximum use. To protect different types of heat recovery units, it is necessary to use control methods that reduce their performance during periods of icing risk, such as cycling for plate heat exchangers, reducing the rotor speed of regenerative heat exchangers, and controlling the flow rate with a three-way valve in RAC and RAMEE systems.

The work of S. M. Anisimov and other authors [24] showed an inverse relationship between the risk of icing and the rated efficiency of the heat exchanger and the relative humidity of the removed air. Together with the decrease in the thermal efficiency coefficient with a decrease in the temperature drop across the heat exchanger [17], it becomes obvious that the range of outdoor air parameters in which passive heat exchangers can achieve high efficiency is limited. In actual operating conditions, the uncertainty of assessing the performance of passive heat exchangers requires reserving heat or cooling capacity at the design stage of ventilation units. This leads to an increase in the size of coolers/heaters, aerodynamic drag, and capital costs for their purchase and installation [25].

This approach, given the need to create an electricity and heat supply infrastructure with excess capacity [26], can have scale implications. Especially during the heating season, when the outside temperature requires active heating on the supply line and the heating systems are not yet activated, there is a drop in the temperature of the ventilated space. In some cases, this drop in temperature is so significant that it is necessary to completely switch off the ventilation system for this period. This leads not only to a decrease in comfort in the building, but also to an increase in comfort in the building, but also increases the risk of spreading colds.

1.3. HVAC Research results

Given the limited efficiency of passive heat exchangers at small temperature differences between supply and exhaust air, their use in air cooling processes cannot provide the necessary energy savings. When cooling the air, there is a need for excessive cooling of the supply air to temperatures below the comfort level to effectively ensure thermal comfort [27]. In such cases, the temperature difference across the passive heat recovery unit can reach negative values, which leads to an undesirable effect of its use. In countries with warm and humid climates, where the temperature difference is also high, the use of the most efficient passive heat recovery methods also has its limitations.

Studies show low efficiency of their application for both temperature and humidity treatment of air [28].

To achieve high energy efficiency when the main task is to cool the air, a key method of reducing energy consumption is to modernise the design of chillers.

Air cooling in HVAC systems is most commonly achieved through evaporative and vapour compression cooling, which are the main and widely used methods. Similarly to air heating, there are several opportunities to reduce the required cooling capacity through passive utilisation. However, in addition to these possibilities, it is also possible to reduce energy consumption by changing the design of chillers while maintaining their cooling capacity.

Evaporative air cooling, whether direct or indirect, has been in active use for over a century. This method provides efficient cooling with minimal energy consumption, low capital and operating costs, and minimises the negative impact on the ecosystem, as it does not require the use of refrigerants that harm the ozone layer and contribute to global warming. However, in modern construction, this method has its limitations, primarily due to the growing demands for indoor comfort. Evaporative cooling cannot cool the air to temperatures that are lower than the wet bulb temperature of the air coming from outside. Therefore, the parameters of the supply air air inflow parameters are often on the verge of the comfort zone, and internal factors, such as an increase in air temperature during its passage through the duct, can cause a violation of the comfort state.

Improving an evaporative cooling system is directly related to solving two problems: reducing the minimum cooling temperature and increasing efficiency. To achieve the first objective, it is possible to reduce the wet bulb temperature by pre-cooling the supply air. There are examples of designs in the literature that use passive ground heat exchangers to pre-cool the air before the evaporative cooler. For example, study [30] describes a design that uses a ground heat exchanger for pre-cooling. According to the authors, for a system with an air flow rate of 1700 m3/h, it was possible to reduce the wet thermometer temperature in the pre-cooler, which is cooled by four vertical wells about 45 m deep, by 8-10 °C. This significantly expanded the range of cooler control and the ability to maintain comfort, but also led to an increase in capital expenditure for its installation.

Foreign authors describe methods of increasing the efficiency of evaporative cooling by combining direct and indirect evaporation devices in one unit [29]. According to this concept, an indirect evaporation pre-cooler is used, which is cooled by the evaporation of water in the recirculation air stream. According to the authors' calculations, this scheme can reduce the building's energy consumption by energy consumption of a building by 15%. Study [30] describes a combination of direct evaporative cooling and nighttime radiation cooling. By accumulating cold coolant in the accumulator tank at night, it is proposed to reduce the wet bulb temperature of the outdoor air before it is supplied to the evaporative cooler.

This approach, in addition to expanding the control range, allows for a significant increase in performance compared to conventional direct evaporative cooling devices. In addition to circuitry solutions, improvements to individual components are also proposed to modernise evaporative cooling systems. For example, Rubtsov and other researchers describe the design of an oscillating spray nozzle that can provide the required degree of water dispersion when the circulation pump pressure is reduced from 30 to 16 bar, which significantly reduces its energy consumption [31].

The vapour-compression cooling method is carried out by means of a thermodynamic cycle, and its energy and economic indicators can also be improved without changing its performance.

The energy efficiency of a vapour-compression cooling system directly depends on the values of the evaporation and condensation temperatures. The evaporation temperature is largely determined by the requirements for the heat treatment of the air. An evaporation temperature that is too high does not allow achieving the required change in air temperature, which is taken into account when considering the possibility of air heating in air distribution systems and air assimilation. A boiling point that is too low can lead to excessive condensation and its freezing on the evaporator surface during direct air cooling. In the case of systems with an intermediate coolant

In the case of systems with an intermediate heat transfer medium, the upper limit of the permissible boiling point is further reduced due to additional heat transfer losses in the pipework, while the lower limit is limited by the risk of freezing of the heat transfer medium.

This latter factor can be avoided by using non-freezing aqueous solutions, but this increases the operating costs of such systems and increases additional environmental risks.

One of the most important areas for improving the operating cycle of a low-pressure vapour compression machine is to control the useful overheating in the evaporator. Ventilation and air conditioning systems are characterised by significant fluctuations in the evaporator air supply temperature, which leads to changes in the heat load and overheating. Thanks to the proliferation of electronic throttling devices, it has become possible to control the amount of useful overheating in HVAC chillers using both traditional dependencies on refrigerant temperature and pressure and new principles of accounting for the dimensional characteristics of the two-phase flow in air evaporators [32].

Since there is a limited range of available boiling points in air cooling, the literature often provides examples of methods for reducing the condensation temperature. In conventional air-cooled heat treatment schemes, heat is removed from the condenser using external air. Obviously, air cooling requires high ambient temperatures, and the higher the temperature, the greater the load on the cooling system. At the same time, high condensing temperatures dependent on the ambient temperature have a negative impact on the energy efficiency of the vapour compression cycle. This creates conditions under which the air cooling system operates at its least efficient air cooling system operates least efficiently during peak loads. Thus, with various attempts to achieve a high level of indoor comfort, the increase in cooling system capacity to overcome peak loads will be non-linear.

The cost per kW of cooling capacity at peak loads will be significantly higher than at average operating conditions.

The above-mentioned examples of reducing the temperature of the evaporative cooling medium also apply to vapour compression cooling. By using heat storage devices and natural heat reservoirs such as soil and groundwater, a low-temperature medium can be stored. This makes it possible to use the steam compression unit in the most "comfortable" conditions, for example at night, with the help of cold storage. This approach has a double positive effect: on the one hand, cold production takes place at low ambient temperatures, which increases the efficiency of the vapour compression cooling system; on the other hand, it allows electricity consumption at a nightly rate and reduces operating costs [33].

However, it is necessary to take into account the significant capital costs associated with the implementation of the proposed measures, as well as the limitations imposed by the location and purpose of the facility. For example, the possibility of installing ground collectors is limited depending on the location of the facility [34], and the installation of storage tanks requires additional space occupied by equipment, which reduces the investment attractiveness of the project as a whole according to [35].

Too low a reduction in condensing temperature in winter for cooling rooms, such as server rooms, data centres and some production facilities, is also unacceptable. Reducing the condensing pressure below a certain threshold leads to a decrease in the capacity of mechanical throttling devices and disruptions in the operation of the evaporator of the air cooling system.

Today, HVAC manufacturers offer devices to control the condensing pressure, but their use makes it difficult to further reduce the compressor's energy consumption by reducing the pressure drop. However, installing a pump in the liquid line ensures a stable pressure upstream of the throttling device and avoids this limitation [36]. Nevertheless, due to the specifics of the task and the limited possibility of using free-cooling in such conditions, this technology has not yet found wide application in practice.

When designing a ventilation system for an industrial building, the specific scheme is chosen by the designers, taking into account the conditions on site.

Typically, in industrial buildings, ventilation, heating and air conditioning systems are combined into one system. This makes it possible to use a common network of air ducts and respond quickly to temperature changes. The most common ventilation equipment in industrial complexes is central air conditioning.

Automation of ventilation and air conditioning control systems is one of the subsystems of industrial building management, so it is important to take into account an integrated approach when formulating requirements for its management. In addition, the overall automation system of an industrial building includes:

- Lighting
- Boiler room and individual heating point
- Refrigeration centre (if required)

- Electricity supply
- Air curtains for entrance groups
- Elevators and escalators
- Fire alarm and smoke removal systems

An industrial building is an object with distributed parameters that can vary in different sections. For example, some materials require a certain temperature and humidity to be stored, and chemical products require special ventilation systems. Industrial complexes are equipped with air ventilation and air conditioning systems that create a specific microclimate for each individual area of the complex. This can lead to differences in temperature between the outside environment, individual areas of the industrial building and public areas. Therefore, each section of an industrial building faces the challenge of creating its own microclimate.

This is achieved through the introduction of automated control systems in industrial buildings. The list of equipment used to control the microclimate in industrial complexes around the world includes Delta Controls ORCA, Johnson Controls, Honeywell, Sauter, Siemens, which provide control from 1000 to 18000 points.

Electricity costs for a large industrial and entertainment complex of about 50,000 m2 can exceed UAH 10 million per year. To save money, it is proposed to use simple automation using Smart grid systems, which include a power quality meter with a digital interface and a current measuring transformer with continuous recording of energy consumption for different groups of equipment and tenants. This system is capable of generating reports, analysing trends and optimising system parameters.

CHAPTER 2
SYSTEMS ANALYSIS OF THE SUBTLE AREA SYSTEMS FOR VENTILATION AND AIR CONDITIONING

2.1. Application of ventilation and air conditioning systems in industrial buildings

The most typical supply and exhaust ventilation system used in industrial production features a broad inflow to the working area, a local zone, and local extraction of hazardous compounds directly from their sources.

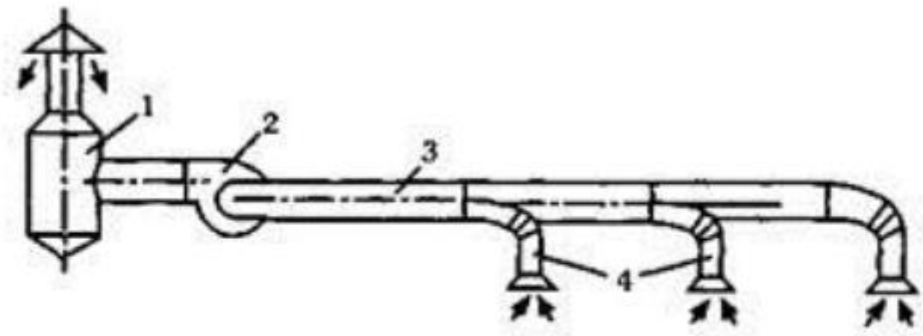

Figure 2.1 - Scheme of exhaust ventilation

A cleaning device 1, fan 2, central fan 3, and exhaust air ducts 4 make up exhaust ventilation (Fig. 2.1).

After cleaning, the air must be released at a height of at least 1 m over the ridge of the roof. Direct exhaust apertures in windows are not permitted.

The most typical ventilation system used in industrial production is a supply and exhaust ventilation system with a general intake to the working area and local extraction of hazardous compounds directly from their sources.

In industrial buildings where there are considerable amounts of harmful gases, vapors, and dusts, the exhaust should be 10% greater than the supply to prevent the forced entry of hazardous materials into nearby rooms with lower levels of hazard.

It is possible to use the air from the premises itself after purification in the supply and exhaust ventilation system in addition to outside air.

Recirculating indoor air is a technique used during the colder months to prevent wasting energy on heating supply air. However, there are a variety of sanitary and fire safety regulations that must be met before recirculation is allowed.

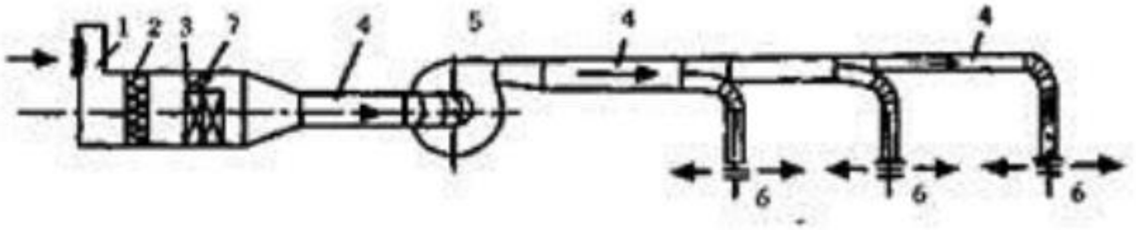

Figure 2.2 - Supply ventilation scheme

Air intake device is part of the mechanical supply ventilation scheme (Fig. 2.2). 1; air purification filter air duct network 4, fan 5, air heater (warmer), 2, air heater, 3, and input nozzles with air inlets with nozzles 6. If supply air does not need to be heated, it is routed through the bypass duct and into the production area. 7.

Devices for bringing in air must be placed where there is no dust or gas contamination of the air. They ought to be situated at least 2 meters underneath.

ground level, no closer than 25 meters horizontally or vertically from the exhaust ventilation ducts.

Typically, a dispersed flow of supply air is provided to the space using specific nozzles.

Hygienic standards state that when using the general exchange type of ventilation, the air condition in a ventilated and air-conditioned premises is maintained by injecting clean air with the necessary temperature and humidity characteristics and removing air that has the maximum permissible concentration of a pollutant.

As a result, general exchange ventilation systems should have tools and components for bringing in outside air (with recirculation - appropriate filters), modifying it to the required standards, transporting it to the premises, and removing air from the premises.

Figure 2.3 displays a condensed diagram of the HVAC system without a refrigerator or sprinklers.

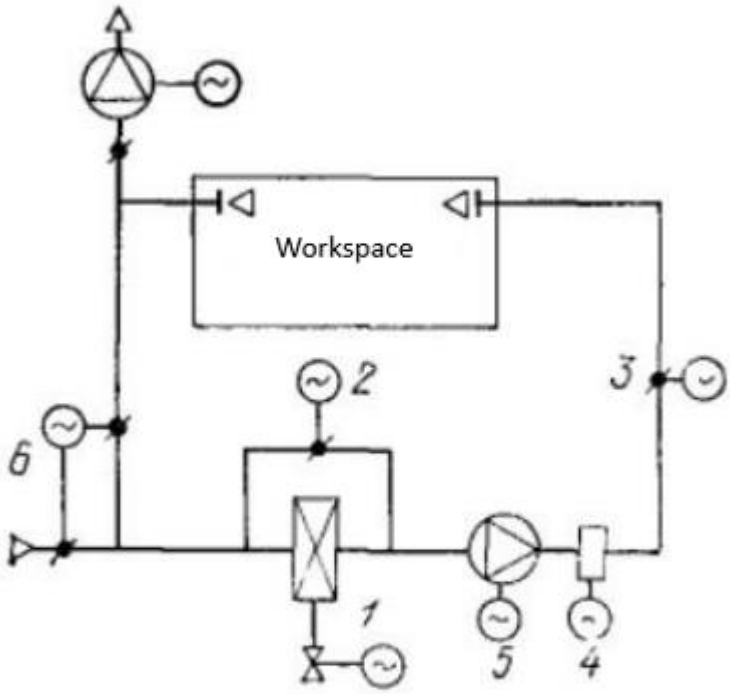

Figure 2.3 Functional block diagram of the general ventilation system

2.2. Problems of energy efficiency of ventilation and air conditioning systems

There are several common problems related to energy efficiency in ventilation and air conditioning (HVAC) systems. These issues can lead to increased energy consumption, higher operating costs, and reduced overall system performance. Here are some key problems:

Inadequate system design: Improper sizing and design of HVAC systems can result in inefficiencies. Undersized systems may struggle to meet the desired cooling or heating demands, leading to longer runtimes and higher energy consumption. Oversized systems can short cycle, turning on and off frequently, which reduces efficiency and increases wear and tear on components.

Poor insulation: Inadequate insulation of ductwork, pipes, and buildings can lead to thermal losses or gains. Leaky ducts, poorly insulated walls, or unsealed openings can cause conditioned air to escape or outside air to infiltrate, forcing the system to work harder to maintain the desired temperature.

Lack of maintenance: Neglected maintenance can significantly impact energy efficiency. Dirty filters, clogged coils, and blocked vents can restrict airflow and hinder heat exchange, causing the system to work harder and consume more energy. Regular maintenance, including cleaning and filter replacement, is crucial to ensure optimal system performance.

Inefficient controls and settings: Inaccurate thermostat placement, outdated controls, or improper settings can lead to unnecessary energy consumption. For example, if the thermostat is located near a heat source or in direct sunlight, it may misinterpret the actual indoor temperature, resulting in inefficient cooling or heating operation.

Inefficient fan systems: Inefficient or oversized fans can consume excess energy. The use of variable speed drives (VSDs) can help adjust the fan speed based on the actual airflow requirements, reducing energy consumption during periods of lower demand.

Ineffective zoning: Lack of zoning or improper zoning can lead to unnecessary cooling or heating of unoccupied areas. Zoning allows for customized temperature control in different zones or rooms, ensuring energy is only directed where needed.

Inadequate air sealing: Poorly sealed windows, doors, and gaps in the building envelope can result in air leaks. Uncontrolled air infiltration or exfiltration can lead to energy losses and decreased system efficiency, as the HVAC system must compensate for the additional load.

Outdated equipment: Aging or outdated HVAC equipment may lack the energy-saving features found in modern systems. Upgrading to newer, more efficient models can significantly improve energy efficiency and reduce operating costs.

2.3. Determination of energy efficiency parameters

Determining energy efficiency parameters for ventilation and air conditioning systems involves measuring and analyzing various factors that affect the system's energy consumption and performance. Here are some key parameters to consider:

Energy consumption: The total energy consumed by the HVAC system is an essential parameter for evaluating energy efficiency. It can be measured in kilowatt-hours (kWh) or other relevant units. This can be obtained from energy bills or through energy monitoring systems installed within the HVAC system.

Coefficient of Performance (COP) or Energy Efficiency Ratio (EER): COP and EER are metrics used to assess the energy efficiency of cooling systems. COP is used for heat pumps and represents the ratio of heat output (in watts or BTUs) to the energy input (in watts) during heating mode. EER is used for air conditioners and is calculated as the ratio of cooling capacity (in BTUs) to the power input (in watts) during cooling mode. Higher COP or EER values indicate better energy efficiency.

Seasonal Energy Efficiency Ratio (SEER): SEER is a measure of the average cooling efficiency of an air conditioner over a typical cooling season. It considers the variations in load and accounts for part-load performance. SEER values are typically calculated using standardized testing procedures and are printed on equipment labels. Higher SEER ratings indicate higher energy efficiency.

Energy Efficiency Ratio (EER): EER is similar to SEER but does not consider seasonal variations. It represents the cooling capacity (in BTUs) divided by the power input (in watts) of an air conditioner under specific conditions. Higher EER values indicate better energy efficiency.

Airflow rates: The volume of air circulated by the HVAC system affects its energy consumption. Measuring and optimizing airflow rates can help ensure that the system is operating efficiently. Airflow can be measured using flow meters or calculated based on fan specifications and static pressure measurements.

Temperature differentials: Temperature differentials across the system, including supply and return air temperatures, condenser and evaporator temperature differences, and temperature differences across heat exchangers, can provide insights into the system's efficiency. Larger temperature differentials may indicate inefficient heat transfer or air distribution issues.

System losses: Identifying and minimizing losses in the HVAC system is crucial for energy efficiency. Losses can occur due to duct leaks, poorly insulated components, air leaks in the building envelope, or inefficient distribution systems. Quantifying these losses through energy audits or measurements helps prioritize energy-saving interventions.

Occupancy and demand patterns: Understanding occupancy schedules, usage patterns, and load variations in the building allows for optimizing energy efficiency. Analyzing occupancy data, including occupancy sensors or building management system (BMS) data, can help fine-tune system settings and avoid unnecessary energy consumption during unoccupied periods.

Heat recovery and energy reuse: Assessing the potential for heat recovery or energy reuse within the HVAC system can contribute to improved energy efficiency. This may involve recovering waste heat from exhaust air, utilizing heat pumps for simultaneous heating and cooling, or integrating renewable energy sources like solar thermal systems.

To determine these parameters accurately, it is often necessary to conduct energy audits, consult equipment specifications, use data logging devices, and collaborate with HVAC professionals or energy experts. These professionals can provide guidance on measurement techniques and analysis tools to assess the energy efficiency of ventilation and air conditioning systems effectively.

CHAPTER 3
HVAC MODEL DEVELOPMENT AND MODELLING APPROACHES

3.1. Indicators of technical and functional excellence of the building HVAC

The time it takes to create a stationary temperature regime in the device and in the ventilated space as a whole is frequently overlooked, as earlier research have demonstrated. The steady-state temperature regime in the device and the ventilated space as a whole are typically calculated mathematically while designing. The performance $\dot{V}_p$ of ventilation and air conditioning is typically determined from diagrams of the aerodynamic characteristics of the fan and the system as a whole. The design typically makes use of mathematical calculations that only consider the steady state of all elements of the system and the room itself.

However, this is not the sole method for calculating the fan's necessary air flow rate $\dot{V}_p$. It is feasible to use the energy balance, or more specifically, the heat balance, if no dangerous compounds are produced. The next phase in designing an HVAC system involves using diagrams of the aerodynamic characteristics of fans, where the operating point of the fan and the system as a whole is used to assure the requisite air flow rate air flow rate $\dot{V}_p$.

The equilibrium in the ventilation and fan combination occurs if the total air flow rate is equal to that of the system as a whole and additionally losses due to leaks and leaks in air pipework and in the ventilated room as a whole. This is based on the law of energy conservation and mass balance in the system. The energy balance is made up of the flow of air's energy and the energy needed to bring it up to normalized thermodynamic properties. The overall losses line up with the total losses due to aerodynamic pressure in the HVAC components:

$$\Delta P_{\Sigma} = \Delta P_{ai} + \Delta P_{ad} + \Delta P_{noz} + \Delta P_{\mathrm{ex.op}} + \Delta P_{le}, \tag{1.1}$$

where ΔP_{Σ} - pressure losses in the air intake (ai), air duct (ad), nozzles (noz), exhaust holes (ex.op), and leakage (le), respectively.

Total fan pressure in P_f is determined by the following expression:

$$P_f = P_{t.v} + \Delta P_{\Sigma} \tag{1.2}$$

where $P_{t.v}$ is the theoretical value of the pressure created by the fan obtained by calculation, Pa.

$$\Delta \mathrm{P}_{\Sigma} = \sum_{\mathrm{i}=1}^{\mathrm{n}} \Delta \mathrm{P}_{\mathrm{i}} \tag{1.3}$$

where $\Delta \mathrm{P}_{\mathrm{i}}$ - losses for air transition into the fan casing, losses in the casing in the fan casing, in the inter-blade channels, in the rotary devices, etc.

In general, these losses can be expressed in the following form:

$$\Delta P_{\Sigma} = k_{d.c} \cdot \dot{V}_p \tag{1.4}$$

where $k_{d.c}$ is the total aerodynamic drag coefficient of the fan.

If the above equations are substituted into equation (1.3), we obtain the following expression:

$$\dot{V} = \sqrt{\frac{P_{t.v}}{k_{d.c}+k_{t.r}}} \tag{1.5}$$

where $k_{t.r}$ is the total resistance coefficient of the system elements of the ventilation system.

The total power consumption by the motor of the air handling unit is respectively:

$$\dot{W}_v = \frac{\dot{P}_f \cdot \dot{V}_p}{\eta_v \cdot \eta_e} \tag{1.6}$$

where η_v , η_e, are the efficiency coefficients of the fan and the electric motor.

According to the aforementioned equations, the fan's aerodynamic characteristics determine its air flow rate $\dot{V}$ and pressure $P_{t.v}$, which should both be operational to achieve the highest energy efficiency possible. If the fan's operating mode is changed (deviation from the calculated nominal values), the fan's energy consumption may go down, but the efficiency will remain the same. Different flow rate controls (such as throttling, changing the motor speed through different gears, and frequency-controlling an electric motor) can cause these efficiency issues.

3.2. Reducing heat consumption in the building HVAC

We can apply the idea of the average relative heat load $\overline{\dot{Q}_{ave}}$ across the year to estimate the heat consumption in the HVAC system, yielding the equation shown below:

$$\Delta Q_t = \psi_{l.\dot{Q}} \cdot \Delta T_{c.t.d} \cdot (1 - \overline{\dot{Q}_{ave}}) \cdot \sum t_i \tag{1.7}$$

where $\sum t_i$ is the time duration of the i-th mode of operation of the HVAC system, when heating and cooling the air, hours;

$\Delta T_{c.t.d} = T_{i.n} - T_{f.i}$ - calculated temperature difference $T_{i.n}$ (internal normalised) and $T_{f.i}$ (at the fan inlet), K;

$\psi_{l.\dot{Q}}$ - is a coefficient that takes into account leakage through leaks and local air suction in channels and pipelines.

The ventilation and air conditioning system's overall energy balance can be significantly improved by reducing heat consumption. If the fan flow rate control is successful, energy savings will result.

3.3. Reducing energy consumption by the fan when using different types of fan performance control

As was previously stated, controlling the fan output and heating or cooling of the air is a requirement for the HVAC system to function effectively. However, there are various methods of regulation, ranging from the most basic ones (throttling) that do not require significant investment to those that do, like variators the frequency of rotation of the magnetic field in the electric motor, which requires significant investments due to the cost of electronic devices used in this method of regulation.

At this point in its evolution, the HVAC uses an automated system to adjust its operating parameters in relation to ambient factors outside of the system.

The following equation can be used as a rough approximation to show how much less electricity is used by fans depending on the method of control:

$$\Delta\dot{W} \approx \psi_m \cdot \left[\frac{\dot{V}_{calc} \cdot p_{calc}}{\eta_v \cdot \eta_e} \cdot \overline{\dot{\mathrm{m}}}_{c.ave}^{\,b}\right] \frac{8{,}77}{3600 \cdot 1000} \qquad (1.8)$$

where p_{calc} is the design pressure of the fan, Pa;

$\dot{V}_{calc}$ is the estimated volumetric air flow rate by the fan, m3/h;

A characteristic value for the appropriate mode of fan performance control is the exponent of the degree b of the dimensionless mass flow rate of the fan. The highest value of b is obtained when employing frequency converters to regulate the fan motor (the indicator is at level 3), when using a guidance device (2-2.3), and when using the least efficient method of control, a throttle valve.

3.4. Economic assessment of ventilation system costs

All of the aforementioned factors, as well as operational indications in the form of a difference in the present value of costs for the implementation and operation of the HVAC system, should be considered in the overall assessment of the economic effect when comparing the HVAC:

$$\mathrm{E} = C'_1 - C'_2 = (K_1 - K_2) \cdot (E_n + 0.18) + c'_T \cdot \Delta Q_T + c'_E \cdot \Delta W_E, \tag{1.9}$$

where $C'_1,\ C'_2$, are the reduced costs for 1.2 variants of the implementation of the HVAC system;

K is the capital intensity for 1.2 variants of the HVAC implementation;

E_n - is the industry standard efficiency ratio.

The operating cost components (depreciation, repairs), which make up 18% of capital expenditures, are taken into account in formula (1.9).

The following equation can be used to calculate the present value of each of the consecutive components of the HVAC:

$$C' = \pi \cdot d \cdot l \cdot (c'_p + c'_i \cdot \delta_i) \cdot \xi \cdot (E_n + 0.18) + + \left[\frac{K_{ins}(p_v, b)}{p_v} \cdot (E_n + 0.18) + \frac{(c'_e \cdot T_v \cdot L^b_{ave}) \cdot \dot{V}}{1000 \cdot \eta_v \cdot \eta_e}\right] \cdot \left(\lambda \cdot \frac{l}{d} + \sum r\right) \cdot \frac{\rho}{2} \cdot \left(\frac{4}{\pi \cdot d^2}\right)^2 \cdot \dot{V} \tag{1.10}$$

Where d, l, is the length and width of the HVAC section, m;

c'_p , c'_i are the cost of 1 m^2 of the pipeline surface and 1 m^3 of insulation, respectively;

δ_i - insulation thickness, m;

ξ - is a coefficient that takes into account approximately the ratio $\sum d_i l_i$ for the entire ventilation system network;

λ - hydraulic friction coefficient;

$\sum r$ - is the sum of the coefficients of local resistances in the sections;

$\dot{V}$ - calculated air flow rate at the site, m^3/c;

ρ - air density, kg/m^3;

c'_E - unit cost of electricity

T_v - the duration of the fan operation;

L_{ave}^b – average relative flow rate in terms of power consumption of air at the site;

$K_{ins}(p_v, b)$ - capital costs for the ventilation unit, which depend on the fan pressure (motor power) and type of control fan performance;

p_v - fan pressure, Pa.

3.5. General approaches to modelling dynamic systems

In order to study energy usage and efficiency, models must be developed. Additionally, models are required to mimic various control scenarios in order to increase energy efficiency.

Heat and mass transfer equipment, such as air conditioning equipment (cooler, humidifier, and dehumidifier), heater (warmer), and a fan unit with an air duct system, make up the complicated structure of HVAC systems (Figure 3.1).

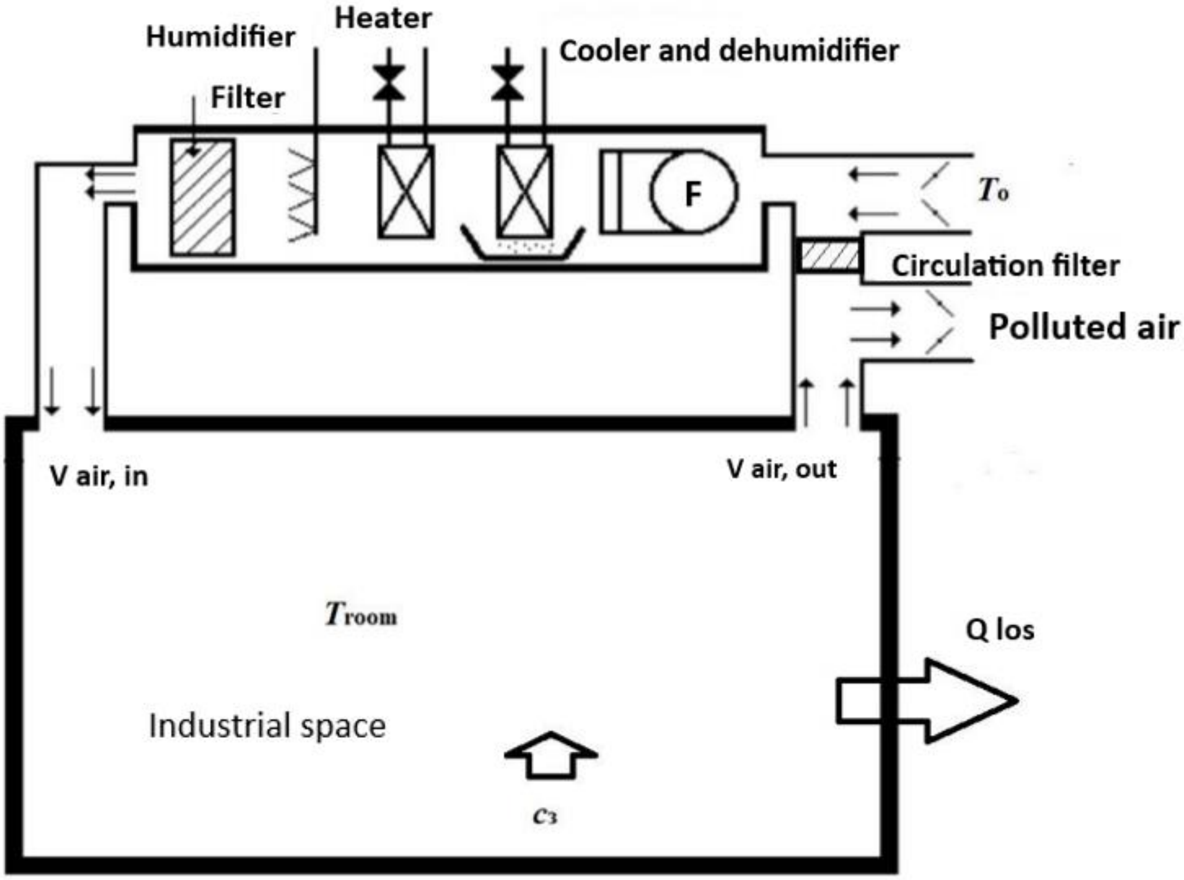

Figure 3.1 - General view of the HVAC system together with a ventilated room when describing the model

In order to adjust operating parameters including room temperature T_{room}, fan speed ω_{fan}, static pressure in the ductwork p_s, and chilled water temperature (or flow rate), HVAC systems also frequently involve a number of sensors and controllers. It is required to model individual components using measurable data and from the physical principles that define the processes in the system in order to accurately anticipate the energy consumption of HVAC systems.

Linear or nonlinear, static or dynamic, explicit or implicit, discrete or continuous, deterministic or probabilistic, as well as deductive, inductive, or flow models are some of the categories that models might fall under. Deductive models are those that are based on physical laws, and inductive models are those that are based on experimental data, according to this classification.

Inductive or deductive models can be used to describe models that combine the application of physical rules and experimental values of quantities.

While models based on statistical data processing typically take the shape of discrete and deterministic or stochastic models, physical approaches are typically utilized to achieve findings in continuous and deterministic models.

Static models are used for rapid parameter changes (e.g., mixed air temperature and carbon dioxide (CO2) concentration in the ventilation concentration in the ventilated room, as well as air and water flow rates (air mobility) and water flow rates), while dynamic models are typically used for processes of slow temperature and humidity (e.g., zone temperature dynamics, zone humidity dynamics, zone dynamics, zone humidity dynamics).

Similarly, a static model of heating/cooling can be created by treating the heat exchanger as if it had a constant heat transfer efficiency. For example, a dynamic model of cooling/heating through heat exchangers (HE) can be created by developing an energy balance based on the equation of water flow and air flow, leading to two differential equations.

Dynamic physics-based models are typically created using the thermal network method. In this method, heat transfer in HVAC components is frequently modelled as a substitution scheme of thermophysical characteristics and values in the form of an equivalent electrical network, where resistors and capacitors reflect thermal resistance (1/(h*A), where h is the specific heat transfer coefficient per unit area, A is the surface area through which heat transfer), and heat capacity $\hat{c}_w$, respectively, while electric current and voltage represent heat flux Q_{loss} and temperature T, respectively (Figure 3.2).

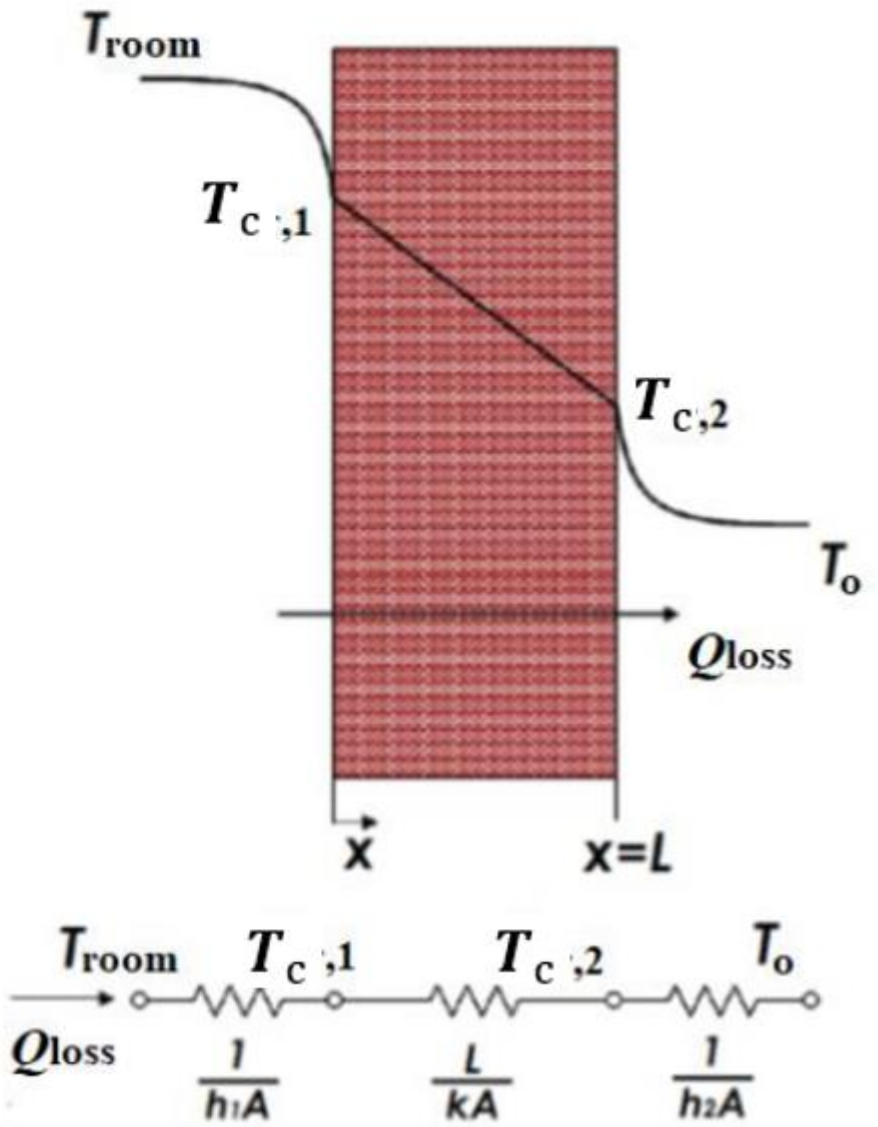

Figure 3.2 - Equivalent circuit diagram for heat flow through a room wall

Heating, ventilation, and air conditioning ("HVAC") system modeling is required to analyze and control energy usage, interior air quality, and ambient air quality. Three different methodologies are typically employed for HVAC modeling. The first way, referred to as the "black box" or empirical approach, collects data on system performance under specified conditions or during normal operation, and then uses mathematical techniques (such as regression analysis and neural networks) to determine correlations between input and output variables.

3.6. Empirical modelling of dynamic systems

The potential applications of empirical modeling have considerably increased as a result of recent advancements in computational artificial intelligence, particularly in the field of machine learning. Empirical modeling, in which the model is constructed on a DDM (data driven model), is the area of modeling that these new methodologies cover. DDM, as its name implies, is based on the analysis of system data, specifically the look for correlations between system state parameters (input data, internal and output values) without understanding of the physical processes occurring in the system (Figure 3.3).

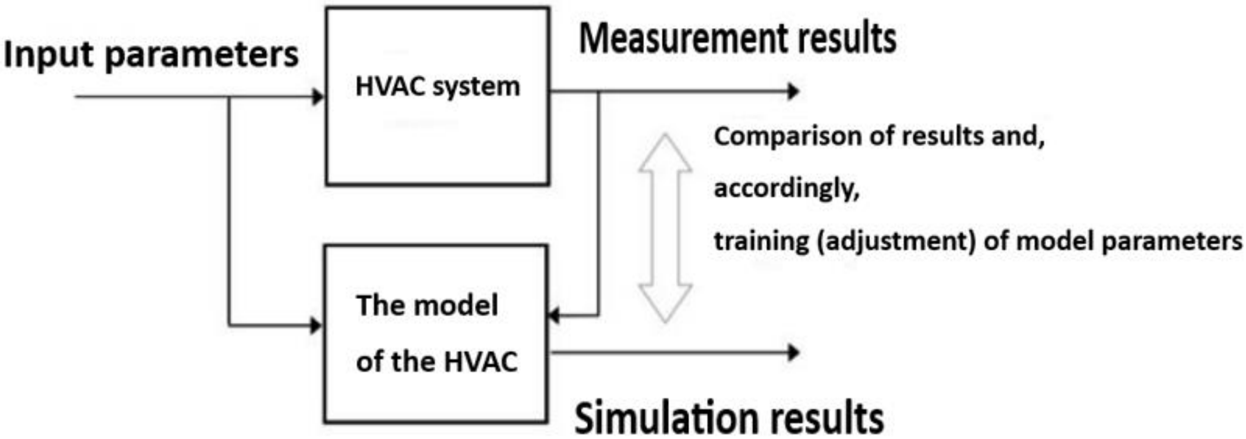

Figure 3.3 - Flowchart of building a model based on empirical methods.

3.7. Mechanistic modelling of dynamic systems

The use of physical laws (open "box"), in which system models are produced from using the fundamental principles of physics and in-depth understanding of the underlying process, characterizes this modeling strategy (Figure 3.4).

The development of mechanical (physical) models is based on the basic concepts of mass, energy, and momentum transmission. They are made up of a number of equations with numerical coefficients that reflect the building's geometry and thermal characteristics.

Numerous software tools that rely on numerical approaches are available to solve such systems. When the model's physical parameters are adjusted, there are issues specific to mechanical models.

If the model is sufficient for a particular mode of operation, then when the parameters at the level of parameter changes in empirical models go beyond beyond the normalized values, the mechanistic model gives more accurate values of the parameters in HVAC than the empirical model, which is accurate only on a distinct interval of parameter values, which are included, for example, in the regression equation.

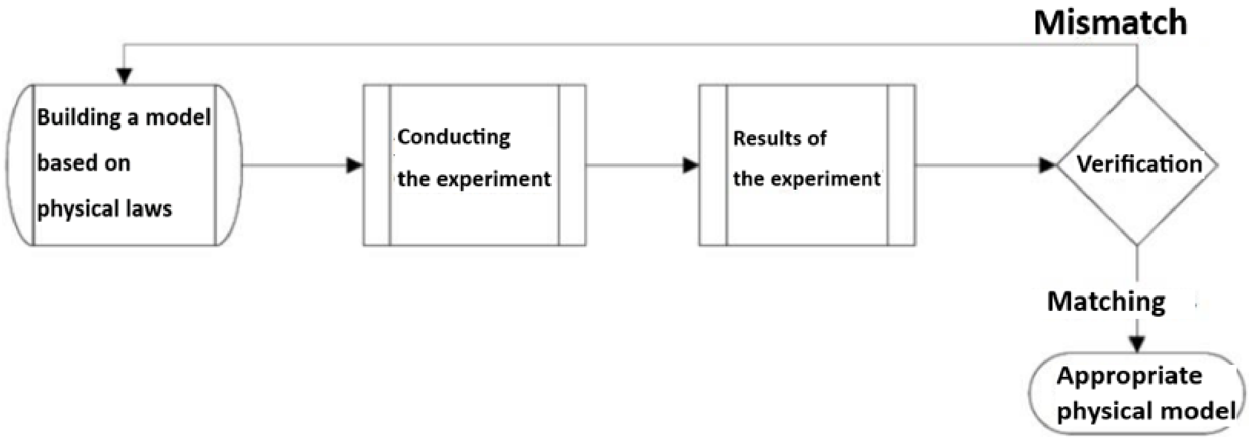

Figure 3.4 - Block diagram when using the physical approach to development system model

3.8. A combined approach to modelling dynamic systems

The basic framework of the model is created using physical methods in this approach, also referred to as the grey box or combined approach of the two preceding approaches, and the model parameters are established using parameter estimation algorithms from the measurement of system data.

It takes in-depth knowledge of the system and its operations to create models based on physical laws and define their parameters. However, data-driven models and their parameters can be derived without enough knowledge about the system and its workings. Grey box models benefit from both physics-based and data-driven models because the model structure is built using physics approaches, and the model parameters are estimated using system performance data.

While data-driven models have very high data accuracy when training the model (neural networks), they suffer from generalization outside the main mode in which the model was formed. Physics-based models have very good generation capabilities but are characterized by relatively poor accuracy. Grey box models combine the benefits of the other two categories, offering better accuracy than models based on physical principles and good options for adapting models in managing input data under particular operating conditions.

An HVAC and building model strikes a balance between the needed accuracy and model complexity. These models are built using information gathered from measurements of characteristics unique to each building, such as airflow rate, relative humidity, indoor and outdoor temperatures, and wind and solar radiation. Due to the lack of physical information about the building used in these models, the model values for the parameters typically have no physical significance. Although empirical models can outperform physics-based ones, this is only true for the specific building and HVAC system where the model was created.

These models may result in irrational and illogical outcomes if the input parameters exceed their bounds or if new parameters arrive (prior parameters vanish). Mechanistic and empirical models are combined in combined approach models, and knowledge of these models is only partially available. They are utilized to estimate parameters.

Stochastic models take into account random events. Certain HVAC installation processes can be characterized by probability density functions since they vary randomly. Standard normal and uniform distributions can represent a wide variety of physical processes. To determine the precise form of the distribution of a quantity, however, a significant amount of data is needed. Prediction of model parameters and environmental variables frequently uses this kind of model.

The goal of this project is to create a basic but adequate mechanical design model for a typical structure and a basic HVAC system that can also be utilized for ongoing control and monitoring.

The model allows for the use of constant thermophysical parameters for each layer of multilayer walls while accounting for the thermal inertia of the walls and roof. It is crucial to create a trustworthy model based on a physics-based model that takes into account all of these impacts. The remainder of the study is structured to give a thorough discussion of how HVAC models were developed, simulations to verify the suggested strategy using actual experimental data, and some closing observations.

3.9. Characteristics of dynamic and static models

A model's development and adjustment are generally difficult processes. When predicting the energy consumption of an HVAC installation (as a whole), it is important to first choose the model parameters that best capture the system's behavior and, consequently, its performance and energy usage.

When applying the modeling strategy based on mathematical and physical equations that explain the behavior of the system and the accuracy of the initial conditions of the system (Figure 3.5), can be expressed in the following ways:

$$f(\dot{x}_1, \dot{x}_2) = 0$$
$$f(x_2) = 0, \tag{2.1}$$

Where $\dot{x}_1$, $\dot{x}_2$ - are the time derivatives of the variables x_1, x_2

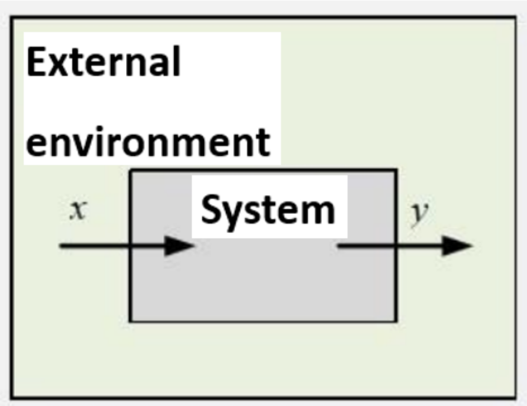

Figure 3.5 - System-environment with input parameters x and output parameters y

The system can have just one differential equation in the most basic scenario, and in the case of HVAC and ventilated rooms, this differential equation must invariably be the temperature of the air inside the room. The system has only one degree of freedom and only needs beginning conditions for this one variable if there is only one differential equation present.

Due to mathematical relationships between the input parameters x and the output parameters y, differential equations enable the modeling of physical systems.

Physical law-based models also include extra variables that describe the system's intermediate positions induce-and-effect interactions, where a system's output parameters induce a change in its input parameters over time, are the fundamental idea in physical models. This provides evidence that specific variables known as system state variables contain data regarding the prior state of the system. This conclusion's impact on the system state variable

System state is a variable that, if the system's state is known at a particular moment, if the state at that time and the future changes in the system parameters are known, it is possible to forecast the future state of the system's parameters.

In other words, you are not need to be aware of the prior values of the model's input parameters.

It should be highlighted that the system's states are not fixed; rather, there is an inverse relationship between the variables that characterize each state and the variables that hold the same information.

We can choose whether the system always retains the same basic parameters when creating system models, such as the geometry of the system, or its spatial description, and if it is changing or remaining stationary. It is evident that there must be a balance between the quantity of exhaust air and the outside air in order to explain the material balance of the system in the case of stationary mass air flow rate in the air system in the HVAC system.

3.10. Description of the characteristics of the industrial premises, such as HVAC load

In the sense that they demonstrate some resistance to change and take time to implement, all systems are dynamic. But the length of time over which the system changes affects how we observe the phenomenon of inertia.

In this work, we will examine the thermal inertia of a structure with ventilation and air conditioning. Typically, such properties are expressed in terms of time constants (Ts) for linear differential heat transfer equations.

Humidity and the amount of dangerous pollutants in the air are the microclimate elements that vary the greatest. The rate of molecule diffusion and mixing determines how variable these characteristics are in the premises. Processes of moisture sorption and desorption

As a result of the inertia and low detectability of non-stationarity, which are characteristics of surfaces on equipment and building envelopes, as well as their accumulation in the air volume of the room, air humidity and gas concentrations change rapidly.

The following characteristics of a particular microclimate parameter's development are noteworthy because they have a big impact on how quickly the HVAC system can get it to a certain value.

Aside from the forced changes in parameters brought on by influencing activities, it is feasible to make targeted changes to them throughout the course of the day.

Recent research has shown that "dynamic microclimate," or periodic variations in indoor air parameters near their average values, is a term used to describe periodic fluctuations in indoor air parameters that are well perceived by the human body, reduce fatigue, and increase labor productivity.

The ideal choice is to take into account air temperature swings with an amplitude of 2-3 °C and a duration of 40-120 minutes that are close to the optimal level.

Variations in air velocity with a period of 10–20 minutes and an amplitude of 0.2–0.5 m/s.

With a changeable working mode, it is only possible to maintain the microclimate of the premises during business hours, as well as, in some situations, technological characteristics. The parameters can be altered at will outside of the range of the maximum permitted values during non-working hours.

In this situation, the system is supposed to run in an enhanced mode prior to the start of business hours to raise the parameters to the desired level.

The mass transfer rate for typical molecular diffusion in the direction, which is calculated by Fick's first law, can be used to represent the distribution of the concentration of the pollutant c_p in the volume of the room:

$$\tilde{I}_p = -D_p \frac{dc_p}{dy} \tag{2.2}$$

where $\tilde{I}_p$ $-$ is the measured pollutant flux, kmol/(m^2 h);

$\frac{dc_p}{dy}$ - is the gradient of pollutant concentration, kmol/(m3 m);

D_p - diffusion coefficient for the pollutant in the air.

3.11. Description of the fan and formation of its model

The difference between the flow pressures behind and in front of the fan correlates to the total fan pressure:

$$p_t = p_2 - p_1 \tag{2.3}$$

Where p_1 - is the average pressure across the inlet section, Pa;

p_2 - is the average pressure across the outlet section, Pa.

The static pressure of fan p_s corresponds to the difference between the total pressure of fan p_t and the dynamic pressure of fan p_d:

$$p_d = p_t - p_s \tag{2.4}$$

The average flow rate v_{out} of the fan's flow output determines the dynamic fan pressure p_d:

$$p_d = \frac{\rho v^2{}_{out}}{2} \tag{2.5}$$

Flow rate from the fan (one of the approaches for averaging):

$$v_{out} = \frac{\dot{V}_{air,in}}{A_{out}} \tag{2.6}$$

Where A_{out} - is the cross-sectional area of the flow outlet from the fan;

$\dot{V}_{air,in}$ - air flow rate by the fan.

The full pressure of the fans, p_t , can range from tens to several thousand Pa, and their performance can range from several cubic meters per second to thousands of cubic meters per second. Typically, the linear speed at the blade ends is less than 150 m/s.

It is challenging to evaluate the effectiveness of the fan for a certain working environment because to the vast variety of dimensional characteristics, which are factors of the fan diameter, speed, and air temperature. To make it simpler to choose the best fan, common characteristics are utilized that are dependent only on the kind of fan (radial, axial, or diametric) and not on the actual size of the fans.

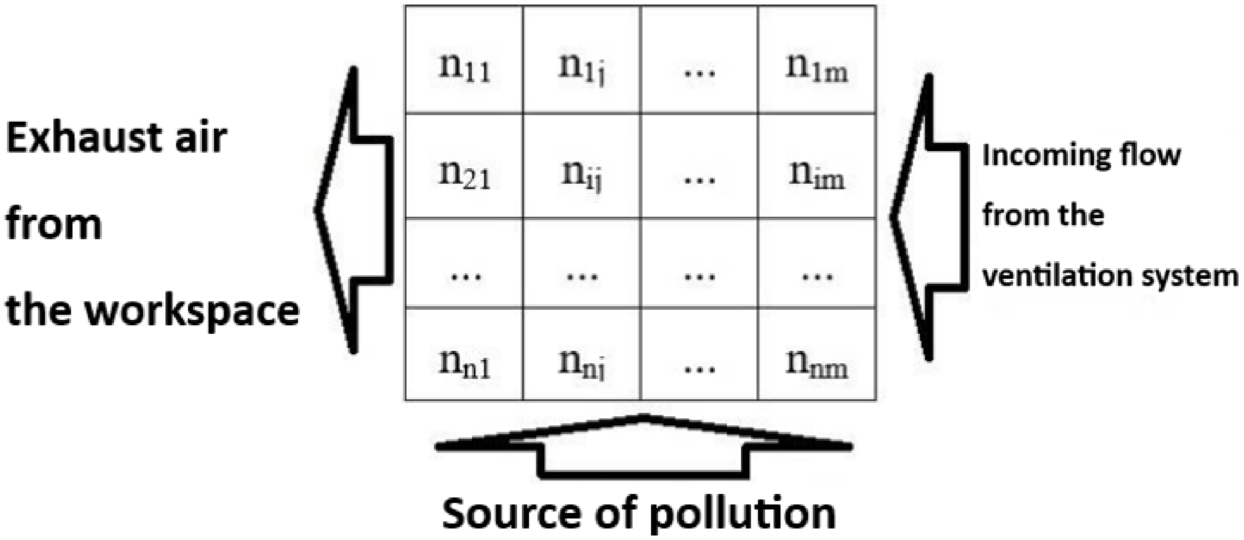

Figure 3.6 - Two-dimensional distribution of pollutant concentration in a room

Fans that are geometrically similar are characterized by common traits (Figure 3.6). The link between the dimensionless values of pressure ψ and power λ on the performance factor φ or φ_a for an axial fan type is an example of a typical fan characteristic. As a result, the performance of the fans is described by the following indicators:

- performance factor $\varphi = \dfrac{\dot{V}_{air}}{A \cdot v_{bl}}$ (2.7)

- axial velocity coefficient $\varphi_u = \varphi \cdot (1 - \bar{d}^2)$ (2.8)

- total pressure coefficient $\psi = \dfrac{2 \cdot p_t}{\rho \cdot v^2{}_{bl}}$ (2.9)

- power factor $\lambda = \dfrac{2 \cdot \dot{W}_v}{\rho \cdot A \cdot v^2{}_{bl}}$ (2.10)

Where $A = \dfrac{\pi \cdot D^2}{4}$ - cross-sectional area through which the air flow passes (for axial and radial fans), m^2 ;

D - is the diameter of the fan wheel, m;

$v_{bl} = \frac{\pi \cdot D \cdot n_f}{60}$ - linear speed at the end of the fan blades, m/s;

n_f - fan wheel speed, rpm;

$\bar{d} = \frac{d_b}{D}$ - relative bushing diameter (only for axial fans).

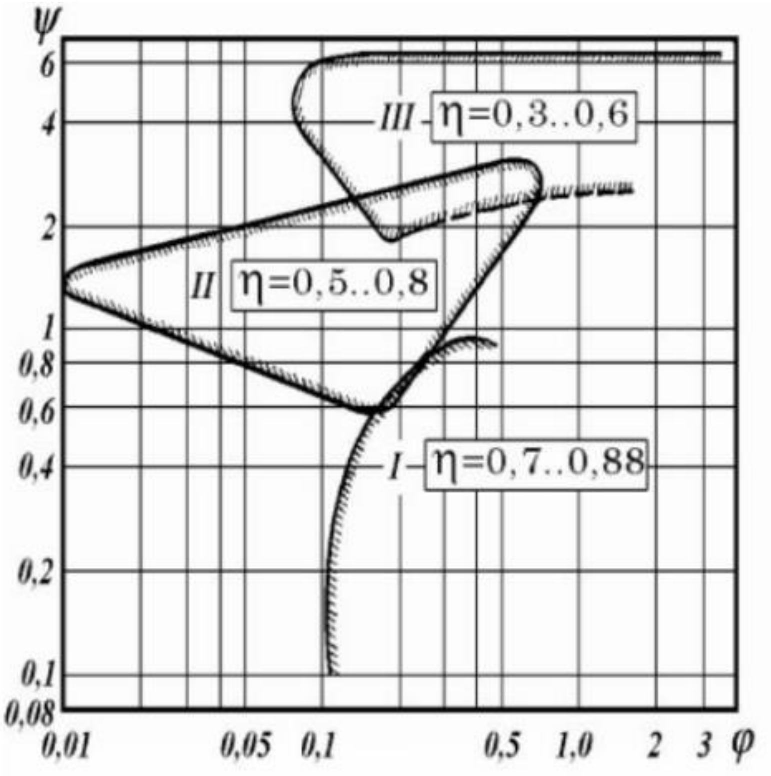

Figure 3.7 - Field of dimensionless parameters of fans of different types:

I - axial; II - radial; III - diameter.

Full and static fan efficiency:

$$\eta = \frac{p_t \cdot \dot{V}_{air,in}}{\dot{W}_v} \; ; \; \eta_s = \frac{p_s \cdot \dot{V}_{air,in}}{\dot{W}_v} \quad (2.11)$$

Where $\dot{W}_v$ - is the power consumed by the fan.

The fan's dimensional parameters when temperature changes (at a fixed speed) are determined by the following equation, where the index and denotes any operating mode, because the fan's dimensionless features are kept when external factors change their characteristics:

- Pressure $$p_i = p\frac{\rho_i}{\rho} \qquad (2.12)$$
- Power $$\dot{W}_{i,v} = \dot{W}_v \frac{\rho_i}{\rho} \qquad (2.13)$$

Where $$\rho_i = \rho\frac{293}{(273+T_i)}, \quad \rho = 1{,}2\,\frac{kg}{m^3} \qquad (2.14)$$

3.12. Description of heat exchange processes in HVAC

The following HVAC system components the heater and cooling chamber (vapour compression unit) perform heat exchange procedures that are based on the temperature of the air entering the HVAC system.

Heat transfer and convective heat transfer are the two main ways that heat exchange happens in nature, but convection which is only conceivable in moving media and in which the transmission of heat is coupled with the transfer of the medium itself represents the most intriguing situation. Heat conduction invariably follows heat convection because the passage of gas inevitably causes the collision of small particles with various temperatures.

The Newton-Richman equation yields the quantity of heat transferred during the heat transfer process:

$$\dot{Q} = \alpha \cdot (T_w - T_{in}) \cdot A, \qquad (2.15)$$

Where α - heat transfer coefficient, W/(m^2*K);

T_w, T_{in} - average temperatures of air at the inlet to the HVAC and walls of the of the heat exchanger, °C;

A - is the surface area of the wall through which heat is exchanged, m^2.

Numerous variables affect the heat transfer coefficient, including the type and modes of gas flow, the gas's physical characteristics, and the size and shape of the walls.

The α is calculated from the criterion equations created by using similarity theory transformation techniques to the differential equations of hydrodynamics and convective heat transport.

A criteria equation of the following form can be used to represent convective heat transfer under stationary conditions without affecting the overall state of matter, according to the principles of the theory of similarity:

$$Nu = f(Re, Pr, Gr...), \tag{2.16}$$

Where the value of the heat transmission is correlated with the Nusselt number, or Nu in the form of the following equation:

$$\alpha = \frac{Nu \cdot \lambda}{d}, \tag{2.17}$$

where d is the equivalent diameter, defined as 4 / A | l ;

A - is the cross-sectional area of the channel, m^2;

l - is the perimeter of the channel, m;

λ - is the thermal conductivity of a liquid at an average flow temperature, W/(m*K).

The flow mode is determined by the dimensionless complex Re:

$$Re = \frac{v_f \cdot d}{\upsilon_f} \tag{2.18}$$

Where V_f - flow velocity, m/s;

υ_f - is the kinematic viscosity of the flow at an average temperature, m^2/s.

The majority of HVAC systems run in laminar flow modes, which correspond to Re values of up to 2300, for both the coolant circulating in the heat exchanger heater and cooling chamber tubes and the air flow.

The following factors are also considered when considering laminar flows when using the dimensionless complex Gr:

$$Gr = \frac{g \cdot \beta \cdot d^3}{\upsilon^2} \Delta T, \tag{2.19}$$

where g is the acceleration of free fall, m/s^2 ;

β - coefficient of thermal expansion, K^{-1} ;

ΔT - is the temperature difference between the flow and the wall, K.

The following should be kept in mind for heat transfer using tubes: the assumption that the temperature distribution does not affect the moving medium's velocity distribution if it does not result in appreciable changes in the substance's viscosity when its temperature changes, since the temperature profile at the wall-moving medium interface has a specific gradient.

Although the idea of the heat transfer coefficient α is helpful, it does not truly let us get away from an issue that is intrinsically complex. The heat transfer coefficient α can be calculated using a number of different techniques. Analytical techniques will be employed for laminar flows, whereas integral techniques, mixing path theory, and dimensional analysis will be used for turbulent flows.

The following equation can be used to determine the thermophysical properties of air:

$$Nu = 0{,}26 \cdot Re^{0{,}65} \cdot Pr^{0{,}33} \cdot \left(\frac{Pr}{Pr_s}\right)^{0{,}25} \cdot \varepsilon, \tag{2.20}$$

Where ε - correction factor depending on the type of tube arrangement in the TA.

As the distance from the pipe entry grows, the local heat transfer coefficients in tubes tend to approach constant values. The inlet section's impact on long pipelines is minimal and frequently ignored in technical calculations.

It is widely understood that the boiling mechanism in the tube of a steam compression unit entails heat transfer from the solid to the liquid and then from the liquid to the surface of each expanding bubble.

Equation 2.15 presents a challenge because α has a large effect on temperature difference (sometimes up to the third order), but in HVAC systems, where the temperature range is often less than 50 K, the heat transfer coefficient can be regarded as constant.

Thus, the balance equation for the heating/cooling of the air flow and heat transfer through the TA's walls, accounting for the recovery of some of the heat, can be expressed as follows:

$$\dot{Q} = \frac{dH_a}{d} = \hat{c}_a \dot{m}_a (T_{out} - T_{in}) = \alpha (T_s - T_{in}) A_{TA}, \tag{2.21}$$

Where H_a – is the enthalpy of air flow, kJ/kg;

$\hat{c}_a$ – specific heat capacity of air, which is usually assumed to be constant and equal to 1 kJ/kg;

T_{out}, T_{in} – is the temperature at the outlet (inlet) of the HVAC system, K;

A_{TA} – heat exchange area of the heat exchanger, m^2.

3.13. Description of the refrigeration compressor

In most cases, dynamic characteristics are not taken into account when operating a compressor because polytropic compression is typically assumed, and the compression process in the compressor is assumed to be a steady-state process with sufficient accuracy as a steady-state process, unlike in the case of a condenser or expansion chamber.

If it becomes required to regulate the intensity of the heat exchange process, the compressor speed might be taken into consideration. In modeling, it is typical to assume that the compressor reaches the necessary speed instantly.

The flow rate λ, which can be modeled as the product of partial coefficients that indicate the losses, is what determines the reduction in refrigerant volume (volumetric losses):

$$\lambda = \lambda_d \cdot \lambda_{th} \cdot \lambda_W \cdot \lambda_{den} \cdot \lambda_o \tag{2.22}$$

Where λ_d – a coefficient that takes into account dead volume losses, 0.9;

λ_{th} – throttling and pulsation coefficient, 0.98;

λ_W – steam heating coefficient, 0.87;

λ_{den} – density coefficient, which takes into account the flow of working fluid through leaks, 0.97;

λ_o – a coefficient that takes into account the presence of oil in the refrigerant, 0.96.

3.14. Description of the electronic expansion valve (EEV)

Usually, a stepper motor and software are used to control an EEV.

It is possible to assume the electrical signal's transmission to be instantaneous when used to regulate a valve. As a result, the dynamic operations in an EEV are often described by the stepper motor's slow process of opening the refrigerant valve. Most of the time, an EEV's process is thought to be isentropic.

The well-known orifice leakage equation (Bernoulli's equation), which can be found here, can be used to calculate the mass flow rate of refrigerant through the EEV:

$$\dot{m}_{\mathrm{EEV}} = C_f A_{\mathrm{EEV}} \sqrt{\rho_3 \Delta\rho_{\mathrm{EEV}}}, \qquad (2.23)$$

Where A_{EEV} - is the cross-sectional area in the EEV, which is assumed to be linearly variable when the valve is opened, m^2;

$\Delta\rho_{\mathrm{EEV}}$ - is the difference in pressure at the inlet and outlet of the EEV, Pa;

ρ_3 - is the density of the refrigerant at the inlet to the EEV, kg/m^3;

C_f - is the flow coefficient determined from:

$$\mathrm{C_f} = 0{,}02005 \cdot \sqrt{\rho_d} + 0{,}634 \cdot \upsilon_4, \qquad (2.24)$$

Where υ_4 – is the specific volume of refrigerant leaving the EEV, m^3 /kg.

3.15. Annual variability of climate parameters

Changes in monthly averages derived from long-term studies best describe the annual pattern of climatic factors. In this instance, the distribution curves of the air parameters are rounded and nearly harmonious. This type of external climate change has enduring reasons, including a confluence of regional characteristics and radiation components that change over time.

The choice of the HVAC system for the annual cycle of operation is the most difficult step in the development process. It is well known that certain properties of technical equipment exhibit significant non-stationarity and nonlinearity with respect to the annual variation of outdoor air parameters. This largely influences the equipment control characteristics, which makes it much more difficult to choose methods and tools for automatic control of the working modes of technical equipment and systems.

Typically, June is the month with the highest annual outdoor temperature, and January is the month with the lowest. The variance in air humidity over a given year and frequently wind speed are correlated with ambient temperature.

It is possible to characterize changes in climate parameters as a function of the season in the form of a trigonometric series due to the harmony of these changes (Figure 3.8).

A series made up of the first two terms typically results in the change approximation being accurate enough:

$$y = y_{an,ave} + A_y \cos \bar{z}, \tag{2.25}$$

where $y_{an,ave}$ - average annual value of any climate parameter (temperature, enthalpy of the outside air, solar radiation intensity);

A_y – annual amplitude of its change;

$\bar{z}$ - is the relative time of the annual change in the parameter:

$$\bar{z} = \frac{2\pi z}{365}, \tag{2.26}$$

Where $z = z' - z^{max}$ - time since the maximum of the parameter, day;

z' $-$ time counted from 1 January, day;

z^{max} $-$ is the time of the maximum value of the parameter, counted from January 1, day.

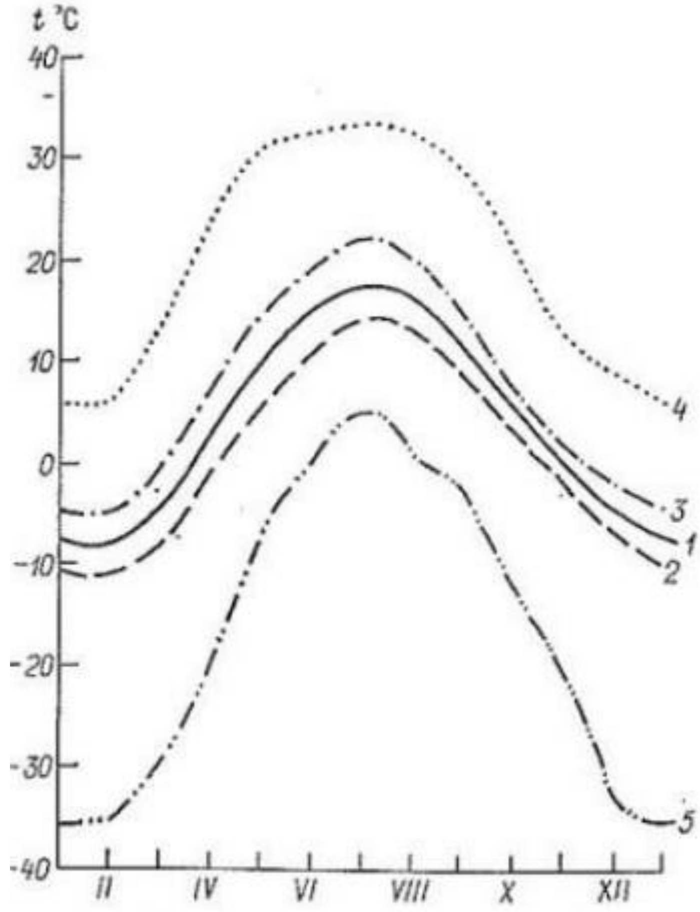

Figure 3.8 - Example of the distribution of outdoor air temperature over the annual cycle

where in Figure 3.8 are curves: 1 - average temperature during the year, 2 average minimum, 3 - average maximum, 4 - absolute temperature maximum, 5 - absolute temperature minimum.

Due to the trigonometric function cos's parity property, we still obtain a positive value for the climate parameter even if *z* turns negative.

CHAPTER 4
MODELLING THE OPTIMAL CONTROL SYSTEM OF HVAC OPERATING MODES BASED ON FEEDBACK

4.1. Identify the elements that significantly affect the HVAC system's energy usage

Modern automated ventilation (AV) and air conditioning (ACS) needs have two opposing conditions: the first is simplicity and dependability of operation; the second - high quality of functioning, for instance, quality of supply air temperature stabilization, minimal energy usage.

Finding a compromise in the technical solution is necessary in order to satisfy these needs.

The advancement of microprocessor technology has made it feasible to design operating modes for numerous pieces of equipment, including HVAC, with greater flexibility.

The simplest method of controlling a plant's operating modes is to turn it on and off while maintaining constant operating parameters, such as volume air flow rate $\dot{V}$ and the amount of heating (cooling) $\dot{Q}$ provided by the air flow during the planned mode for a specific time interval T. The use of automatic devices for controlling the parameters of operating modes, which include having the tasks of collecting, processing, transferring information, and generating signals and control signals, is another more sophisticated but capital-intensive method of regulation.

Electricity consumption is rationed depending on the energy characteristics of the equipment, showing how productivity $\dot{V}$ and the specific value of energy consumption $\dot{W}$ are related to each other. It is advisable to utilize correlation and regression analysis to modify the mathematical model based on physical laws, by evaluating the components as correlative values associated to energy consumption $\widehat{W}$, when creating a mathematical model of the plant's energy consumption.

For industries the issue of scheduling the energy consumption of an HVAC installation is crucial because it enables them to enhance the performance of their machinery while lowering energy overruns resulting from a lack of knowledge about the theoretically justified value of the amount of energy required to complete a particular work plan. Utilizing forecasting techniques like time series analysis and regression modeling, the operating modes of the HVAC plant should be planned.

When creating an HVAC model, it is necessary to separate the following HVAC system components that have the biggest effects on the installation's energy consumption in a given interval of changes to internal system parameters, as a specific vector value $\vec{X}_{sys} = \dot{m}_{p,p}, c_p, T_{in} \ldots$ relative to which the signal is generated control signal $\vec{u}$(t), which is expressed in the level of the model's abstraction from the real system.

In MIMO models, $\vec{X}_{sys}(t)$ is a component associated with the the input vector of parameters with which the automatic control device ACD operates to create a control signal $\vec{u}$(t), to change the parameters of the operation mode of the HVAC plant, the relationship between $\vec{X}_{sys}(t)$ and $\vec{u}$(t) is characterised by a transformation function:

$$Tr(t) = \frac{\vec{u}(\mathrm{t})}{(\vec{X}_{sys}(t) - \vec{X}_{sys,ref}(t))}, \tag{4.1}$$

Where $\vec{X}_{sys,ref}(t)$ - is a given vector of internal system parameters.

It is best to use correlation and regression analysis during the initial stages of the mathematical modeling of HVAC operation because they enable the identification of the most significant energy consumption factors and enable the model to be asked by eliminating the "ballast" elements of the vector $\vec{X}_{sys}(t)$ that are uninformative and only serve to exacerbate the mathematical model of HVAC.

You can precisely identify the component variables of the vector that most closely correlate with the energy consumption of the HVAC unit by creating a parameter vector for the HVAC $\vec{X}_{sys}(t)$ parameters that have the greatest impact on the installation. Using this method, you can benefit from both the advantages of mechanistic (based on physical laws) type of modeling and empirical modeling.

Table 3.1 displays the results of measurements of the HVAC unit's energy consumption W_c as a function of the average monthly outdoor air temperature T_o, the amount of water used for humidification, and the average hourly air flow rate $V_{ave.v}$.

A quantifiable measurement of the link between two variables under study, X and Y, is established via the correlation coefficient:

$$r_{Y,X} = \frac{\sum_{i=1}^{n}(y_i - \bar{y})(x_i - \bar{x})}{\sum_{i=1}^{n}(y_i - \bar{y})^2 \sum_{i=1}^{n}(x_i - \bar{x})^2} \tag{4.2}$$

Where y_i, x_i - is the value of the i-th sample element Y , X;

$\bar{y}, \bar{x}$ - average values of the samples Y , X;

i = {1, n }, n - sample size.

Table 4.1

Month	$T_{o,}$ °C	V_w, m^3	$V_{ave.v}$, $10^3 m^3/h$	W_c kWh
1	-6	105	3.4	14340
2	-4	97	3.2	13570
3	4	112	2.7	11779
4	12	115	2.3	9440
5	17	111	2.4	8988
6	22	108	2.7	9665
7	20	100	2.7	9449
8	17	130	2.4	9250
9	14	110	2.7	9410
10	9	127	2.9	10160
11	4	109	3.2	11350
12	-4	102	3.5	12799

In the case of an HVAC installation, the table of the installation's energy consumption is used to produce the matrix of correlation coefficients, which establishes a quantitative measure of the relationship between parameters:

$$r_{x_i,x_j} = \begin{vmatrix} r_{x_1,x_1} & r_{x_1,x_2} & \cdots & r_{x_1,x_n} \\ r_{x_2,x_1} & r_{x_2,x_2} & \cdots & r_{x_2,x_n} \\ \cdots & \cdots & \cdots & \cdots \\ r_{x_n,x_1} & r_{x_n,x_2} & \cdots & r_{x_n,x_n} \end{vmatrix} \quad (4.3)$$

By using the built-in statistical add-on "Data Analysis Package" in the home version of Microsoft Excel, determine the correlation coefficient values for the table (Table 4.2).

Table 4.2

	T_o	V_w	$V_{ave.v}$	W_c
T_o	1	-0.2232	-0.5421	-0.9294
V_w	-0.2232	1	0.25362	0.02064
$V_{ave.v}$	0.5421	0.25362	1	0.86364
W_c	-0.9294	0.02064	0.86364	1

The table shows that there is a significant inverse link between the value of consumption W_c and the outside air temperature T_o as well as a significant direct association with the average fan performance $V_{ave.v}$.

Since the value of air humidity throughout the year is fairly equally distributed, the value of water consumption V_w has less of an impact on the dynamics of variations in energy consumption. Instead, water consumption is more closely tied to the value of fan performance $V_{ave.v}$.

As a conclusion, it should be mentioned that it is advisable to take into account the variables relating to the mechanical and thermal loads placed on the fan motor.

4.2 Calculation of the energy characteristics of the steam compressor

We employ the CoolPack software, a set of models for simulating the functioning of refrigeration systems, for the calculation. The air temperature drop values for evaporators that cool air can typically be assumed to be $\Delta T_{air} = 6–10$ °C, and the total difference between the refrigerant temperature and the inlet air temperature is $\Delta T_{air} = 16–20$ °C.

In air conditioning, the condensation temperature should be 10-15°C higher than the ambient temperature, and the air provided to the room from the air conditioner should have a positive temperature. Therefore, a boiling point of +5 °C and a condensation temperature of +40 °C are chosen.

The most popular refrigerant, R22, is utilized in the refrigeration chamber. Figures 4.1 and 4.2 show the description of the initial data and the results of the cycle construction. Point 1 describes the subcooled refrigerant's condition at 6 °C. Throttling is described in Process 1-2, and in this model, an isentropic pressure reduction process i.e. one without losses—is assumed. The refrigerant then evaporates in the evaporator (2–3), overheating as a result so that the compressor can function on a "dry run". Up until the evaporator outlet, where the steam is overheated by 5-8 °C above the boiling point, heat is evacuated from the system.

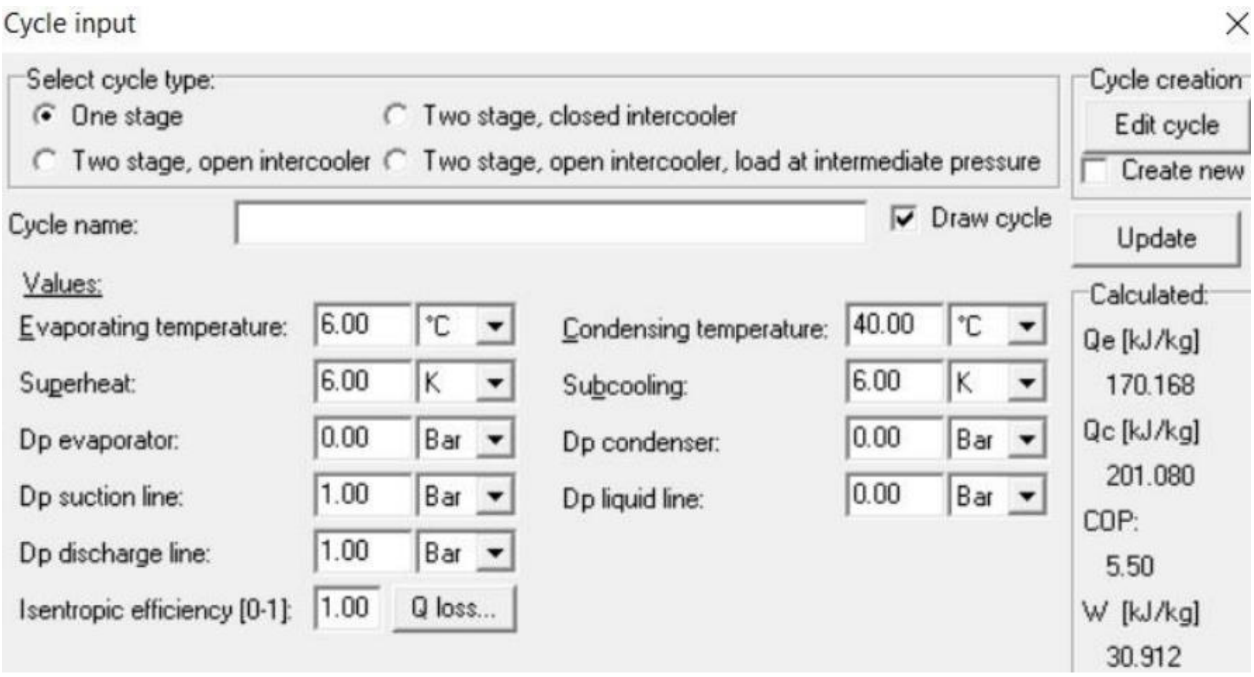

Figure 4.1 - Description of the input values for the refrigeration cycle

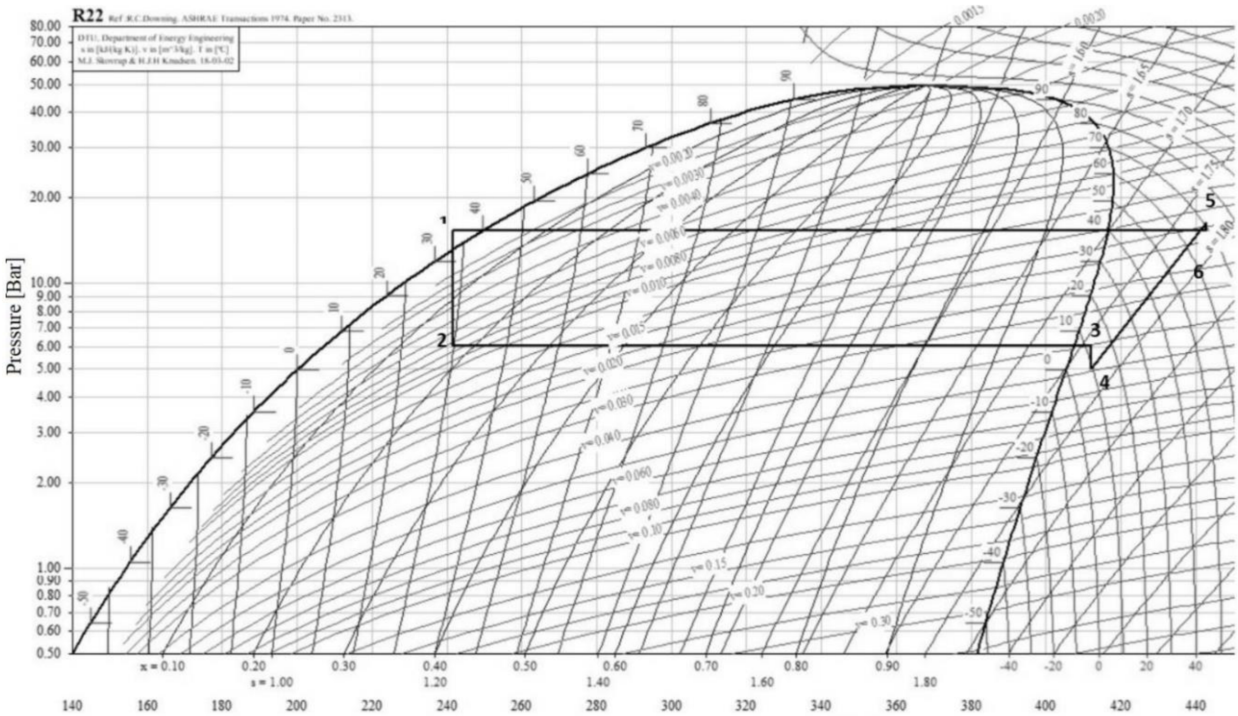

Figure 4.2 - Refrigeration unit cycle in the CoolPack software.

The losses at the compressor inlet are considered in links 3–4 and 5–6, while outlet losses are only considered in links 5–6. As the refrigerant pressure rises, the condenser starts to condense, which is followed by subcooling (step 6).

There is a 5 to 10 degree differential in temperature between the air and the condenser. Condensing temperatures need to be 10–15 °C higher than ambient temperatures.

Additionally, there is a pressure loss at the compressor's inlet, which affects the pressure drop there. Additionally, the compressor needs to generate more overpressure than is necessary for condensation because of pressure losses at its exit. The cycle's overall efficiency is decreased as a result of the additional compression work required to make up for these losses.

The amount of heat released by the condenser is:

$$\hat{Q}_c = \hat{Q}_0 + \widehat{W} \tag{4.4}$$

$where\ \hat{Q}_0$ - is the amount of heat absorbed by the refrigerant in the evaporator, kJ/kg;

$\widehat{W}$ - is the amount of work performed by the compressor, kJ/kg.

The heat transferred by the condenser $\hat{Q}_0$ to the work done $\widehat{W}$ is expressed as the refrigeration coefficient:

$$\varepsilon = \frac{\hat{Q}_0}{\hat{W}} \tag{4.5}$$

As a 1 °C rise in evaporation temperature results in a 3-5% increase in cooling capacity, the evaporation temperature should be chosen as high as possible.

Since a 1°C decrease in condensation temperature results in a 1% improvement in cooling capacity, the gap between condensation temperature and ambient temperature should be as little as possible.

The value of the compressor work per unit mass of refrigerant $\hat{W}$ and the refrigeration coefficient ε are obtained from the calculation results:

$$\varepsilon = 5{,}5$$

$$\hat{W} = 30{,}9\ kJ/kg$$

This means that the total cooling capacity of the refrigeration unit is:

$$\dot{Q}_{cl} = \dot{m}_{R22} \cdot \varepsilon \cdot \hat{W}, \tag{4.6}$$

Where $\dot{Q}_{cl}$ - is the amount of heat extracted per unit time, kW;

$\dot{m}_{R22}$ – mass flow rate of refrigerant, kg/s.

Dead space must be accounted for in the calculation. The piston pumps significantly less refrigerant than necessary due to hydraulic losses and refrigerant leakage through the O-rings.

4.3 Approximation of fan aerodynamic characteristics by a mathematical function

Based on the experimental data, we have the following values of the fan efficiency $\eta_f(\dot{V}_f)$ and the total pressure generated by the fan $p_{t.f}(\dot{V}_f)$ (Table 4.3 and Figure 4.3).

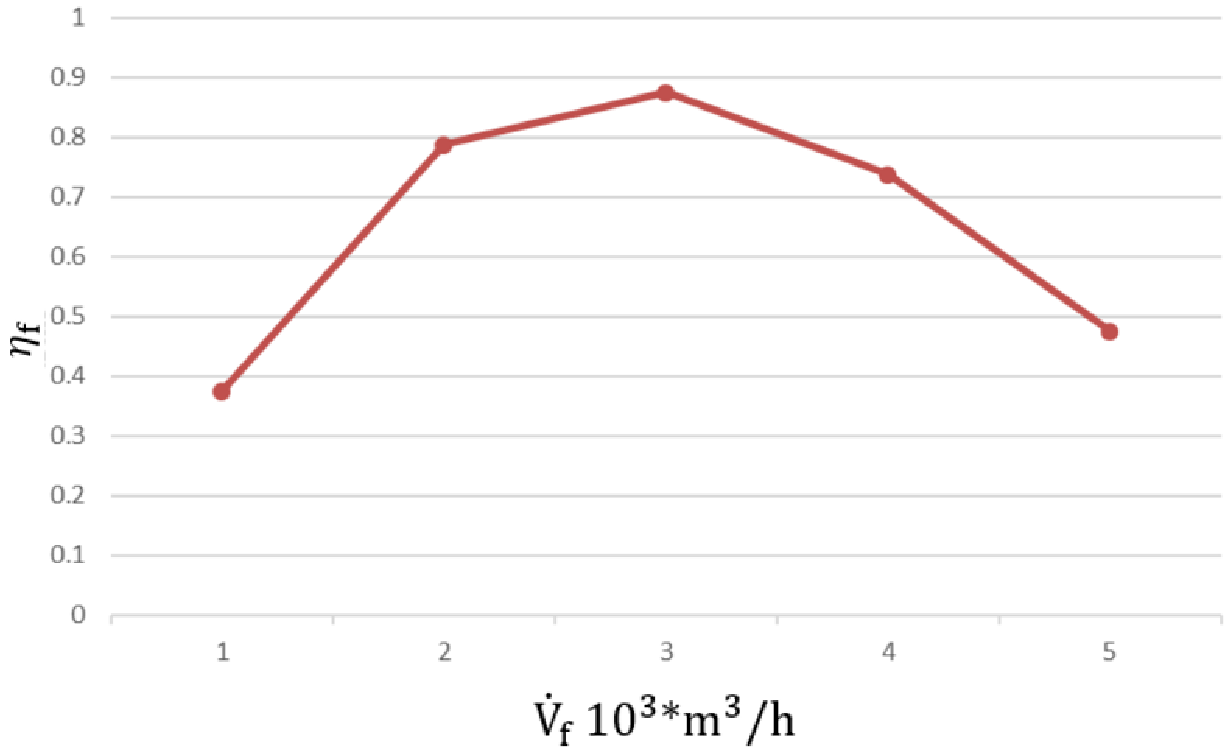

Figure 4.3 - Changes in fan efficiency as a function of changes in fan performance

In general, a 2nd order polynomial regression equation can be used to express the power consumption of a fan $\dot{W}_f(\dot{V}_f)$ for a specific performance range:

$$\dot{W}_f = \frac{(a_0+a_1\cdot\dot{V}_f + a_2\cdot(\dot{V}_f)^2)\cdot\dot{V}_f}{a_3+a_4\cdot\dot{V}_f + a_5\cdot(\dot{V}_f)^2} \quad (4.7)$$

Table 4.3 - Values of fan efficiency $\eta_f(\dot{V}_f)$ (Figure 4.3) and the total pressure generated by the fan $p_{t.f}(\dot{V}_f)$ (Figure 4.4).

$\dot{V}_f, 10^3 * m^3/h$	1	2	3	4	5
$\eta_f(\dot{V}_f)$	0,36	0,77	0,86	0,72	0,46
$p_{t.f}(\dot{V}_f)$, Pa	750	690	600	450	170

As a result, we will use the MATLAB software's built-in function to calculate the coefficients for the equation (see APPENDIX A).

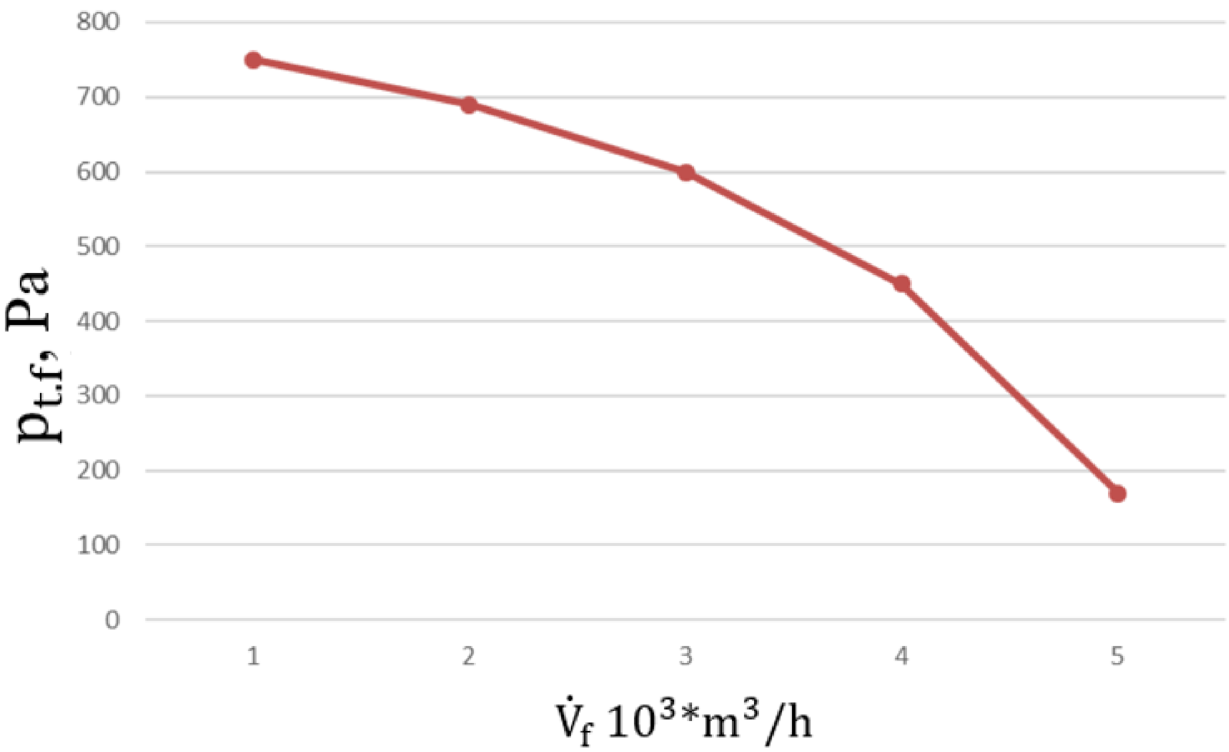

Figure 4.4 - Changes in total fan pressure as a function of changes in fan performance

Let's get the coefficient values for the regression equation describing the power consumption of fans (Table 4.4).

Table 4.4 - Values of the coefficients of the regression equation of the consumed power consumption by the fan

a_0	a_1	a_2	a_3	a_4	a_5
703	74,3	-35,8	0,19	0,67	-0,108

Accordingly, the regression equation will look like this:

$$\dot{W}_f = \frac{(703+74{,}3\cdot\dot{V}_f - 35{,}8\cdot(\dot{V}_f)^2)\cdot\dot{V}_f}{0{,}19+0{,}67\cdot\dot{V}_f - 0{,}108\cdot(\dot{V}_f)^2} \tag{4.8}$$

where the fan performance $\dot{V}_f$ changes accordingly in the permissible interval of 0.15 $\cdot\, \dot{V}_{air,max} < \dot{V}_{air,in} < \dot{V}_{air,max}$.

Let's compare the regression equation for fan power consumption $\dot{W}_f(\dot{V}_f)$ with the experimental values of Table 4.3, - Appendix A.

Table 4.5 displays the values that the equation's output yielded as results. The following table shows the relative difference between the actual values of total fan pressure $p_{t.f}(\dot{V}_f)$, and fan efficiency $\eta_f(\dot{V}_f)$ and the values that the regression equation theoretically yields:

$$M_{P(\dot{V})} = \frac{100}{n}\sum_{i=1}^{n}\left|\frac{p_{f,a,i}(\dot{V})-p_{f,e,i}(\dot{V})}{p_{f,a,i}(\dot{V})}\right| \tag{4.9}$$

$$M_{\eta(\dot{V})} = \frac{100}{n}\sum_{i=1}^{n}\left|\frac{\eta_{f,a,i}(\dot{V})-\eta_{f,e,i}(\dot{V})}{\eta_{f,a,i}(\dot{V})}\right| \tag{4.10}$$

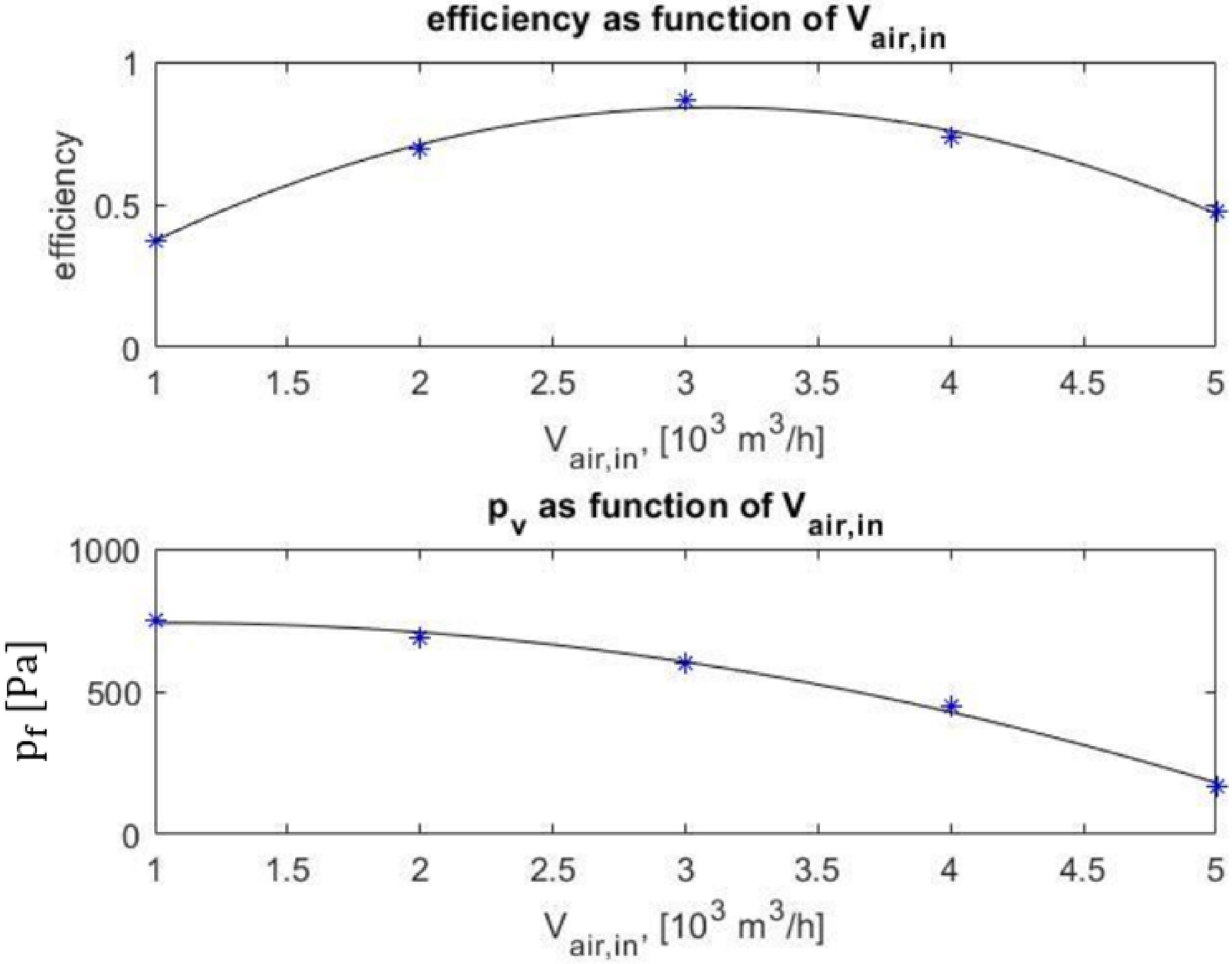

Figure 4.5 - Graphical representation of the regression equation and actual values for the fan

Table 4.3 - Theoretically approximated values of fan parameters in compared to the experimental ones

$p_{f,a,i}(\dot{V}), Pa$	750	690	600	450	170
$p_{f,e,i}(\dot{V}), Pa$	740	707	603	427	180
$\eta_{f,a,i}(\dot{V})$	0,36	0,77	0,86	0,72	0,46
$\eta_{f,e,i}(\dot{V})$	0,37	0,71	0,83	0,75	0,46

Figure 4.6 depicts a graph of the theoretical power consumption $\dot{W}_f$ of a radial fan as a function of performance $\dot{V} = \{1 \leq \dot{V} \leq 3.5\}$.

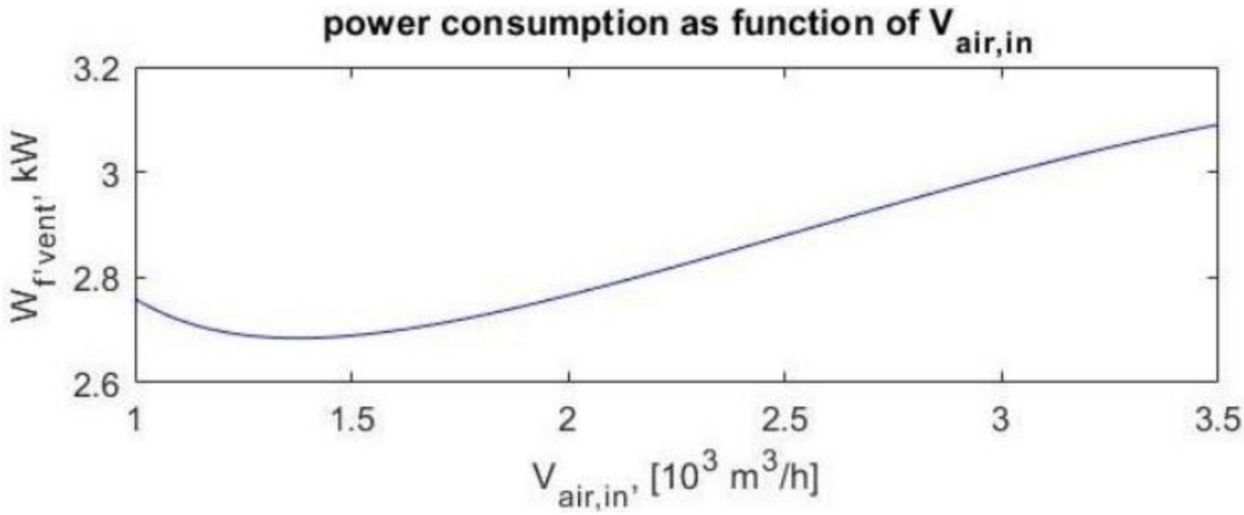

Figure 4.6 - Theoretical power consumption by the fan at

$$\dot{V} = \{1 \leq \dot{V} \leq 3.5\}$$

According to the values in the table, the relative errors for $p_{t,f,i}(\dot{V}_f)$, and $\eta_f, t, i(\dot{V}_f)$ will be:

$$M_{P(\dot{V})} = 3\%$$

$$M_{\eta(\dot{V})} = 4\%$$

4.4 Modelling of the HVAC operation mode

As a starting point, we assume that the level of maximum production in an industrial premises is respectively $\dot{m}_{p,p} = 0{,}146\ g/s$ because industrial premises typically include a pollution source with a particular amount of pollutant output. The concentration of a specific pollutant type $c_p = 0{,}007955\ mol/m^3$ at the same time as the concentration of a different pollutant type c_p in the volume of a vented room is the same value is considered to be the maximum allowable and is not affected by how the concentration is distributed throughout the room's volume V.

Accordingly, the material balance equation for the pollutant will be as follows:

$$\frac{dm_p}{dt} = \dot{m}_{p,in} - \dot{m}_{p,out} \dot{m}_{p,p} \tag{4.11}$$

Where $\dot{m}_{p,in} - \dot{m}_{p,out}$ - mass flow of the pollutant with the input (output) flow from the ventilation system;

m_p - mass of the pollutant in the volume of the industrial premises.

We also have the following formulae for the volume of air in the room under the assumption of constant room pressure and disregarding the thermal expansion of air:

$$\frac{dV_{air}}{dt} = 0 \tag{4.12}$$

$$\dot{V}_{air,in} = \dot{V}_{air,out} \tag{4.13}$$

Where V_{air} - is the air volume in the room, which in this case is 2000 m3 (length, breadth of the working area is 20 m, and the height of the room is 5 m), estimated as the interior of the industrial premises;

$\dot{V}_{air,in}$, $\dot{V}_{air,out}$ - inlet (outlet) air flow from the ventilated room.

Equation 4.11 can also be expressed in terms of air parameters:

$$\frac{dm_p}{dt} = \dot{V}_{air,in} \cdot (c_{p,in} - c_{p,out}) + \dot{m}_{p,p} \tag{4.14}$$

where the concentration of the pollutant in the outgoing air stream $c_{p,out}$ corresponds to to the concentration of the pollutant in the ventilated room c_p.

For the stationary case $\frac{dm_p}{dt}$ =0 the required performance of the ventilation system is:

$$\dot{V}_{air,in} = \frac{\dot{m}_{p,p}}{M_p c_p} \tag{4.15}$$

Where M_p - molar mass of the pollutant, which is 44 mg/mol.

The next step in modeling is to specify the room's temperature $T_{i.n}$ while accounting for heat losses $\dot{Q}_{loss,i}$ through the i-th constituent of the room, which are represented in turn as follows:

$$\dot{Q}_{loss,i} = U_i \cdot A_i \cdot (T_{i,n} - T_i) \tag{4.16}$$

Where U_i - is the heat transfer coefficient through the i-th element of the room;

A_i - is the surface area of the i-th element of the room through which heat is exchanged.

Changes in room temperature $T_{i,n}$ can be described by the following equation:

$$\hat{c}_{\mathrm{air}} m_{air,ins} \frac{dT_{i,n}}{dt} = V_{air,in} \rho_{air} \hat{c}_{\mathrm{air}} (T_{in} - T_{i,n}) - \sum U_i \cdot A_i (T_{i,n} - T_i) + q(t), \tag{4.17}$$

$$\hat{c}_{\mathrm{w}} m_w \frac{dT_w}{dt} = U_w A_w (T_{i,n} - T_w) - U_w A_w (T_{i,n} - T_o) \tag{4.18}$$

$$\hat{c}_{\mathrm{ce}} m_{ce} \frac{dT_{ce}}{dt} = U_{ce} A_{ce} (T_{i,n} - T_{ce}) - U_{ce} A_{ce} (T_{i,n} - T_o) \tag{4.19}$$

Where $m_{air,ins}$ - mass of air in the room where ventilation and air conditioning;

ρ_{air} - - the room's air density, which is taken as a constant because air's thermal expansion is negligible. It is equal to: $\rho_{air} = 1{,}2 \frac{kg}{m^3}$;

q(t) - heat input from people, lighting, and installations;

w, ce - indices that correspond to the wall and ceiling.

The fan capacity $\dot{V}_{air,in}$ corresponds to the minimum required flow rate to remove the pollutant and maintain it at the level of the maximum allowable concentration in the first scenario, where the HVAC system runs in the mode of balancing the heat loss $\dot{Q}_{loss,i}$ and heat input from the HVAC heater.

The room's initial temperature is $T_{i,n}$= 10 °C. We will simulate the idle model in the "OpenModelica" software, where the mode of operation corresponds to the existing system characteristics without employing forecasting techniques or creating typical load graphs. Appendix B shows the program's listing.

Graphs of changes in room temperature and input flow from the HVAC system are shown in Figures 4.7.a and 4.7.b

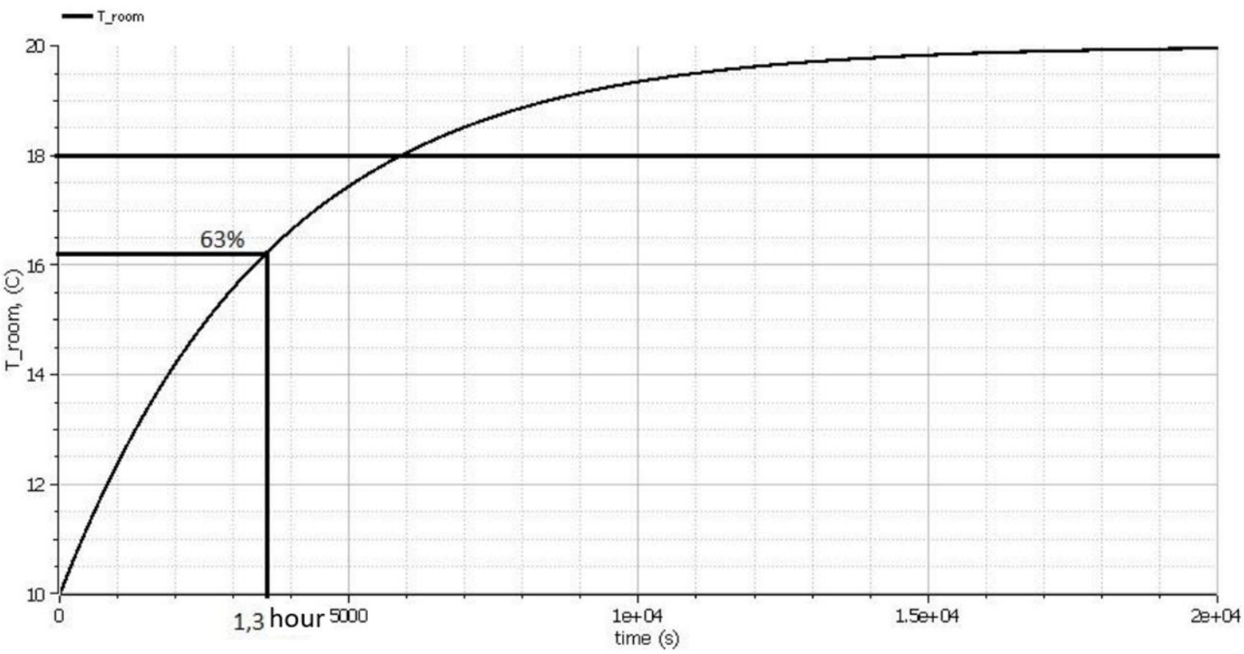

Figure 4.7.a - Change in the internal temperature of the room during the first six hours of the simulation

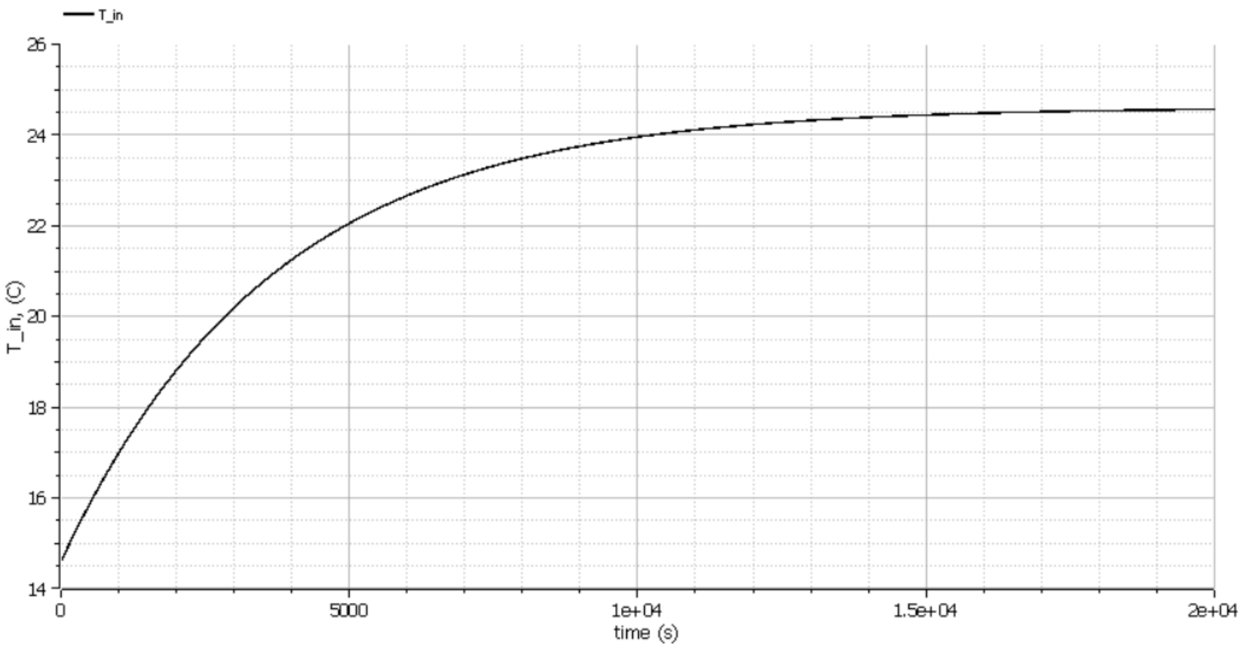

Figure 4.7.b - Temperature change of the inlet air flow after heating in the heater during the first six hours of the simulation

The appropriate indicators are provided in Table 4.4. As a result of the simulation, it is necessary to indicate the indicators of energy consumption and power consumption at a constant ambient temperature T_o = -1 °C for a constant period T_s = 1.3 hours.

Table 4.4 - Power and energy consumption by the heater and fan of the plant during the time constant T_s = 1.3 hours.

$\dot{Q}_h$, kW	Q_h, kWh	$\dot{W}_f$, kW	W_f, kWh	$\dot{W}_{tot}$ kW	W_{tot}, kWh
7,1	9,46	4,83	6,309	13,2	15,86

According to the simulation results, it takes the system 1.3 hours to achieve a 63% reduction in the difference between the working room's initial temperature and the set temperature. As a result, with this kind of control, the system takes a long time to attain the desired temperature without experiencing overruns.

4.5 Modelling of the HVAC operation mode using automation tools

Equation 4.1, which may be put in a more visual form and is typically used to regulate systems whose environmental characteristics change over time:

$$\vec{e}(t) = \vec{X}_{sys}(t) - \vec{X}_{sys,ref}(t) \tag{4.20}$$

$$\vec{T}r(t) = \frac{\vec{u}(t)}{\vec{e}(t)} \tag{4.21}$$

Where $\vec{e}(t)$ - a vector with the values of the differences between the specified system parameters and the current ones.

The best form of control $\vec{T}r_{opt}(t)$ for a given vector of system parameters $\vec{X}_{sys}(t)$ and time constraints for obtaining the system state $\vec{X}_{sys,ref}(t)$ is expressed as the optimization of the plant control process as follows T_{lim}:

$$\vec{T}r_{opt}(t) = f(\vec{e}(t), T_{lim}) \tag{4.22}$$

Optimization can be carried out according to the energy consumption of the plant W_{pl} depending on the type of conversion function:

$$W_{pl}(Tr(t)) \rightarrow min \quad (4.23)$$

When only one system parameter is being controlled, such as the temperature of the indoor air, a proportional-integrating controller is employed to do it. This parameter's discrete form is described as follows:

$$e_i = T_{ref,ins} - T_{ins,i}$$
$$u = K_p + Z_{i-1}$$
$$Z = Z_{i-1} + \Delta t \frac{K_p}{T_i} \cdot e_i \quad (4.24)$$

Where K_p = const - proportional component of the controller;

T_i = const - integrating component of the controller.

For instance, increasing the air heating capacity of the heater will speed up the process of achieving the desired temperature (Figure 4.8), but this will also take more energy.

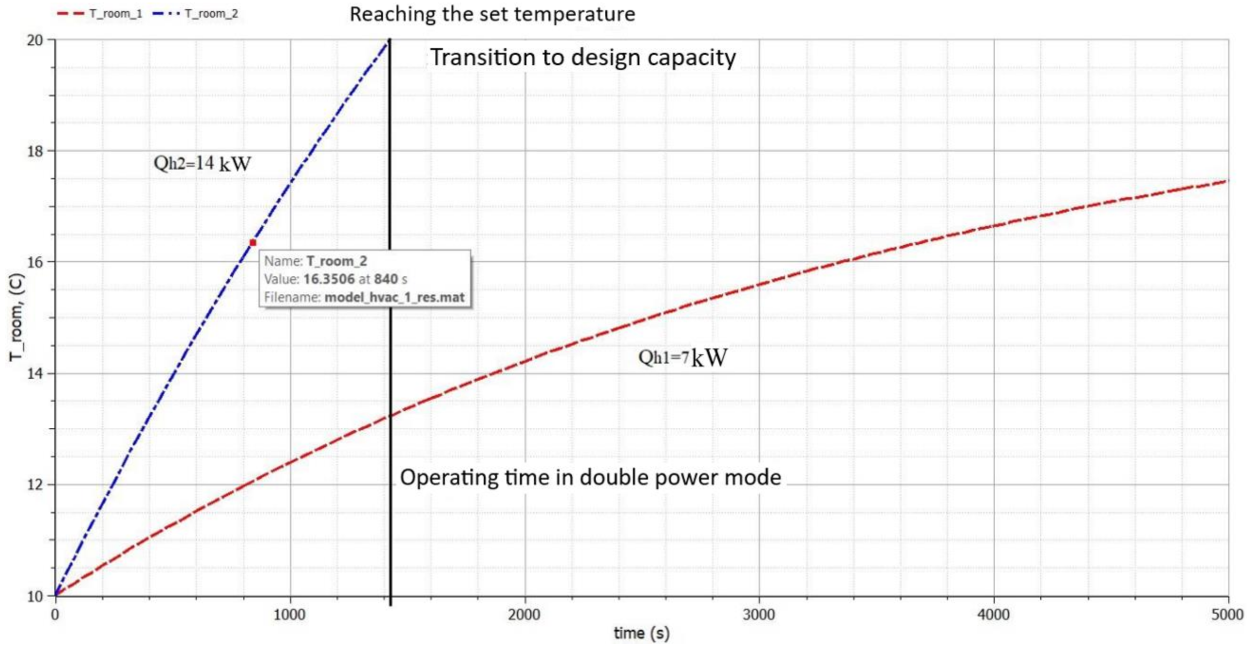

Figure 4.8 - Graphical representation of the effect of increasing power increase on the speed of reaching the set temperature

The most cost-effective method for determining the operating mode of the HVAC system is to identify the design capacity at which the specified values of the working space's parameters are achieved as cost-effectively as possible, and then to adjust it further based on the system's maximum allowable time to do so.

The time to reach the set values decreases with each increment when the power is increased in steps of 20% of the design value (Figure 4.9), but relatively minor increases in power greatly shorten the time to reach the set characteristics

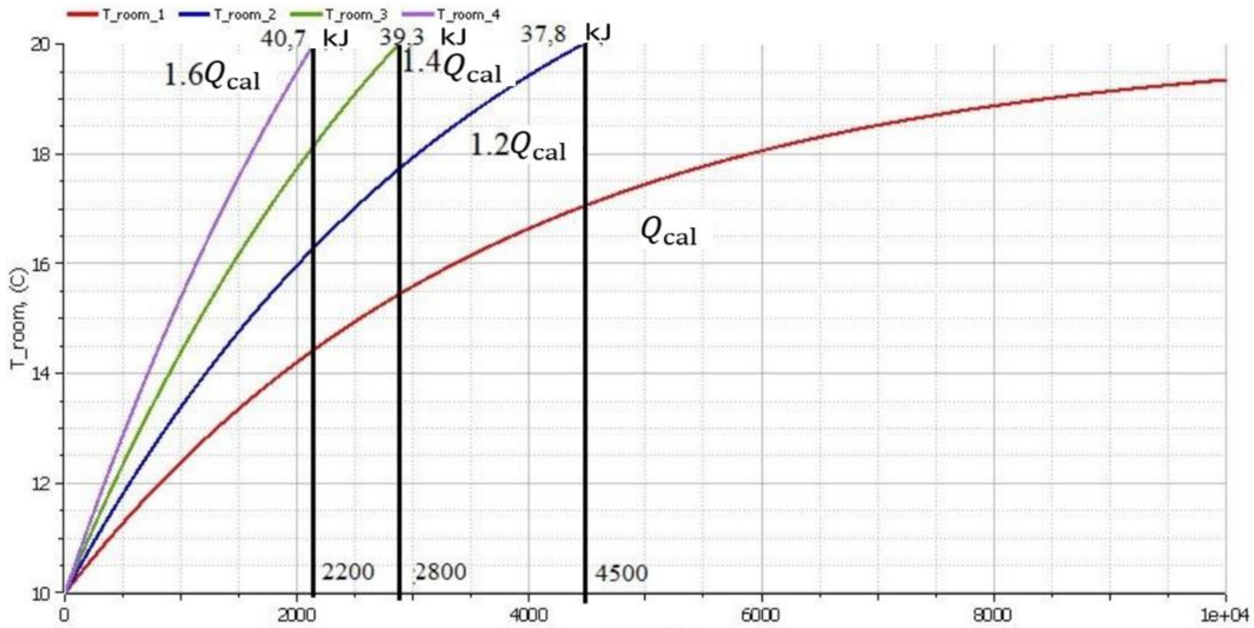

Figure 4.9 - Different duration of reaching the set temperature depending on the power value

CONCLUSIONS

1. It is a difficult work to optimize ventilation and air conditioning systems, and the solution should take into account economic, climatological, and aerodynamic heating aspects. Technical and functional indicators should be used to evaluate the ventilation system's technical level, effectiveness, and quality. A radial fan is one of the most crucial components of the industrial ventilation system since it delivers the necessary system performance. Significant heat and electrical energy losses are caused by high energy consumption and ineffective performance and air parameter adjustment depending on the load. As a result, it becomes crucial and necessary to research and improve the radial fan's specifications in order to offer changeable air flow.

2. It is essential to use a combined modeling approach when simulating and analyzing the performance of the HVAC system because it enables evaluation of the model's accuracy based on physical laws and adjustment of its components using empirical techniques (regression analysis) using measurements from a real object. Determining the timeline for ways of controlling operating modes to achieve optimal energy performance is the most difficult aspect of the development process. It is well known that certain features of technical equipment exhibit significant non-stationarity and nonlinearity depending on the annual variation in outdoor air conditions.

3. The issue at hand was resolved by analyzing the key factors that have a significant impact on the energy consumption of the HVAC system. This was accomplished by constructing a correlation coefficient matrix using the "Data Analysis Package" in MS Excel. To simplify the model, particular attention was given to the HVAC components that consume substantial amounts of energy. This involved approximating the fan characteristics and determining the heating/cooling characteristics of the airflow from the HVAC system.

The operational mode of the system was simulated during the calculations, taking into account the required airflow rate and the characteristics of the transient process associated with temperature changes resulting from alterations in the amount of air being heated. The time it takes to achieve the desired parameters within the working space is influenced by additional power consumption, which exhibits a nonlinear nature.

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APPENDIX A

Listing of the program MATLAB for finding the coefficients of the regression equation

```
V = [1, 2, 3, 4, 5] ;
x = 1:0.01:5 ;
y1 = [0.375, 0.6975, 0.865, 0.7375, 0.475] ;
y2 = [750, 690, 600, 450, 170] ;
pl = polyfit (V, y1,2);
p2 = polyfit (V, y2,2);
p_3 = polyval (p1, V);
p_4 = polyval (p2, V);
```

APPENDIX B

Listing of the description of the HVAC model in the OpenModelica software in stationary operating conditions

```
    parameter Real a=20 "Length of industrial premises, m";
    parameter Real b=20 "Width of industrial premises, m";
    parameter Real h=5 "Height of industrial premises, m";
    parameter Real A_w=2*(a+b) *h "Wall surface area, m2";
    parameter Real A_r=a*b "Ceiling surface area, m2";
    parameter Real V=a*b*h "Internal volume of the room or
volume of
    air volume, m3";
    parameter Real R_w=1.3 "Specific thermal resistance of
the wall, m2*K/W";
    parameter Real R_r=1.1 "Specific thermal resistance of
the ceiling, m2*K/W";
    parameter Real w_w=0.93 "Wall thickness, m";
    parameter Real rho_w=1200 "Wall material density, kg/m3";
    parameter Real m_w=2*(a+b) *h*w_w*rho_w "Wall mass for
determining
    thermal inertia, kg";
    parameter Real w_r=0.8 "Ceiling thickness, m";
    parameter Real m_r=a*b*w_r*rho_w "Wall mass for
determining
    thermal inertia, kg";
    parameter Real c_p_w=780 "Specific heat capacity of the
wall material, J/(kg*K)"
    parameter Real rho_air=1.23 "Air density, kg/m3"; ;
    parameter Real m_air=rho_air*V "Mass of the air in the
room, kg";
```

```
    parameter Real c_p_air=1000 "Specific heat capacity of
air, J/(kg*K)";
    parameter Real c_gdk=0.007955 "Maximum permissible
concentration of
    of the pollutant, mol/m3";
    parameter Real M_p=0.044 "Molar mass of the pollutant,
kg/mol";
    parameter Real m_f_g_p=0.00087 "Amount of pollutant
generation, kg/s";
    parameter Real eps=5.5 "Refrigeration coefficient of
cooling equipment";
    parameter Real kkd_recup=0 "Air recovery rate";
    parameter Real T_ref=20 "Set indoor air temperature, C";
    parameter Real kkd_ed=0.8 "Fan and motor drive
efficiency";
    parameter Real T_outd=-1 "Outdoor air temperature, C";
    Real T_room(start=10, fixed=true) "Initial air
temperature in the
    the working room, C";
    Real T_in "Temperature of the inlet air flow, C";
    Real V_f_in "Air flow rate, m3/s";
    Real Q_f_h_loss "Total heat loss through walls and
ceiling, kW"
    Real Q_f_h "Calculated heat flux, kW";
    Real W_f_vent "Power consumption by the fan, kW";
    Real W f tot "Total energy consumption, kW";
    equation
    V_f_in=m_f_g_p/M_p/c_gdk;
    Q_f_h_loss=1/R_w*A_w*(T_room-T_outd)+1/R_r*A_r*(T_room-
T_outd);
```

```
    Q_f_h=(1/R_w*A_w*(T_ref-T_outd)+1/R_r*A_r*(T_ref-
T_outd));
    Q_f_h=V_f_in*rho_air*c_p_air*(T_in-T_room);
    W_f_vent=1/3600*V_f_in*(-
35.7*V_f_in^2+74.28*V_f_in+702)/
    (-0.1073*V_f_in^2+0.6616*V_f_in+0.184)/kkd_ed;
    der      (T_room)=(V_f_in*rho_air*c_p_air*(T_in-T_room)-
Q_f_h_loss)/(c_p_air*m_air);
    W_f_tot=Q_f_h/1000+W f vent;
```