Solution of HVAC System in Clean Spaces in Hospitals

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Diploma thesis

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MASTER'S THESIS ASSIGNMENT

(PROJECT, ARTWORK, ARTISTIC PERFORMANCE)

Degree, First Name and Surname:	**Bc. Manh Ha Nguyen**
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Thesis Guidelines:

1. **Analyse problems in clean spaces generally and in hospitals particularly (especially in hospitals, operation theaters and clean laboratories).**
2. **Based on the anylysis specify the requirements on building physics and HVAC systems and equipment.**
3. **For the specified clean space in a hospital check the building parameters and resolve the building structure parameters if required.**
4. **Design the HVAC system in particular clean space.**
5. **Design the control system of HVAC with required visualization (SCADA).**

Thesis Extent:

Appendices:

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Bibliography:

1. **ASHRAE Handbook – HVAC applications (SI) 2007**
2. **ASHRAE Handbook – HVAC System & Equipment 2008**
3. **ASHRAE Fundamentals 2009**
4. **Cleanrooms and associated controlled environments Part 1–7**

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ABSTRACT

Scope of this Thesis is designing a HVAC system for a hospital, with focus on supplying clean air, provision of comfort for doctors, nurses and patients. This Thesis also describes and uncovers thermal-technical parameters of the hospital to choose suitable equipments for HVAC system. Moreover, this Thesis explains the way of controls, monitoring and communication using KNX bus.

Keywords: integrated building system, HVAC system, clean space, hospital, energy efficiency,building physics, HEPA filter, KNX.

ABSRAKT

Předmětem této práce je návrh system klimatizace pro nemocnici, se zaměřením nadodá v kučerstvého čistého vzduchu do čistých prostor, s opatřením pro zabezpečení komfortu pro lékaře a pacienty. Tato práce popisuje a vysvětluje tepelně technické vlastnosti budov, v tomto případě nemocnice, podle kterých jsou následně vybírána zařízení a prostředky pro HVAC systémy. Mimo to, tato práce vysvětluje způsoby řízení, monitorování a komunikace s využitím KNX sběrnice.

Klíčováslova: integrované systémy v budovách, systémy vnitřního prostředí, čisté prostory, nemocnice, energetická účinnost, stavební fyzika, HEPA fitrace, KNX.

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CONTENT

INTRODUCTION

Issues related to provision of supplying clean air for clean spaces, especially for the hospital. For the hospital, there are some special requirements. The temperature in operating room must be the same in winter and summer, and the speed of air in working point (1.5 m from the floor) must be less than 0.5 m/s. The pressure inside hospital must be greater than outside to avoid dirty air from outside. Hospital also requires more fresh air than other places that means we have to calculate and design a suitable HVAC system, which can handle this problem.

Current Thesis analyzes requirements of building physics and requirements of designing HVAC system for the hospital. From that we can design a HVAC system, which can undertake duty of supplying clean air in summer and winter for the hospital.

Firstly, theoretical part of the Thesis provides information of thermal parameters, sucks as: thermal conductivity, thermal transmittance and thermal resistance. It also provides the definition of heat loss, cooling load of the building, and the way how the calculate them. Moreover, the Thesis gives us the definition of clean room, classifications of clean room, HVAC system with its components. Finally, the theoretical part provides the requirement of building physics and HVAC system for the hospital.

Practical part contains calculations of the hospital physic parameters, and actual design of HVAC system. In order to satisfy requirements of HVAC system for the hospital, suitable equipments are chosen. Finally, the practical part describes the design of control system, which uses KNX-based intelligent system.

I. THEORATICAL PART

1 THERMAL PROPERTIES OF MATERIALS

Every material used in an envelope assembly has fundamental physical properties that determine their energy performance like conductivity, resistance, and thermal mass. Understanding these intrinsic properties will help you chose the right materials to manage heat flow.

1.1 Thermal Conductivity

Thermal conductivity, λ (also known as Lambda), is the rate at which heat passes through a material, measured in watts per square meter of surface area for a temperature gradient of one Kelvin for every meter thickness.This is expressed as W/(m.K). Thermal conductivity is not affected by the thickness of the product. [1]

Each material has a characteristic rate at which heat will flow through it. The faster heat flows through a material, the more conductive it is.

Example:

PIR Board: λ = 0.022 W/(m.K)

Glass Fibre Roll: λ= 0.044 W/(m.K)

Thermal conductivityis a material property given for homogeneous solids under steady state conditions.

It is used in the follow equation:

$$q = \frac{\lambda * A * \Delta T}{s} \quad (1)$$

Where:

- q *: resultant heat flow [W]*

- ***λ*** *: thermal conductivity of the material [W/(m.K)]*
- ***A*** *: the surface area through which the heat flows [m²]*
- Δ***T*** *: the temperature difference between the warm and cold sides of the material [K]*
- ***s****: the thickness / length of the material [m]*

1.2 Thermal Transmittance And Thermal Resistance

Thermal transmittance, also known as ***U***-value, is the rate of transfer of heat through a structure (which can be a single material or a composite), divided by the difference in temperature across that structure. The units of measurement are W/(m².K). The better-insulated a structure is, the lower the ***U***-value will be. Workmanship and installation standards can strongly affect the thermal transmittance. If insulation is fitted poorly, with gaps and cold bridges, then the thermal transmittance can be considerably higher than desired. Thermal transmittance takes heat loss due to conduction, convection and radiation into account. [1]

For instance, the overall ***U***-value of a window includes the conductance of the glass panes, the air inside, the framing material, and any other materials in their different thicknesses and locations. Except in special cases, the conductance of the materials cannot be added to determine ***U***-value of the assembly.

The ***U***-value is an overall coefficient of heat transfer, and includes the effects of all elements in an assembly and all sensible modes of heat transfer (conduction, convection, and radiation), but not latent heat transfer (moisture related).

Thermal Resistance

Thermal resistance is the material's ability to resist heat flow. Designated as ***R*** (***R***-value), thermal resistance indicates how effective any material is as an insulator.

The reciprocal of thermal conductance, ***R***, is measured in second needed for 1 J to flow through 1 m^2 of a given thickness of a material when the temperature difference is 1 K. In the SI units are **(**m^2.K)/W. Relationship between ***U***-value and ***R***-value is showed in the following equation:

$$\boldsymbol{U}_{value} = \frac{1}{\boldsymbol{R}_{value}} \quad (2)$$

1.2.1 Calculating *U*-value

The basic ***U***-value calculation is relatively simple. In essence, the ***U***-value can be calculated by finding the reciprocal of the sum of the thermal resistances of each material making up the building element in question. Note that, as well as the material resistances, the internal and external faces also have resistances, which must be added. These are fixed values.

Simple ***U***-value calculations can be made in the following way, by considering the building element's construction layer-by-layer. Note, however, that this does not account for cold bridging (by wall ties for example), air gaps around insulation, or the different thermal properties of e.g. mortar joints. This example considers a cavity wall:

Table 1: Parameters of thermal conductivity of some building materials

Material	**Thickness** [m]	**Thermal Conductivity (*λ*)** [W/(m·K)]	**Thermal Resistance (*R*-value)** [(K.m²)/W]
Outside surface	–	–	0.040
Clay bricks	0.100	0.77	0.130
Glasswool	0.100	0.04	2.500
Concrete blocks	0.100	1.13	0.090
Plaster	0.013	0.50	0.026
Inside surface	–	–	0.130
Total			**2.916**
U-value		1 ÷ 2.916 =	0.343 [W/m²K]

1.2.2 U-value calculators

As calculation of U-values can be time consuming and complex (particularly where for example cold bridging needs to be accounted for), numerous online U-value calculators have been released. However, many of these are only available on subscription, and those that are free tend to be too simplistic. Another option is to request a calculation from for example an insulation manufacturer, whose product is being specified. However the common way to calculate U-value is based on the following formula:

$$\boldsymbol{U}_{value} = \frac{1}{\frac{1}{\boldsymbol{hi}} + \sum \frac{\boldsymbol{s_i}}{\boldsymbol{\lambda_i}} + \frac{1}{\boldsymbol{he}}} \tag{3}$$

Where:

- $\boldsymbol{h}i$ *: heat transmission from inside*
- $\boldsymbol{h}e$ *: heat transmission from outside*
- $\boldsymbol{s}_i$*: thickness of the layer [m]*
- $\boldsymbol{\lambda}_i$*: conductivity of the layer [W/(m.K)]*

We can also use some software that helps us to calculate the ***U***-value.

1.3 Heat Loss

Just as the human body has heat exchange processes with the environment, the building can be similarly considered as a defined unit and its heat exchange processes with the outdoor environment can be examined. Heat energy tends to distribute itself evenly until a perfectly diffused uniform thermal field is achieved. Heat tends to flow from higher temperatures to lower temperature zones by conduction, convection and radiation. The rate of heat flow by any of these three forms is determined by the temperature difference between the two zones or areas considered. The greater the temperature difference, the faster the rate of heat flow. [2]

There are 2 type of heat loss: heat loss by convection and heat loss by ventilation.

Heat loss by convection for one room can be calculated as following equation:

$$\boldsymbol{P_c} = \sum_{j=1}^{n} \boldsymbol{U_{Kj}}.\boldsymbol{A_j}.\Delta\boldsymbol{\theta} \quad (4)$$

Where:

- $\boldsymbol{P_c}$ *:Heat transferred per unit time [W]*
- $\boldsymbol{U_K}$*:* heat transfer coefficient
- $\boldsymbol{A}$ *: Area of the surface [m^2]*
- $\Delta\boldsymbol{\theta}$*: Temperature difference between temperature inside and outside [K]*

Heat loss by ventilation can be calculated as following equation:

$$\boldsymbol{P_v} = \boldsymbol{V_{air}} * \boldsymbol{\rho} * \boldsymbol{c_p} * \Delta\boldsymbol{\theta} \quad (5)$$

Where:

- $\boldsymbol{P_v}$ *: Heat transferred per unit time [W]*
- $\boldsymbol{V_{air}}$*: Air flow rate [m^3/s]*
- $\boldsymbol{\rho}$ *: Density of air [kg/m^3]*
- $\Delta\boldsymbol{T}$*: temperature difference between temperature inside and outside [K]*
- $\boldsymbol{c_p}$*: Specific heat of air [kJ/(kg.K)]*

1.4 Cooling Load

Cooling load is the rate at which a cooling system or process must remove heat from a conditioned zone to maintain it at a constant dry bulb temperature and humidity. The cooling load can be further decomposed into sensible and latent cooling loads. Sensible heat into the

space causes its air temperature to rise while latent heat is associated with the rise of the moisture content in the space. The building design, internal equipment, occupants, and outdoor weather conditions may affect the cooling load in a building using different heat transfer mechanisms. The SI units are watts. [9]

For a thorough calculation of the zones and whole-building loads, one of the following three methods should be employed:

a. Transfer Function Method (TFM): This is the most complex of the methods proposed by ASHRAE and requires the use of a computer program or advanced spreadsheet.

b. Cooling Load Temperature Differential/Cooling Load Factors (CLTD/CLF): This method is derived from the TFM method and uses tabulated data to simplify the calculation process. The method can be fairly easily transferred into simple spreadsheet programs but has some limitations due to the use of tabulated data.

c. Total Equivalent Temperature Differential/Time-Averaging (TETD/TA): This was the preferred method for hand or simple spreadsheet calculation before the introduction of the CLTD/CLF method.

These three methods are well documented in ASHRAE Handbook Fundamentals, 2001.

2 CLEAN ROOMS AND REQUIREMENTS OF CLEAN ROOMS (HOSPITAL)

Typically used in manufacturing or scientific research, a clean room is a controlled environment that has a low level of pollutants such as dust, airborne microbes, aerosol particles, and chemical vapors. To be exact, a clean room has a controlled level of contamination that is specified by the number of particles per cubic meter at a specified particle size. The ambient air outside in a typical city environment contains 35,000,000 particles per cubic meter, 0.5 mm and larger in diameter, corresponding to an ISO 9 clean room which is at the lowest level of clean room standards.

Clean Room Classifications

Clean rooms are classified by how clean the air is. In Federal Standard 209 (A to D) of the USA, the number of particles equal to and greater than 0.5mm is measured in one cubic foot of air, and this count is used to classify the clean room. This metric nomenclature is also accepted in the most recent 209E version of the Standard. Federal Standard 209E is used domestically. The newer standard is TC 209 from the International Standards Organization. Both standards classify a clean room by the number of particles found in the laboratory's air. The clean room classification standards FS 209E and ISO 14644-1 require specific particle count measurements and calculations to classify the cleanliness level of a clean room or clean area. In the UK, British Standard 5295 is used to classify clean rooms. This standard is about to be superseded by BS EN ISO 14644-1. [3] [4]

Clean rooms are classified according to the number and size of particles permitted per volume of air. Large numbers like "class 100" or "class 1000" refer to FED_STD-209E, and denote the number of particles of size 0.5 mm or larger permitted per cubic foot of air. The standard also allows interpolation, so it is possible to describe e.g. "class 2000."

Small numbers refer to ISO 14644-1 standards, which specify the decimal logarithm of the number of particles 0.1 μm or larger permitted per m^3 of air. So, for example, an ISO class 5 clean room has at most 10^5 = 100,000 particles per m^3 of air.

Both FS 209E and ISO 14644-1 assume log-log relationships between particle size and particle concentration. For that reason, there is no such thing as zero particle concentration. Ordinary room air is approximately class 1,000,000 or ISO

Table 2. ISO 14644-1 Clean room Standards

Class	**Maximum particles/m^3**						**FED STD 209E equivalent**
	0.1 µm	**0.2 µm**	**0.3 µm**	**0.5 µm**	**1 µm**	**5 µm**	
ISO 1	10	2					
ISO 2	100	24	10	4			
ISO 3	1,000	237	102	35	8		Class 1
ISO 4	10,000	2,370	1,020	352	83		Class 10
ISO 5	100,000	23,700	10,200	3,520	832	29	Class 100
ISO 6	1,000,000	237,000	102,000	35,200	8,320	293	Class 1,000
ISO 7				352,000	83,200	2,930	Class 10,000
ISO 8				3,520,000	832,000	29,300	Class 100,000
ISO 9				35,200,000	8,320,000	293,000	Room Air

Table 3. BS 5295 Clean room Standards

	Maximum particles/m^3				
Class	**>=0.5 µm**	**>=1 µm**	**>=5 µm**	**>=10 µm**	**>=25 µm**
Class 1	3,000		0	0	0
Class 2	300,000		2,000	30	
Class 3		1,000,000	20,000	4,000	300
Class 4			20,000	40,000	4,000

2.1 Clean rooms Overview

Clean rooms are used in practically every industry or hospital, where small particles can adversely affect the manufacturing process. They vary in size and complexity, and are used extensively in industries such as laboratory and operation rooms in hospital, semiconductor manufacturing, pharmaceuticals, biotech, medical device and life sciences, as well as critical process manufacturing common in aerospace, optics, military and some mechanical rooms.

A clean room is any given contained space where provisions are made to reduce particulate contamination and control other environmental parameters such as temperature, humidity and pressure. The key component is the High Efficiency Particulate Air (HEPA) filter that is used to trap particles that are 0.3 μm and larger in size. All of the air delivered to a clean room passes through HEPA filters, and in some cases where stringent cleanliness performance is necessary, Ultra Low Particulate Air (ULPA) filters are used.

Personnel selected to work in clean rooms undergo extensive training in contamination control theory. They enter and exit the clean room through airlocks, air showers and/or gowning rooms, and they must wear special clothing designed to trap contaminants that are naturally generated by skin and the body.

Low-level clean rooms may only require special shoes having completely smooth soles that do not track in dust or dirt. However, shoe bottoms must not create slipping hazards since safety always takes precedence. A clean room suit is usually required for entering a clean room. Class 10,000 clean rooms may use simple smocks, head covers, and booties. For Class 10 clean rooms, careful gown wearing procedures with a zipped cover all, boots, gloves and complete respirator enclosure are required.

2.2 Clean room Air Flow Principles

Clean rooms maintain particulate-free air through the use of either HEPA or ULPA filters employing laminar or turbulent air flow principles. Laminar, or unidirectional, air flow systems

direct filtered air downward in a constant stream. Laminar air flow systems are typically employed across 100% of the ceiling to maintain constant, unidirectional flow. Laminar flow criteria is generally stated in portable work stations (LF hoods), and is mandated in ISO-1 through ISO-4 classified clean rooms.

Proper clean room design encompasses the entire air distribution system, including provisions for adequate, downstream air returns. In vertical flow rooms, this means the use of low wall air returns around the perimeter of the zone. In horizontal flow applications, it requires the use of air returns at the downstream boundary of the process. The use of ceiling mounted air returns is contradictory to proper clean room system design.

3 HVAC – REQUIREMENTS FOR CLEAN ROOMS (HOSPITAL)

HVAC design for health care facilities is all about providing a safer environment for patients and staff. The basic difference between air conditioning for healthcare facility and that of other building types stem from:

1. The need to restrict air movement in and between the various departments (no cross movement).

2. The specific requirements for ventilation and filtration to dilute and reduce contamination in the form of odor, airborne micro organisms and viruses, and hazardous chemical and radioactive substances. Ventilation effectiveness is very important to maintain appropriate indoor air quality.

3. The different temperature and humidity requirements for various areas and the accurate control of environmental conditions.

4. The design sophistication to minimize the risk of transmission of airborne pathogens and preserve a sterile and healing environment for patients and staff.

These requirements demand very high quantities of outside air along with significant treatment of this ventilation air, including cooling, dehumidifying, reheating, humidifying and filtration. [10]

3.1 Infection Control

In a hospital environment, there tend to be high concentrations of harmful microorganisms. From an infection control perspective, the primary objective of hospital design is to place the patient at no risk for infection while hospitalized. The special technical demands include hygiene, reliability, safety and energy-related issues.

Infections, which may result from activities and procedures taking place within the facility, are a cause for great concern. Three main routes responsible for infections are contact, droplet, and airborne transmission, which are quite affected by room design and construction factors.

3.2 Contact Transmission

Contact transmission is the most important and frequent mode of transmission of infections (nosocomial). It can be subdivided into direct-contact transmission and indirect-contact transmission.

a) Direct-contact transmission involves direct body to body contact for the transfer of micro-organisms from an infected person to a susceptible host.

b) Indirect-contact transmission involves the contamination of an inanimate object (such as instruments or dressings) by an infected person.

3.3 Droplet Transmission

Droplet transmission occurs when an infected person generates droplets containing microorganisms which are propelled at a short distance through the air and deposited on the conjunctivae, nasal mucosa or mouth of a host. Droplets do not remain suspended in the air, so

special air handling and ventilation are not required to prevent droplet transmission. A person's coughing, sneezing and talking generate droplets. Other procedures such as suctioning and bronchoscopy are also a source of droplets.

Airborne Transmission

Airborne transmission occurs when either airborne droplet nuclei or dust particles disseminate infectious agents.

a) **Droplet nuclei** - The high velocity with which coughing and sneezing expel droplets from the respiratory tract results in large numbers of bacteria or viruses entering the air in smaller droplets. These droplets rapidly evaporate in the air leaving a residue of typically 5 μm or smaller in size. These droplet nuclei settle so slowly that they remain airborne in occupied spaces and circulate on air currents until mechanically removed by the ventilation system. Control of environmental factors (such as special air handling and ventilation) is necessary to prevent nosocomial airborne transmission of microorganisms.

b) **Dust** - Dust contaminated by viable infectious agents may build up as a reservoir capable of causing an outbreak of infection, even after the departure of the infectious patient from whom the pathogens originated. Dust may become contaminated when dried sputum and other infectious secretions suspended in the air as dust particles mix with environmental dust.

3.4 Isolation Rooms

The infected patient can contaminate the environment. A single room with appropriate air handling and ventilation is particularly important to prevent direct or indirect contact transmission and also for reducing the risk of airborne transmission of microorganisms from a source patient to susceptible patients and other persons in hospitals. This is often termed "Isolation Room" in medical terminology. There are two types of isolation rooms: 1) airborne infection isolation (AII) rooms and 2) protective environment (PE) rooms.

1. **Airborne infection isolation (AII)** refers to the isolation of patients infected with organisms spread via airborne droplet nuclei <5μm in diameter. These include patients

suffering from measles, chicken pox and tuberculosis. Other areas include: the emergency department, intensive care units (adult, pediatric, newborn) and procedure areas such as bronchoscope suites or sputum induction rooms.

2. **Protective environment (PE)** is a specialized area for patients who have undergone allogeneic hematopoietic stem cell transplant (HSCT). The patients whose immune mechanisms are deficient because of immunologic disorders (e.g., human immunodeficiency virus [HIV] infection or congenital immune deficiency syndrome), chronic diseases (e.g., diabetes, cancer, chemotherapy, emphysema, or cardiac failure), or immunosuppressive therapy (e.g., radiation, organ transplant, cytotoxic chemotherapy, anti-rejection medication, or steroids) are also placed in protective environment.

How does above classification affect HVAC designer?

The differentiating factor between "AII" and "PE" rooms is the pressure relationships.

• The protective environments (PE) are set at POSITIVE air pressure relative to adjoining spaces. These areas require frequent air exchanges (>12 per hour) and require all supply air passing through high efficiency particulate air (HEPA) filters.

• The isolation rooms housing infectious patients (AII) must be maintained at NEGATIVE pressure. These areas require frequent air exchanges (>12 per hour) and require all supply air to be exhausted without recirculation.

Both these areas require inline monitoring to ensure that they remain under set pressure. Doors to the rooms should be self-closing, and the walls, windows, ceiling, floor, and penetrations well sealed.

3.5 HVAC Control Parameters for Isolation Rooms

An airborne infectious isolation room is constructed to minimize the migration of air from an isolation room to other areas of health care facilities. The risk of being infected through the

airborne route is a function of particle concentration. The chance of a particle that is carrying an organism falling into an open wound increases with particle concentration. By reducing the concentration we reduce the chance of infection and, hence, the number of patients infected.

Four main factors affect the local concentration around a person in a room:

1. Firstly, the concentration of particles would tend to increase with rate of production of particles in the room.

2. Secondly, the proportion of supply and exhaust air quantity in relation to the size of the room.

3. Thirdly, the level of filtration of the supplied air will affect the ability of the ventilation system to dilute the room air particle concentration and

4. Fourthly, air turbulence and air movement in the room can transport particles so the method of air distribution will affect local concentrations.

The last three of these are attributes of the ventilation system that can be engineered to limit the effect of the first. Recommendations for engineering controls to contain or prevent the spread of airborne contaminants center on:

1. General ventilation

2. Air cleaning (primary and secondary filtration)

3. Local exhaust ventilation (source control)

3.6 General Ventilation

The most effective means of controlling contaminants, odor and indoor air pollution is through ventilation, which requires simultaneous control of number of conditions:

1. Air change rates

2. Pressure gradient appropriate with class of isolation

3. Appropriate air distribution in the compartments being air conditioned.

4. High quality air filtration including absolute filtration

5. Precise temperature and humidity control ensuring maintenance of the intended microclimate

3.7 Air Change Rates

Ventilation supply rates for health care facilities require large expenditure of fresh air to dilute and remove the contaminants generated in the space. The ventilation rates for healthcare facilities is expresses as air changes air per hour (ACH), which is a measure of how quickly the air in an interior space is replaced by outside (or conditioned) air. For example, if the amount of air that enters and exits in one hour equals the total volume of the space, the space is said to undergo one air change per hour. Air flow rate is measured in appropriate units such as cubic meter per hour [CMH] and is given by:

$$\dot{V} = \frac{(\text{ACH}) * (\text{room volume})}{1\ \text{hour}} \tag{6}$$

In this equation:

- $\dot{V}$ *is the volume flow rate of air being calculated* $[m^3/hour]$
- *ACH is the number of air changes per hour.*

To determine the airflow required to adequately ventilate an area,

1) Calculate the room volume to be ventilated Width x Length x Height = m^3 (cubic meter).

2) Calculate the air volume requirement by multiplying the room volume by the air change rate per hour = m^3/h.

Studies carried out be AIA, indicate that just one air-change with fresh air can remove 63% of suspended particles from the room air. If a ventilation system can perform 10 air changes per hour (ACH), it takes 14 minutes to remove 90% of airborne contaminants in a room and 28 minutes to remove 99%. Thus increased number of fresh air changes per hour is effective for cleaning airborne contaminants. However, the higher air change rate (>20 ACH) may cause turbulence and the cost for ventilation itself will be too high. Therefore, a recommended compromise of 12 ACH is proposed which should be achievable when the filters have reached their maximum pressure drop. Higher ACR also equates to higher energy use.

The selection of 12 air changes per hour is largely a matter of convention. Ventilation rates are voluntary unless a state or local government specifies a standard in healthcare licensing requirements. These standards typically apply to only the design of a facility, rather than its operation. Based on the scientific knowledge and professional judgment reflected in the AIA guidelines, ASHRAE and the American National Standards Institute (ANSI) have developed design recommendations for ventilation and pressure relationships for various patient-care areas. Healthcare facilities without specific ventilation standards should follow ANSI/ASHRAE Standard 62, Ventilation for Acceptable Indoor Air Quality or otherwise in the absence of any specified supply air change/hour following guidelines may be used:

- For the space to be maintained under negative pressure exhaust 10 to 15 percent more air than the supply.
- For the space to be maintained under positive pressure, exhaust 10 to 15 percent less air than the supply air.

3.8 Room Pressure Control

Building room pressurization is a critical factor to monitor in a hospital as it can greatly affect the controllability of the environment. If the building pressure is allowed to become negative due to supply filters being loaded, supply fans running too slow, or return fans running too fast, humid and dirty air can be drawn into the building through cracks and openings. This air is

completely unconditioned and can provide several of the necessary ingredients to promote mold growth (e.g., moisture, more spores, and nutrients.)

Building room pressure gradient is achieved by controlling the quality and quantity of intake and exhaust air, maintaining differential air pressures between adjacent areas, and designing patterns of airflow for particular clinical purposes.

3.8.1 Class N – Negative Pressure Isolation Rooms

The basic principle of pressurization for microbial contaminant control is to ensure that air flows from less contaminated to more contaminated areas. Air in an open Class N room, for example, should flow from corridors INTO the isolation room to prevent the spread of airborne contaminants from the isolation room to other areas. The purpose of this design is to eliminate the spread of infectious contaminants and pathogens into the surrounding environment via the airborne route.

Class N is applicable to all infection isolation rooms where the patients known to or suspected to have infections are placed.

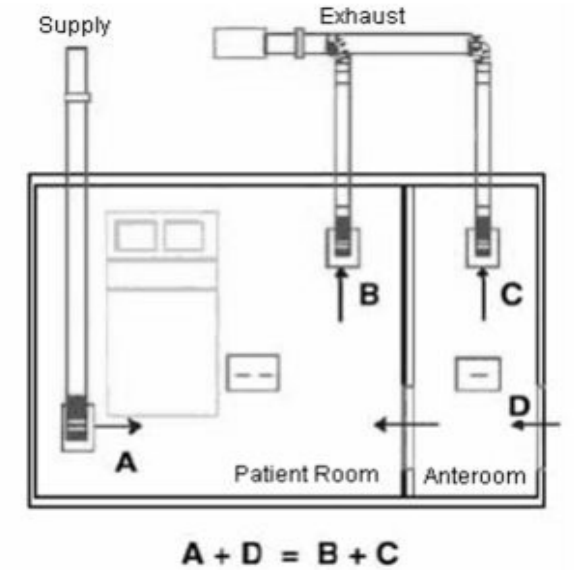


Fig. 1. Negative Pressure Isolation Rooms

The schematic above shows HVAC air flow arrangement for class N rooms. An anteroom designed to provide an "air-lock" (no mix of air) between the infectious patient and the common space is placed adjacent to the patient room. The air would flow from the ante room to the isolation room. Pressure control is maintained by modulating the main supply and exhaust dampers based on a signal from a pressure transducer located inside the isolation room.

Infection-Control and Ventilation Requirements for "AII" Rooms

Use AIA guidelines as minimum standards where state or local regulations are not in place for design and construction of ventilation systems in new or renovated health-care facilities.

Recommended elements include:

1. Ensure that the airborne infectious isolation rooms are designed to maintain negative pressure.

2. Maintain continuous negative air pressure no less than (2.5 Pa [0.01 inch water gauge]) in relation to the air pressure in the corridor. This is accomplished via a separate exhaust system sized to remove at least 15% more air than that of the supply system.

3. Monitor air pressure periodically, preferably daily, with audible manometers or smoke tubes at the door (for existing AII rooms), or with a permanently installed visual monitoring mechanism.

4. Provide ventilation to ensure >12 ACH for renovated rooms and new rooms, and >6 ACH for existing AII rooms, when supply or exhaust air filters are at their maximum pressure drop.

5. The recommended air filtration for the class N, airborne infectious isolation rooms is MERV 14 rating air filters (90% dust spot test filters) on the supply side and HEPA (99.97% @ 0.3µm DOP) on the exhaust side.

6. Recirculation of exhausted air is discouraged, from class N rooms. The exhaust air should be directed to outside, away from air-intakes and populated areas. However, where recirculation may be deemed acceptable in some circumstances, HEPA filters (99.97% @ 0.3µm DOP) capable of removing airborne contaminants on the supply side must be incorporated.

7. The disposal of effluents should not create a hazard to persons outside or the staff maintaining these systems. Where supplemental engineering controls for air cleaning are indicated from a risk assessment of the "AII" area, also install Ultraviolet Germicidal Irradiation (UVGI) units in the exhaust air ducts of the HVAC system to supplement HEPA filtration. For example in TB clinics, the air is often HEPA filtered and sometimes given UVGI exposure before exhausting to the outside, though the reasons for this are primarily because of litigation concerns and not based on any known realities.

8. Consider UVGI fixtures on or near the ceiling to irradiate upper room air. Note that UVGI, may be used to augment HEPA filters, but cannot be used in place of HEPA filters, as their effectiveness on airstreams is limited.

9. The supply air should be located such that clean air is first passed over the staff/other occupants and then to the patient. Air distribution should reduce the staff's exposure to potential airborne droplet nuclei from infectious patients, accounting for the positions of the staff and the patient, and the procedures undertaken in the isolation room.

10. Insider patient room, the supply air should be from the ceiling diffuser located at the perimeter near to the entry and the exhaust air should be drawn at lower levels approximately 6 inches above the floor in the room.

11. Exhaust air ducts should be independent of the building's common exhaust air system to reduce the risk of contamination from back draught.

12. Locate the exhaust fan at a point in the duct system that will ensure the duct is under negative pressure throughout its run within the building.

13. The makeup air intakes should be located so that no contaminated air from nearby exhaust stacks or any sources of air contaminants is drawn into the makeup air system.

14. Ensure supply air ducts are independent of the building's common supply air system. If sharing of supply ducts with other isolation rooms is unavoidable, provide the ducts with terminal HEPA filters (or other failsafe back draught prevention system). Install a high efficiency bag filter as a pre-filter to protect the HEPA filter.

15. Design the supply air and exhaust systems to be of a constant volume system. Variable air volume (VAV) systems are NOT recommended.

16. A monitoring system should be provided to signal any malfunction of the supply/exhaust air system. Consider differential low -pressure instrumentation in a prominent location outside the room along with a local audible alarm in case of supply/exhaust failure.

17. Ensure that rooms are well-sealed for better maintenance of pressure gradients that will also eventually reduce load on the air handling plant. Ensure air tightness by

- Properly constructing windows, doors, and intake and exhaust ports
- Maintain plasterboard ceilings that are smooth and free of fissures, open joints, and crevices
- Sealing all penetrations on the walls above and below the ceiling
- Monitoring for leakage and making any necessary repairs

18. Install self-closing devices on all 'AII' room exit doors considering the direction of door swing in relation to room pressure.

19. Provide a staff hand-wash basin in the anteroom and include personal respiratory protection for persons entering these rooms and for staff who lack immunity to airborne viral diseases (e.g., measles or varicella zoster virus [VZV] infection).

20. Do not use a room with a through-the-wall ventilation unit unless it can be demonstrated that all required 'AII' engineering controls are met.

21. Maintain backup ventilation equipment (e.g. portable units for fans or filters) for emergency provision of ventilation requirements for AII rooms, and take immediate steps to restore the fixed ventilation system.

22. Label the area as being a negative pressure isolation room.

Emergency Rooms and Reception Areas

In public areas of a health care facility such as an emergency room, reception and waiting areas, persons with undiagnosed active infection can come in contact with and infect others prior to

examination and treatments. The likelihood of airborne contaminants leaving these rooms is reduced by keeping these rooms under NEGATIVE pressure, relative to surrounding areas. Air is exhausted from these rooms either directly to the outside or through high efficiency particulate air (HEPA) filters.

3.8.2 Class P – Positive Pressure Isolation Rooms

Class P - positive pressure isolation rooms are set at positive pressure relative to ambient pressure, meaning that air flow must be from the "cleaner" area towards the adjoining space (through doors or other openings). This is achieved by the HVAC system providing more air into the "cleaner" space than is mechanically removed from that same space. Class P is applicable to all protective environments housing severely neutropenic and immuno-suppressed patients.

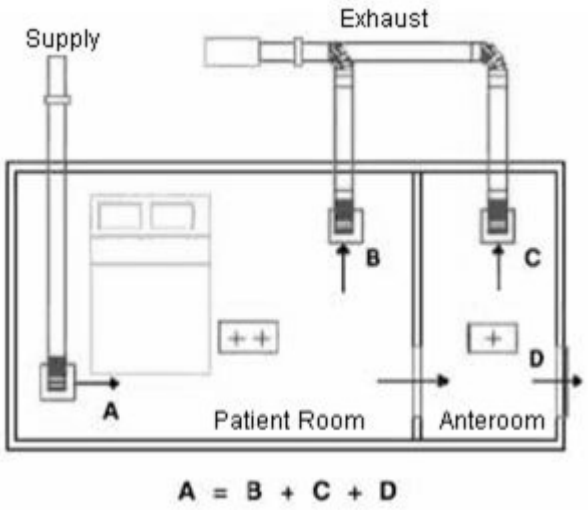


Fig. 2. Positive Pressure Isolation Rooms

In the schematic diagram above an airlock or anteroom is provided adjacent to the patient room. For a positive pressure room, air would flow from the isolation room to the anteroom and then to the corridor. Pressure control is maintained by modulating the main supply and exhaust dampers based on a signal from a pressure transducer located inside the isolation room.

Infection Control and Ventilation Requirements for Protective Environment (PE) rooms

Use AIA guidelines as minimum standards where state or local regulations are not in place for design and construction of ventilation systems in new or renovated health-care facilities. Recommended elements include:

1. Ensure that the protective environment room is designed to maintain positive pressure.

2. Maintain positive room air pressure (>2.5 Pa [0.01-inch water gauge]) in relation to the corridor. Ideally it should be >8 Pa (0.03 inch-water gauge).

3. Ventilate the room to maintain > 12 ACH or 145 liters per second per patient (whichever results in the greatest air quantity), when the supply air filter is at the maximum pressure drop.

4. Class P rooms can be either 100% fresh airs or can use re-circulated air usually a 60/40 mix of outdoor air/re-circulated air. As rule of thumb, air pressure should be maintained positive with respect to any adjoining rooms by supplying 10 to 15% excess airs.

5. The recommended air filtration for the class P, protective rooms is HEPA (99.97% @ 0.3µm DOP) on the supply side and NO filtration is needed on the exhaust side. The HEPA filter could be centrally located at the air handling unit or point-to-use HEPA filters may be used. A terminal HEPA filter at the point of use is recommended.

6. UVGI systems are sometimes used in conjunction with HEPA filters. When ultraviolet germicidal irradiation (UVGI) is used as a supplemental engineering control, install fixtures 1) on the wall near the ceiling or suspended from the ceiling as an upper air unit; 2) in the air-return duct of an 'AII' area; or 3) in designated enclosed areas or booths for sputum induction.

7. The supply air should be located such that clean air is first flows across the patient bed and exits from the opposite side of the room. Air distribution should reduce the patient's exposure to potential airborne droplet nuclei from occupants.

8. Positive pressure rooms may share common supply air systems.

9. Differential pressure indication device should be installed to permit air pressure readings in the rooms and provide a local audible alarm in case of fan failure.

10. Ensure that rooms are well-sealed for better maintenance of pressure gradients that will also eventually reduce load on the air handling plant. Ensure air tightness by

• Properly constructing windows, doors, and intake and exhaust ports

• Maintain plasterboard ceilings that are smooth and free of fissures, open joints, and crevices

• Sealing all penetrations on the walls above and below the ceiling

• Monitoring for leakage and making any necessary repairs

11. Maintain airflow patterns and monitor these on a daily basis by using permanently installed visual means of detecting airflow in new or renovated construction, or by using other visual methods (e.g., flutter strips or smoke tubes) in existing PE units. Plan for automatic documentation of the monitored results.

12. Install self-closing devices on all room exit doors in PE rooms. All emergency exits (e.g., fire escapes, emergency doors) in PE wards should be kept closed (except during emergencies) and equipped with alarms.

13. Do not use laminar air flow systems in newly constructed PE rooms.

14. Do not use a room with a through-the-wall ventilation unit as PE room.

15. Install an ensuite bathroom along with a staff hand-wash basin in the anteroom.

16. Label as a positive pressure isolation room.

Infection-Control and Ventilation Requirements for Operating Rooms

The room pressure requirement for operating rooms is similar to PE rooms with following exceptions:

1. Maintain positive-pressure ventilation with respect to corridors and adjacent areas; maintain >15 ACH, of which >3 ACH should be fresh air.

2. Filter all re-circulated and fresh air through the appropriate filters, providing 90% efficiency (dust-spot testing) at a minimum.

3. In rooms not engineered for horizontal laminar airflow, introduce air at the ceiling and exhaust air near the floor.

4. Do not use ultraviolet (UV) lights to prevent surgical-site infections.

3.9 Air Distribution

In conventional air conditioning, filtered air is typically distributed from the ceiling, with return air is collected from the ceiling on the other side of the room.

In special situations in health care facility (e.g., operating rooms, delivery rooms, catheterization laboratories, angiography rooms, HEPA-filtered rooms for immune suppressed patients) the direction of air movement needs to be controlled. The air is introduced from ceiling registers on the perimeter and is returned or exhausted through registers located at least 6 inches above the floor. This arrangement provides a downward movement of clean air through the breathing and working zones to the contaminated floor area for exhaust.

Figure below shows the introduction of low velocity air near the ceiling at the entrance of the room, flowing past the patient, and exhausted or returned close to the floor at the head of the patient bed. An airflow pattern is thus established which helps to move microorganisms from the point of patient's expulsion to the exhaust / return air terminal to prevent health care workers or visitors from inhaling the bacteria.

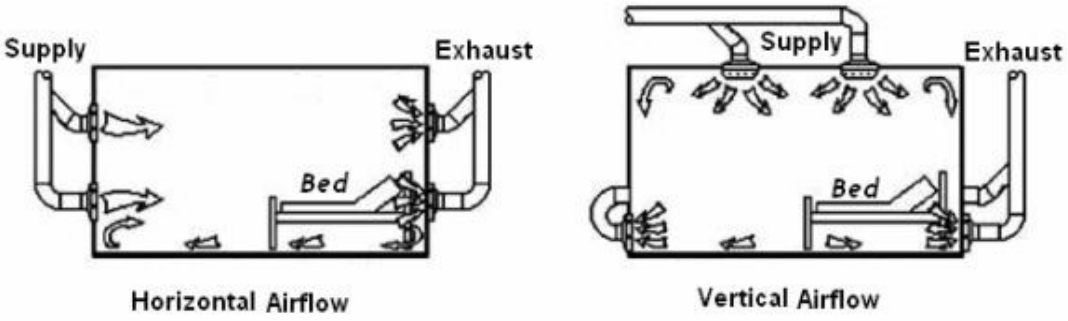


Fig. 3. Room Air Distribution in Isolation Rooms

Non-aspirating diffusers (typically perforated face) are recommended. These diffusers entrain large amounts of air, achieve good mixing, prevent updrafts and provide a laminar flow of air that will flush the isolation room of unwanted airborne particles.

The diffuser should be placed away from patient bed, preferably near the point where a health care worker or visitor would enter the room.

Do not place diffuser immediately over the patient bed as it would result in uncomfortable drafts projected directly at the patient.

Laminar Flow

Other mechanism of air distribution in ultra-clean areas of hospitals is the laminar or unidirectional airflow distribution. Laminar airflow ventilation systems are designed to move air in a single pass with parallel streamlines, usually through a bank of HEPA filters either along a wall or in the ceiling.

Laminar flow systems use perforated ventilation grills across the entire ceiling or side wall at air flow rates significantly greater than normal to force a steady constant stream of air across the entire room, similar to a smooth steady flow out of an open water faucet versus one that splashes as the water comes out of the faucet.

Laminar flow distribution requires a very high volume of air flow and is designed for an air velocity of 90 + 20 ft/min. This unidirectional approach optimizes airflow and minimizes air

turbulence and ensures that any contamination that is generated within the area is quickly and effectively removed. Laminar airflow systems are often used in operating rooms to help reduce the risk for healthcare-associated airborne infections.

Note: The data that demonstrate a bona-fide application and support of laminar airflow in PE rooms is lacking. Given the high cost of installation and operation, the value of laminar airflow is questionable and shall be ascertained through life cycle analysis.

3.9.1 Air Filtration

All of the air that is drawn into an air handling system is "contaminated" to some degree. It is commonly accepted that airborne particles (solid particles, liquids, fumes, smoke, or bacteria) that are larger than 5 microns in size tend to settle quickly out of the air onto horizontal surfaces. Airborne particles that are less than 5 microns in size (especially those less than 2 microns in size) tend to settle slowly out of the air and remain suspended (airborne) for larger period of time.

Concerns over hospital-acquired infections have propelled filtration solutions into the forefront as a primary tool for infection control. There are five methods of filtration.

1. Straining - Particles in the air are larger than the openings between the filter fibers. This technique is suitable for gross removal of large particles. Filtration efficiency is low.

2. Impingement – Particles collide with filter fibers and remain attached to the filter. The fibers may be coated with adhesive. Filtration efficiency is low.

3. Interception – Particles enter into the filter and become entrapped and attached to the filter fibers. Filtration efficiency is medium.

4. Diffusion – Small particles, moving in erratic motion, collide with filter fibers and remain attached. Filtration efficiency is high.

5. Electrostatic – Particles bearing negative electrostatic charge are attracted to filter with positively charged fibers. Filtration efficiency is high.

All public areas of health care facilities are required to have two banks of filters — a 30% (ASHRAE 52.1) pre-filter and 90% final filter. Provided that the final filter is properly installed and maintained and provided that there is little or no bypass around the filter, the combined efficiency of two bank filters is nearly 100% in removing particles of 1μm - 5 μm in diameter. This filtration system is adequate for most patient-care areas in ambulatory care facilities, and the operating room environment.

A common metric for filter performance is the minimum efficiency reporting value (MERV), a rating derived from a test method developed by ASHRAE. The MERV rating indicates a filter's ability to capture particles between 0.3 and 10.0 microns in diameter. A higher MERV value translates to better filtration, so a MERV-13 filter works better than a MERV-8 filter. In health care facilities a final filter of MERV-14 is satisfactory.

3.9.1.1 High Efficiency Particulate Air (HEPA)

Filters HEPA filters have a minimum initial efficiency of 99.97% for removing particles 0.3 microns in size. This is a critical point as these filters are being used to remove mold and bacteria, typically 1 to 5 microns in size when airborne, as well as viral particles which are submicron in size (as a reference, Aspergillus spores are 2.5 - 3 μm in diameter).

Each HEPA filter is individually tested at the factory in order to confirm their conformance to this standard. They may also be field-tested in order to confirm their ongoing adherence to efficiency requirements.

Where to use HEPA Filters

HEPA filters should be used:

1. On the supply air distribution of the protective rooms.

2. On the return air of the infectious isolation rooms when the air is re-circulated within the space in order to increase ACH while reducing the total exhaust requirements. Ideally the infectious isolation rooms should be designed for 100% fresh air and exhaust.

3. On the exhaust of the infectious isolation rooms and local exhaust hoods when exhausting air to the outside is not practical or when the exhaust is located near a potential air intake. (Refer note below)

4. When the HVAC system configuration dictates recirculation of air from the isolation room to other parts of the facility.

Note - The guidelines do not mandate the exhaust air from an infectious isolation room to be HEPA filtered before being discharged outdoors unless there is any chance that the exhaust air could reenter the system. However, there is always a possibility of exhaust re-entry under certain wind and climatological conditions. It is, therefore, preferable to filter all exhaust air.

3.9.1.2 HEPA Maintenance

Efficiency of the filtration system is dependent on the density of the filters that may create a pressure drop unless compensated by stronger and more efficient fans so that flow of air is maintained. When HEPA filters are used in infection control applications it is imperative to have a meticulous maintenance program in place. For optimal performance, it is critical that:

1. HEPA filters to be installed in equipment which seals the filter in place in order to prevent contaminated air from bypassing the filter.

2. HEPA filters to be tested on site when they are first installed and every six months thereafter to confirm that they are operating at their design efficiency.

3. HEPA filters to be monitored (with manometers or other pressure indicating devices) on regular basis and replaced in accordance with the manufacturer's recommendations and standard preventive maintenance practices. Gaps in and around filter banks and heavy soil and debris upstream of poorly-maintained filters have been implicated in healthcare-associated outbreaks of aspergillosis, especially during times of nearby construction.

HEPA filters are a costly budget item. In order to extend the life of a HEPA filter and reduce ongoing replacement costs, it is strongly recommended to provide a roughing pre-filter prior to the HEPA. Studies indicate that a low-efficiency pre-filter may extend the life of a HEPA filter

by 25%, while adding higher efficiency intermediate filters such as a MERV 14 (95% by ASHRAE 52.1 dust spot test) filter can extend the life of the HEPA filter by as much as 900%. This concept, called "progressive filtration," allows HEPA filters in special care areas to be used for 10 years or more. HEPA filter efficiency is monitored with the dioctylphthalate (DOP) particle test using particles that are 0.3 μm in diameter.

4 PRINCIPALS OF CALCULATIONS OF HVAC SYSTEMS AND UNITS

There are a number of issues that must be resolved before the proper HVAC system can be designed, whether it is intended for the isolation rooms, surgical suite, the patient rooms, or the administration offices.

Initially, the proper ambient design conditions must be selected. Too often, only the peak cooling design conditions are considered for sizing the capacity requirements of the system. These ambient conditions are listed in the ASHRAE Handbook – Fundamentals as the dry-bulb temperatures with mean coincident wet-bulb temperatures, representing conditions on hot, mostly sunny days. These conditions are used in sizing cooling equipment such as chillers or package equipment for cooling control. In some climates, this might be satisfactory; however, in geographic areas known for higher humidity levels, considering only this cooling condition might not be sufficient. Extreme dew-point temperature conditions may occur on days with moderate dry-bulb temperatures, resulting in high relative humidity's and peak absolute moisture loads from the weather.

These values from tables found in the Fundamentals Handbook are useful for humidity control applications, such as desiccant cooling and dehumidification, cooling-based dehumidification, and fresh air ventilation systems. These values can also be used as a checkpoint when analyzing the behavior of cooling systems at part load conditions, particularly when such systems are used for humidity control as a byproduct of temperature control.

4.1 Type of HVAC System

4.1.1 Isolation Rooms and Critical Examination Rooms

For the critical areas such as isolation rooms, intensive care units and operating rooms, critical diagnostic and examination rooms, consider only the centralized HVAC system encompassing “all air systems”.

In all air systems, the outdoor air enters the system via a low - efficiency or "roughing" filters, which removes the large particulate matter. It is mixed with the return air and is made to pass the fine filters, which removes small size particles and many microorganisms. The air is than conditioned and delivered to each zone of the building. After the conditioned air is distributed to the designated space, it is withdrawn through a return duct system and delivered back to the HVAC unit. A portion of this "return air" is exhausted to the outside while the remainder is mixed with outdoor air and filtered for dilution and removal of contaminants. In some critical areas the air again filtered through HEPA filters located downstream the cooling/heating coil or at the terminal end of the duct.

All air systems can be classified as single-zone, multi-zone, dual-duct and reheat systems.

Single-zone systems: Single-zone systems serve just one zone having unique requirement of temperature, humidity and pressure. This is the simplest of all air systems. For this type of system to work properly, the load must be uniform all through the space, or else there may be a large temperature variation.

Multi-zone systems: Multi-zone systems are used to serve a small number of zones with just one central air handling unit. The air handling unit for multi-zone systems is made up of heating and cooling coils in parallel to get a hot deck and a cold deck. For the lowest energy use, hot and cold deck temperatures are, as a rule, automatically changed to meet the maximum zone heating (hot deck) and cooling (cold deck) needs. Zone thermostats control mixing dampers to give each zone the right supply temperature.

Dual-duct systems: Dual-duct systems are much like multi-zone systems, but instead of mixing the hot and cold air at the air handling unit, the hot and cold air are both brought by ducts to each

zone where they are then mixed to meet the needs of the zone. It is common for dual-duct systems to use high-pressure air distribution systems with the pressure reduced in the mixing box at each zone.

Reheat systems: Reheat systems supply cool air from a central air handler as required to meet the maximum cooling load in each zone. Each zone has a heater in its duct that reheats the supply air as needed to maintain space temperatures. Reheat systems are quite energy-inefficient and have been prohibited by various codes. Energy may though be saved through the recovery of the refrigeration system's rejected heat and the use of this heat to reheat the air.

Caution: Air from infectious patient rooms is normally NOT recirculated and is exhausted directly to the outside via a HEPA filter. Use of terminal heating and cooling units such as fan coil units is NOT acceptable in isolation rooms, surgical suites and other critical areas where maintaining the room pressure relationships is important.

4.1.2 Normal Patient Care Rooms, Administrative and Noncritical Areas

For the patient bedrooms and other non-critical areas, any one of the following HVAC systems can be used.

1. All air systems as discussed above.

2. Terminal heating and cooling units, such as fan coil units or radiant ceiling panels.

3. Radiant heating and cooling system

The amount of outdoor air and how it is supplied to the occupied spaces would depend upon the type of HVAC system used. When the fan coil units or radiant ceiling panels are used, a central ventilation unit supplies conditioned air to the spaces. With this arrangement, the source of outdoor air being external to the principle cooling and heating equipment, it is possible to ensure the predetermined amount of outdoor air distribution to all the spaces.

4.2 Air Handling System

An air handling systems is a means of providing conditioned air to the space in order to maintain the environmental requirements. Often termed as the heart and lungs of the health care facility, these must be selected and sized properly for its very important task. And that task is to control the environment to promote the healthiest conditions possible for the patients and other occupants.

4.2.1 Air Handling Equipment Sizing Criteria

Air must be delivered at design volume to maintain pressure balances. The air handling equipment must be sized in accordance with the following guidelines:

1. Load Calculations: Heat gain calculations must be done in accordance with the procedure outlined in the latest ASHRAE Handbook of Fundamentals. The calculations performed either manually or with a computer program.

2. The calculated supply air shall be the sum of all individual peak room air quantities without any diversity.

3. Safety Margin: A safety factor of 5 percent shall be applied to the calculated room air quantity to allow for any future increase in the room internal load.

4. The adjusted supply air shall be, thus, 5 percent in excess of the calculated supply air.

5. Air leakage: The air leakage through the supply air distribution ductwork shall be computed on the basis of the method described in the SMACNA Air Duct Leakage Test Manual. The maximum leakage amount shall not exceed 4 percent of the adjusted supply air.

6. Supply Air Fan Capacity: The capacity of the supply air fan shall be calculated per the following example:

a. Calculated Supply Air Volume = 20,000 m^3/h

b. Safety Margin = 5 percent of item (a) = 1,000 m^3/h

c. Adjusted Supply Air Volume = 21,000 m^3/h

d. Duct Air Leakage = 4 percent of item (a) = 840 m^3/h

e. Supply Air Fan Capacity = 21,840 m^3/h

7. Equipment Selection: selection of the supply air fan, cooling coil, preheat coil, energy recovery coil (if any), filters, louvers, dampers, etc., shall be based on the supply fan capacity, 21,840 m^3/h calculated in the example above. A psychrometric chart shall be prepared for each air-handling unit. Make sure heat gains due to the fan motor and duct friction losses are taken into account for sizing cooling coils.

8. Air Distribution:

• The main supply air ductwork shall be sized to deliver the supply air fan capacity, 21,840 CMH as calculated in the example above.

• The individual room air distribution system including supply, return, exhaust air ductwork, air terminal units, reheat coils and air outlets/inlets shall be sized and selected on the basis of the adjusted supply air volume, 21,000 m^3/h.

• The fan and motor selection shall be based on the supply air fan capacity and static pressure adjusted, as necessary, for the altitude, temperature, fan inlet and discharge conditions, and the AMCA 201 System Effect Factors. The fan selection shall be made within a stable range of operation at an optimum static efficiency. The fan motor W (BHP), required at the operating point on the fan curves, shall be increased by 10 percent for drive losses and field conditions to determine the fan motor horsepower. The fan motor shall be selected within the rated nameplate capacity and without relying upon NEMA Standard Service Factor.

4.2.2 Air Handling Units Specifications

The air handling equipment requires special attention to disinfection, and cleanliness; clusters of infections due to Aspergillus spp., Pseudomonas aeruginosa, Staphylococcus aureus, and

Acinetobacter spp. have been linked to poorly maintained and/or malfunctioning air conditioning systems.

The failure or malfunction of any component of the HVAC system may subject patients and staff to discomfort and exposure to airborne contaminants. AIA guidelines prohibit United States hospitals and surgical centers from shutting down their HVAC systems for purposes other than required maintenance, filter changes, and construction. Often the routine maintenance and troubleshooting functions need to be addressed without necessarily disabling the units.

The following key elements need to be addressed when procuring these units.

1. Specify the cabinet construction with stainless steel or galvanized steel sheets polyester-coated both from the inside and outside. Ensure cabinet framework is constructed from aluminum profiles for increased rigidity.

2. Specify a layer of non-flammable mineral wool between the inside and outside sheets for the cabinet casing.

3. Specify oblique floors for the air handling unit, tubs for the cooling units and drip channels made of stainless steel construction. Specify vacuum seal P-trap on the drain pan.

4. Specify all edges and offsets to be filled with fungicidal silicon certified for hygienic applications in health care facilities which precludes formation of the microbe expansion centers.

5. Specify provision for pressure gauges on the filter section casing of AHU along with audible alarm. This is to confirm that NO air stream will elude filtration, if openings are present because of filter damage or poor fit.

6. Specify access and inspection openings with the lighting elements installed in covers of the sections for humidification, filtration, heat exchangers and fans.

7. Specify modular construction with all the subunits to be assembled in a manner enabling their washing from all sides. All subunits and materials shall be resistant to commonly used disinfecting agents.

8. Specify a drum fan with an inspection flap and an outflow pipe which enables the drum cleaning OR a centrifugal and axial-flow fan with an open rotor.

9. Specify driving motor manufactured in the IP class, enabling washing and disinfection.

10. Specify multistage filtration with minimum of MERV 14 final filtration installed in plastic frames and mounted in frameworks made of resistant materials. The filters shall be provided with differential pressure gauge and pollution level indicators.

11. Specify UV bactericidal lamp ensuring disinfection of the re-circulated air.

12. Specify cable glands providing connection of motors and the lighting system, ensuring the appropriate tightness and cleanliness class.

4.3 Exhaust Fans

Exhaust fan must be selected to produce the rate of airflow required by the exhaust system. The flow must be developed against the total system resistance, including pressure losses through the air distribution network including air cleaning devices. A fan of proper size and operating speed should than be selected from the ratings published by the fan manufacturer.

The exhaust fan should be located downstream of the air cleaning filter and as close to the discharge point as possible. The preferred location for an exhaust fan is outdoors, normally on the roof. A straight duct section of at least 6 equivalent duct diameters and 3 equivalent duct diameters should be used when connecting to the fan inlet and outlet respectively before any bend or fittings. When this is impractical due to space constraints, corrective devices such as turning vanes or flow dividers should be used, or the associated loss must be accounted for.

Fan selection should consider long term contaminant effects on the fan and the fan wheel. Where severe conditions of abrasion or corrosion are present, special lining or metals could be used in fan construction. Safe means should be provided to allow the wheel of an exhaust fan to be examined without removing the connecting ducts.

A flexible sleeve or band should be incorporated onto the fan inlet and outlet ducts to minimize vibration of the ductwork.

4.4 Air Distribution Ductwork

Recommended Elements:

1. In an effort to save installation dollars, the return duct is often deleted from the plans and the interstitial space between the suspended ceiling and the roof assembly, or the floor assembly above, is used as a return plenum. Open return air path directly over the false ceiling is NOT recommended for isolation rooms or elsewhere in health care facilities.

2. Any air leakage through duct joints will disrupt the pressure balance raising possibility of infectious material entraining into the air supply. The supply and return air ducts should be properly sealed and insulated during construction. On the return side of the equipment, leaky ducts will draw in far more moisture than the cooling coils were designed to remove. The result is a higher than designed and desired humidity level in the space.

3. Supply and exhaust systems should be designed as failsafe (for example, using duplex fans) to prevent contamination of any area within the facility in the event of fan failure.

4. The ductwork of a negative pressure isolation room must not communicate with the ductwork of the rest of the hospital. Ductwork should be designed to reduce the possibility of cross contamination in the event of fan failure. This can be accomplished by ducting each negative pressure isolation room separately from the air-handling unit.

5. The exhaust fan should be located at a point in the duct system that will ensure that the entire duct is under negative pressure within the building.

6. Position the exhaust discharge duct to prevent the contamination of intake air. In acute cases, the discharge plume may need to be modeled to prevent entrainment.

7. Round duct should be used for the construction of the exhaust system. Rectangular ducts, if used, should be as square as possible.

8. All branches should enter the main duct at gradual expansions at an angle not exceeding 45 and preferably 30 or less. Connections should be to the top or side of the main and directly opposite each other. Elbows and bends should be at a minimum of 2 gauges heavier than straight length ducts of equal diameter and have a centerline radius of at least 2 and preferably 2.5 times the duct diameter. The smaller branches should enter the main near the high suction end, closer to the fan inlet.

9. Exhaust stacks should be vertical and terminated at a point where height or air velocity would preclude re-entry of the contaminated air into the work environment.

10. Duct velocities should be sufficient to prevent the settling of dry aerosols. The recommended minimum duct velocity for most areas of the healthcare facility is 12.7 m/s.

11. Ductwork should be located so that it is readily accessible for inspection, cleaning and repairs; Keep provisions for routine test ports for appropriate airflow and pressure balance.

12. Labeling the ductwork helps prevent unnecessary exposure to maintenance personnel who may unknowingly cut into the ductwork for the purpose of testing airflow or repairing equipment. Using a HEPA filter at the point of exhaust in the room allows you to use non-sealed ductwork (after the HEPA), which may be on a shared exhaust run. The ductwork located after the HEPA filter does not need to be labeled as potentially contaminated.

4.4.1 Noise Criteria

1. The noise level should be restricted to 35 NC level for all patient rooms, operating rooms (major or minor), diagnostic rooms, audio suites, examination rooms, conference rooms, large offices, lobbies and waiting areas.

2. The noise level should be restricted to 40 NC level for all small private offices, nursing stations, auditoriums, treatment areas, corridors, pharmacy and general work rooms.

3. The noise level should be restricted to 45 NC level for all laboratories, Dining, Food Service/Serving , Therapeutic Pools

4. The noise level should be restricted to 50 NC level for all gymnasiums, recreation rooms, laundries and HVAC plant rooms.

4.4.2 Duct Sizing Criteria

Duct systems should be designed in accordance with the general rules outlined in the latest ASHRAE Guide and Data Books, SMACNA Manuals and Design Guide Section of the Associated Air Balance Council Manual.

1. Supply duct system, with total external static pressure 500Pa and larger, shall be designed for a maximum duct velocity of 12.7 m/s for duct mains and a maximum static pressure of 60Pa per 30.5m duct length. Static pressure loss and regain shall be considered in calculating the duct sizes. Size supply branch ducts for a maximum duct velocity of 7.62 m/s.

2. All other duct systems such as return and exhaust, including branch ducts, shall be designed for a maximum velocity of 7.62 m/s for the duct mains and a maximum static pressure of 25Pa per 30.5m duct length, with the minimum duct area of $0.3m^2$ (or 0.2m x 0.15m) size.

3. Indicate Duct Static Pressure Construction Classification according to SMACNA (0.005m, 0.025m, 0.05m, 0.075m and 0.1m) on drawings.

4.5 HVAC Equipment Location and Installation

Equipment shall be located to be accessible for installation, operation and repair. Mechanical spaces shall be of suitable size to permit inspection and access for maintenance, and to provide space for future equipment when required.

The effect that equipment noise or vibration might have on areas adjacent to, above, and below equipment shall be considered. Design shall comply with specified room sound ratings.

Location of equipment remote from sound sensitive areas should be emphasized.

Make provisions for all necessary stairs, cat walks, platforms, steps over roof mounted piping and ducts, etc., that will be required for access, operation and maintenance. Access to roofs by portable ladder is not acceptable.

4.5.1 Air Handling Equipment

Air handling units and similar equipment shall be housed in a mechanical equipment room or in a mechanical penthouse enclosure. Penthouse type of fully weatherized roof top units constructed in standard sections of modules would be acceptable in lieu of the mechanical equipment rooms or mechanical penthouses. These units shall provide excess sections for walk through servicing, maintenance, and shall ensure that the piping connections and electrical conduits are fully enclosed within the units.

4.5.2 Air Intakes and Outlets

1. Ensure that air intakes and exhaust outlets are located properly in construction of new facilities and renovation of existing facilities. a. Locate exhaust outlets >7.62m from air-intake systems. b. Locate outdoor air intakes >1.83m above ground or >0.91m above roof level… (The air intake shall be located as high as practical or not less than stated). c. Locate exhaust outlets from contaminated areas above roof level to minimize recirculation of exhausted air.

2. Operating Room system air intakes shall be at least 9.14m above the ground.

3. Laboratory and Research exhaust shall be terminated at the highest point of the building (NFPA 99, 5-3.3.4). [14]

4. Outside air intake shall not be near hot exhaust discharging horizontally or deflected down, nor be near plumbing vents, animal room exhausts, generator exhausts, loading docks, automobile entrances, driveways, passenger drop-offs, cooling towers, incinerator and boiler stacks.

5. Louvers shall be designed for a maximum velocity of 3.81 m/s through the free area of 35%. Drainable louvers may be designed for a maximum velocity of 5.08 m/s and 45 percent free area.
6. Ensure that the intakes are kept free from bird droppings, especially those from pigeons.

5 REQUIREMENTS

5.1 Requirements of Building Physic

Under building physics, we understand the variety of hygrothermal properties of the building structures. Building physics have to deal with different criteria, which require finding a compromise. On the one hand, requirements relate to assuring human comfort and health, on the other hand, economical, environmental and architectural restrictions should be respected [1].

Specific requirements for minimal heat transfer coefficient, sensible, temperature decrease, thermal resistance, temperature of inner surface for Czech Republic is given by ČSN 73 0540, ČSN EN ISO 13788 and ČSN EN ISO 6946. [11] [12][13]

According to ČSN 73 0540, there are requirements of thermal transmittance for outside walls, roof, ceiling and windows that the building must achieve:

- Thermal transmittance of outside walls must be smaller or equal 0.3 W/(m^{2}·K)
- Thermal transmittance of roof and floor must be smaller or equal 0.3 W/(m^{2}·K)
- Thermal transmittance of windows must be smaller or equal 1.5 W/(m^{2}·K)

5.2 Requirements of Equipment HVAC systems

Room air should be supplied by an external air conditioning system, preferably dedicated to the facility. Partial recirculation of room air is appropriate and this allows for optimal energy utilization. Sufficient fresh air should be supplied in accordance with ventilation codes, to balance exhaust air and to maintain specified pressures. Unless, otherwise specified, typical temperature range for this kind of room is within the range of 19 to 25 °C and relative humidity

of approximately 50% is maintained. The type of equipment and number of people in the room may dictate where in the range you need to be to assure that during production the operations area is maintained at the right temperature and humidity levels.

Only HEPA filtered air should enter the clean room and the gowning areas. These modules are available in fan assisted with fan speed control and should operate at a velocity of 0.45 ± 0.05 m/s. The location of the HEPA filters and air return grilles should create air movement from the designated ‘clean zone’ to the ‘less clean’ zones. Return air grilles should be at a lower level to aid in laminar flow requirements.

Air supply to the clean room should provide a room air change rate of >20 per hour. Air cleanliness will be enhanced by higher air change rates. When the doors are opened the supply air volume should maintain an outward flow of air.

II. PRACTICAL PART

6 DESCRIPTION OF THE RELEVANT CLINIC

The relevant hospital is a small clinic that deals with problems of andrology (the medical specialty that deals with male health, particularly relating to the problems of the male reproductive system and urological problems that are unique to men) even in fertility of women and artificial pregnancy. It contains 2 rooms for doctors, 3 rooms for nurses, 2 rooms for patients, 1 laboratory, 1 bank, 1 operation room, 1 andrology room, 2 storages, and WCs. This hospital requires clean air supply. Especially, operation room and laboratory require highest level of clean space.

7 BUILDING PARAMETERS AND THE BUILDING STRUCTURE PARAMETERS

The picture below gives an overview of the hospital:

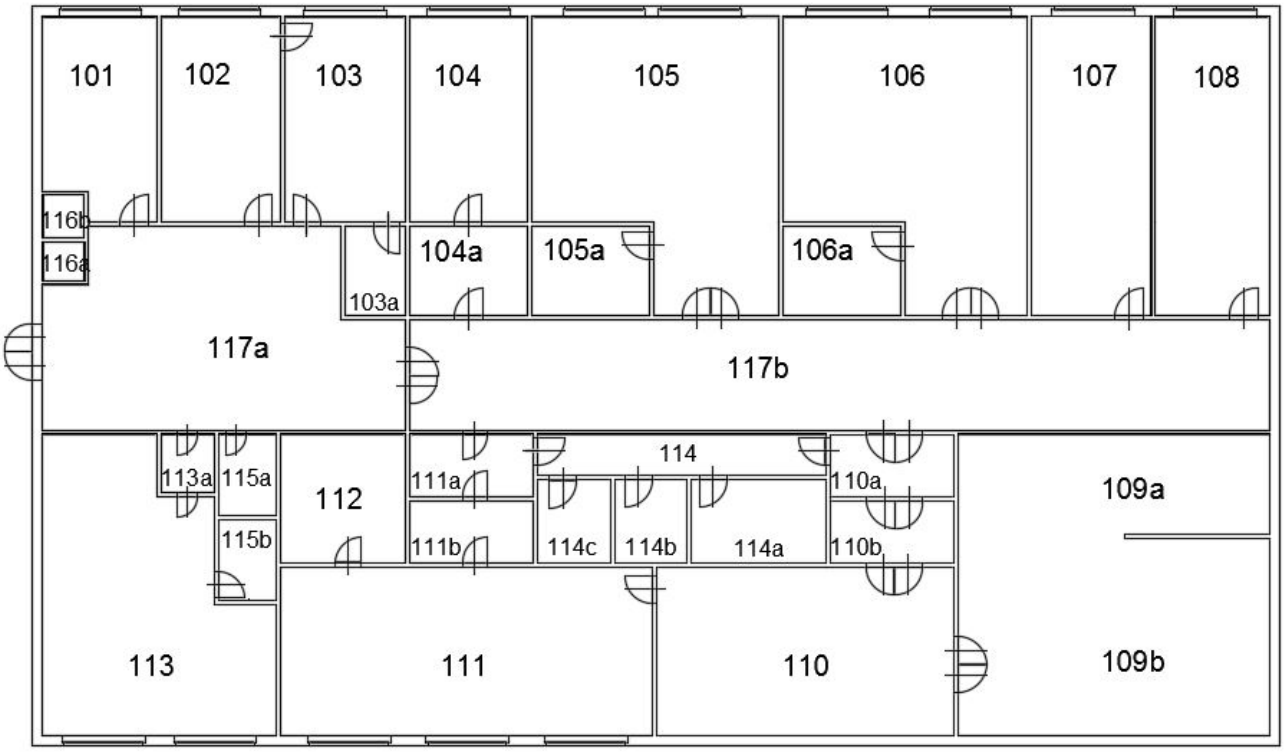


Fig. 4. Schema of the hospital

The table that shows detail information of each room:

Table 4. Hospital rooms' information

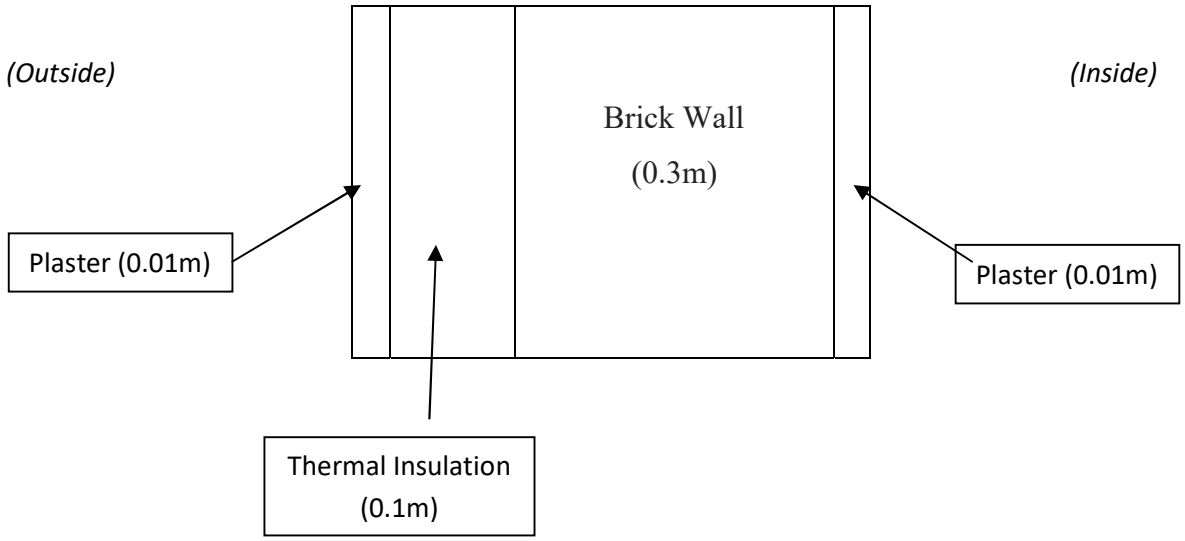
Room Number	Room's Name	Area of room [m2]	Num. of persons	Room temperature	
				Summer [°C]	Winter [°C]
101	Storage	11.3	0	-	-
102	Sister's room	11.3	2	25±2	20±2
103	Doctor's room	11.3	2	25±2	20±2
103a	WC	2.6	0	-	-
104	Reception	14.4	2	25±2	20±2
105	Patient's room	26.1	2	25±2	20±2
105a	WC	5.1	0	-	-
106	Patient's room	26.1	2	25±2	20±2
106a	WC	5.1	0	-	-
107	Sister's room	14.4	2	25±2	20±2
108	Storage	14.4	0	-	-
109	Operation room	44.9	5	24±2	24±2
110	Bank	21.2	0	24±2	24±2
110a + 110b	Dirty filter	7.6	0	25±2	20±2
111	Laboratory	30	3	25±2	20±2
111a + 111b	Dirty filter	7.6	0	25±2	20±2
112	Andrology	7.6	2	25±2	20±2
113	Embryology	30.3	2	25±2	20±2
114 a b c	Corridor	5.8	0	-	-
115a + 115b	WC	4.63	0	-	-
116a + 116b	WC	3.9	0	-	-
117a + 117b	Corridor	78.2	0	25±2	20±2

7.1 Design of Outside Walls, Floor, Roof and Windows

By using software Stavební Fyzika ver.2014, I had created the design of outside walls, floor and roof of hospital, with details of the layers: thickness, material. This software also provided very necessary and important information: U-value.

7.1.1 Outside walls

The outside wall contains 4 layers, from inside to outside we have: Plaster (0.01m), Brick wall (0.3m), Thermal insulation (0.1m), and Plaster (0.01m).



By using software Stavební Fyzika, I already created a simulation of outside wall following above schema:

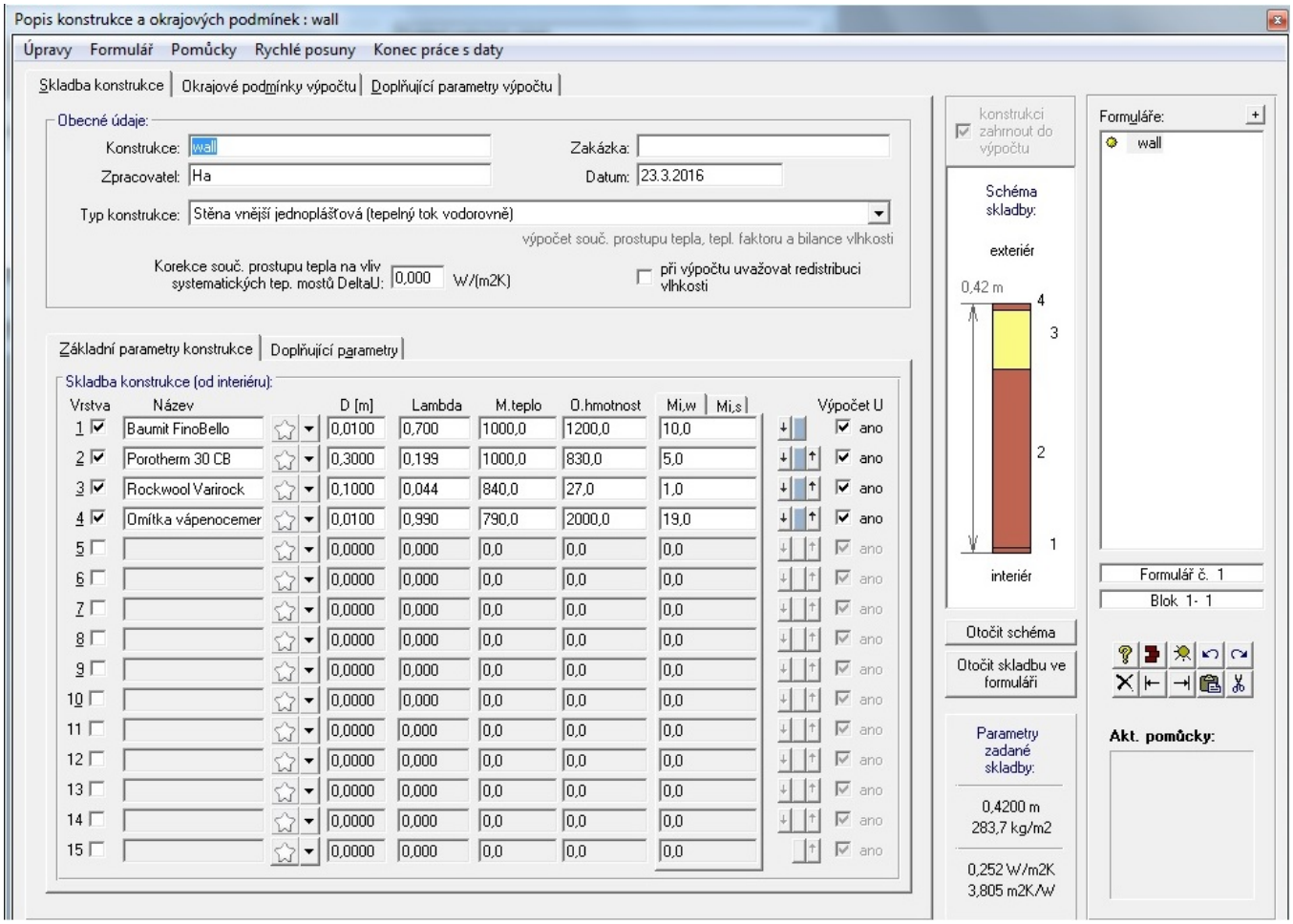


Fig. 5. Schema of outside wall created by software Stavební Fyzika

Table 5. Detail parameters of layers

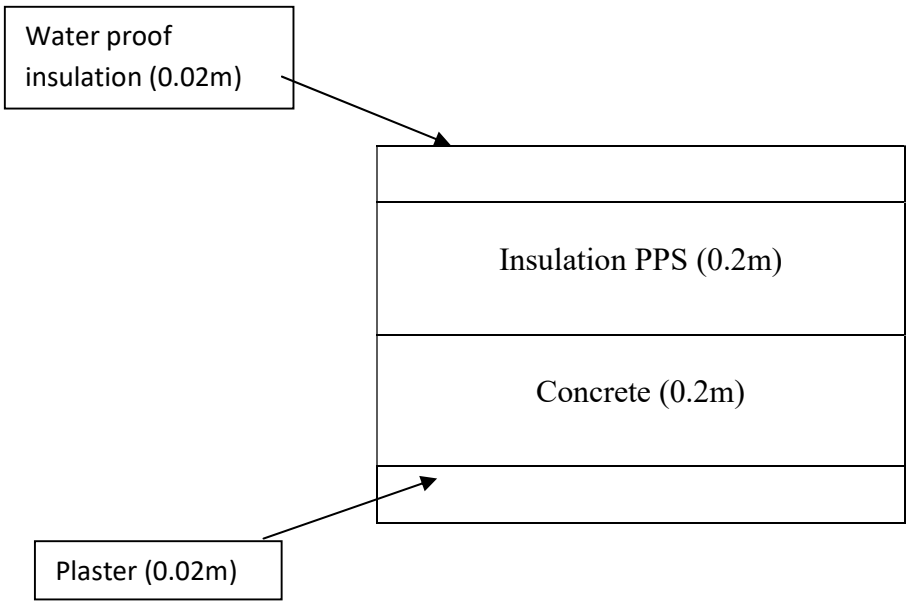
Num.	Name	D [m]	Lambda [W/(m.K)]	c [J/(kg.K)]	Ro [kg/m^3]	Mi [-]	Ma [kg/m^2]
1	BaumitFinoBel	0,0100	0,7000	1000,0	1200,0	10,0	0.0000
2	Porotherm 30 C	0,3000	0,1990	1000,0	830,0	5,0	0.0000
3	RockwoolVarir	0,1000	0,0440	840,0	27,0	1,0	0.0000
4	Omítka vápenoc	0,0100	0,9900	790,0	2000,0	19,0	0.0000

This software also calculated the U-value:

$$\textbf{\textit{U}-value (wall) = 0.252 W/(m}^{2}\textbf{·K)}$$

7.1.2 Roof

Roof of this building contains 4 layers: water proof insulation (0.02m); Insulation PPS (0.2m); Concrete (0.2m); Plaster (0.01m)



Structure of the roof created by software:

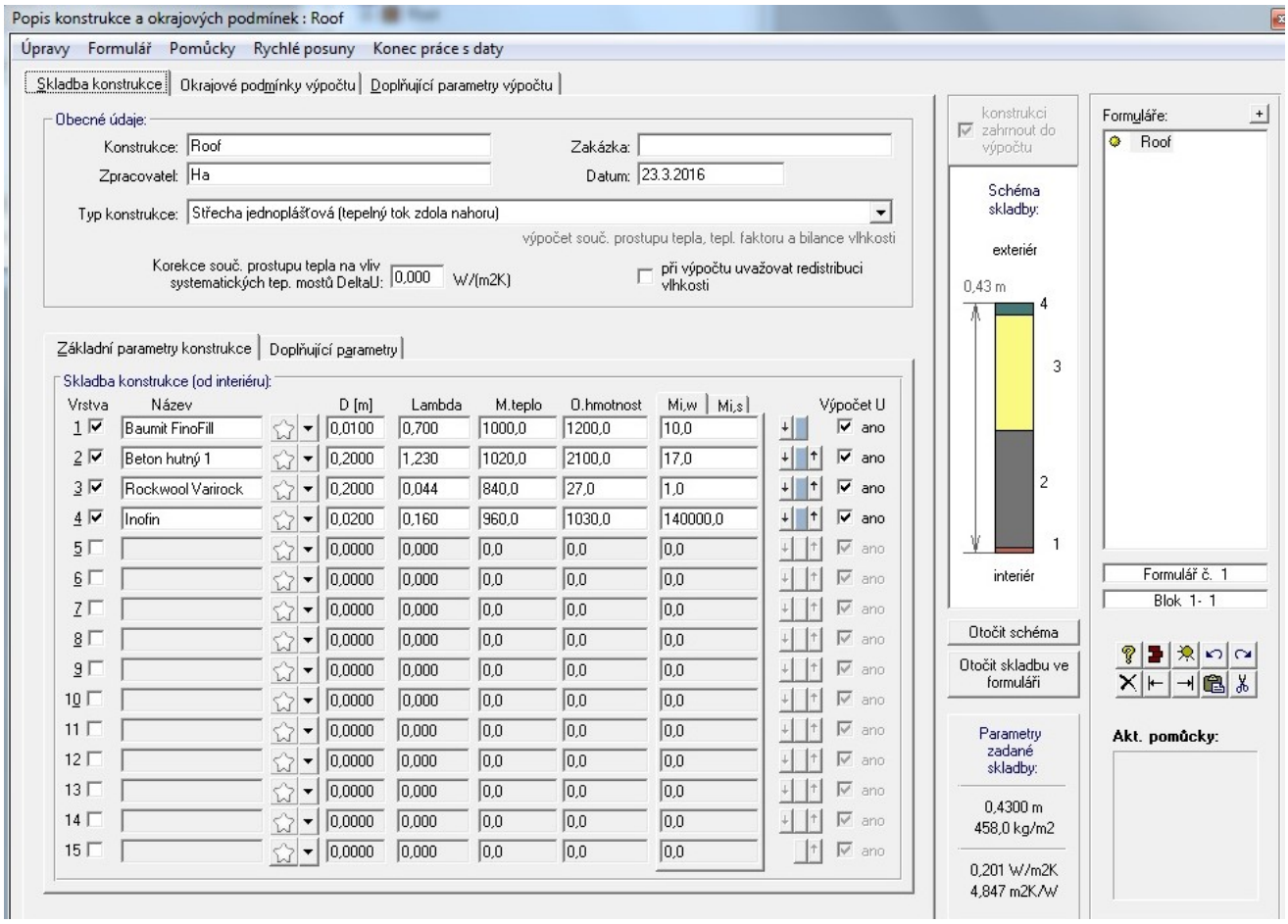


Fig. 6. Schema of roof created by software Stavební Fyzika

Table 6. Detail parameter of the roof

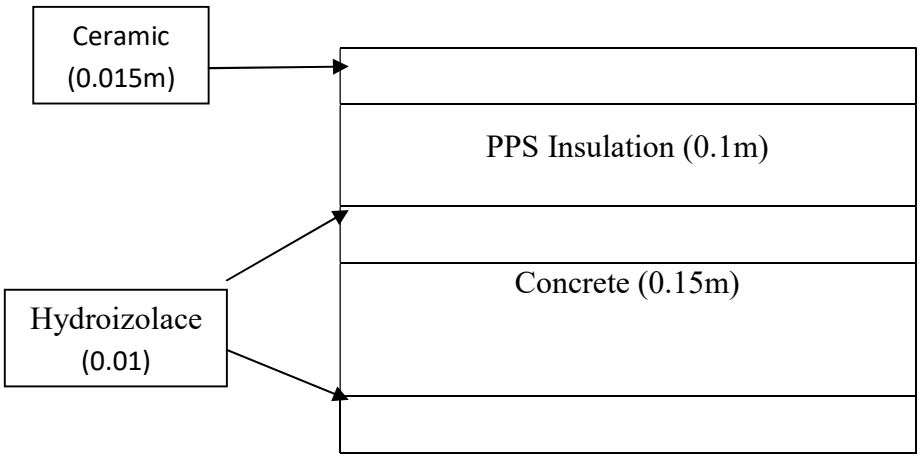
Num.	Name	D [m]	Lambda [W/(m.K)]	c [J/(kg.K)]	Ro [kg/m^3]	Mi [-]	Ma [kg/m^2]
1	BaumitFinoFil	0,0100	0,7000	1000,0	1200,0	10,0	0.0000
2	Beton hutný 1	0,2000	1,2300	1020,0	2100,0	17,0	0.0000
3	RockwoolVarir	0,2000	0,0440	840,0	27,0	1,0	0.0000
4	Inofin	0,0200	0,1600	960,0	1030,0	140000,0	0.0000

This software also calculated the ***U***-value:

$$\textbf{\textit{U}-value (roof) = 0.201 (W/m}^2\textbf{.K)}$$

7.1.3 Floor

Floor of the hospital contains 5 layers: Ceramic (0.015m); PPS Insulation (0.1m); Hydroizolace (0.01m); Concrete (0.15m); Hydroizolace (0.01m)



Simulation of the floor created by software:

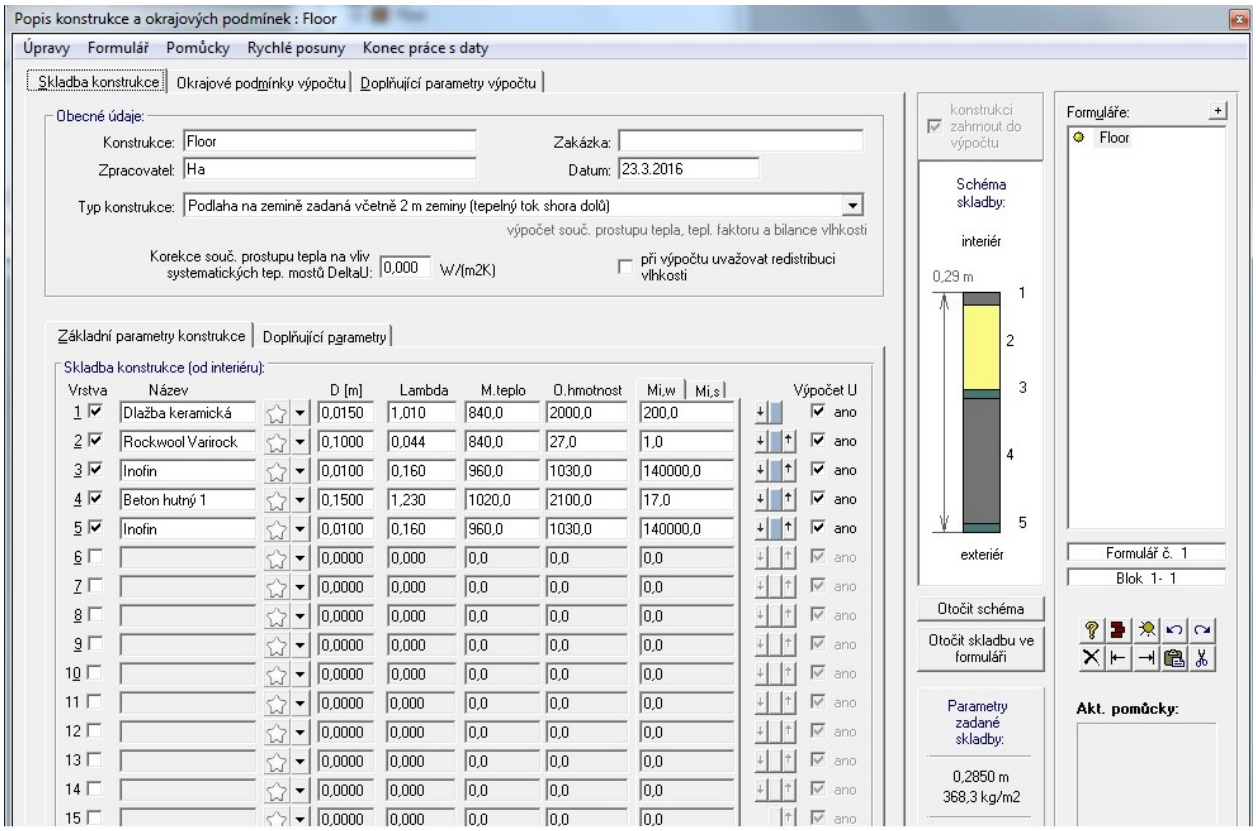


Fig. 7. Schema of floor created by software Stavební Fyzika

Table 7. Details parameters of the floor

Num.	Name	D [m]	Lambda [W/(m.K)]	c [J/(kg.K)]	Ro [kg/m^3]	Mi [-]	Ma [kg/m^2]
1	Dlažba keramic	0,0150	1,0100	840,0	2000,0	200,0	0.0000
2	RockwoolVarir	0,1000	0,0440	840,0	27,0	1,0	0.0000
3	Inofin	0,0100	0,1600	960,0	1030,0	140000,0	0.0000
4	Beton hutný 1	0,1500	1,2300	1020,0	2100,0	17,0	0.0000
5	Inofin	0,0100	0,1600	960,0	1030,0	140000,0	0.0000

This software also calculated the U-value:

***U*-value (floor) = 0.241 W/(m^2.K)**

7.1.4 Windows

This hospital use 2 layers window, with U-value = 1.1 W/(m^{2}·K). Thewindow's area ofeachroomisshowed in thefollowing table:

Table 8. Hospitalwindows' parameters

Room	Name	width (m)	height (m)	total Area of outside wall (m^2)	**total Area of windows** (m^2)
102	Sister's room	2.5	3.5	5.25	**3.50**
103	Doctor's room	2.5		5.25	**3.50**
104	Reception	2.5		5.25	**3.50**
105	Patient's room	5.4		11.34	**7.56**
106	Patient's room	5.4		11.34	**7.56**
107	Sister's room	2.5		5.25	**3.50**
110	*Bank*	*5.76*		*20.16*	0
111	Laboratory	*8.14*		17.13	**11.45**
114	*WC*	*6.32*		22.12	**0**
115	*WC*	*1.27*		4.45	**0**
101	Storage	2.5		21	**3.50**
108	Storage	2.5		21	**3.50**
109	*Operation*	*6.82*		*44.95*	*0*
113	Doctor's room	5.14		33.86	**7.20**

7.2 Air Distributions

Total Air Supply

The total air supply for each room (not include WC) is calculated based on the ACH:

(7)

$$\dot{V}_s = V_{room}.ACH$$

In that:

- $\dot{V}_s$: *Volume of supply air* $[m^3/h]$
- V_{room}: *Volume of the room* $[m^3]$
- ACH: *Air change per hour*

Note: Supply air must be greater or at least equal fresh air.

The requirement of ACH for each room is:

- 15 ACH for operation room
- 12 ACH for the bank and laboratory
- 6 ACH for other rooms
- 3 ACH for corridor

Fresh Air

Also following that standard, the fresh air that is supplied for each room must achieve the value as:

- 30 $m^3/(m^2.h)$: for Operation theaters
- 15 $m^3/(m^2.h)$: for the room that are next to the operation theaters
- 10 $m^3/(m^2.h)$: for other rooms
- And at least $30m^3$/(person*hour)

Return Air

For every room in hospital, the return air is a 15% less than the total air supply. Because we want to keep the pressure inside hospital higher than outside, so we can avoid dust or dirty air from outside flow inside.

For same reason, amount of the return air from the bank and operation is 10% less than supply air.

Exhaust air

Air from WC is exhausted directly to outside by distinct pipe. Exhausting air from WC will make pressure inside WC lower than other room. Therefore we can protect other rooms from dirty air of WC.

Detail information of air supply for each room is described in the following table:

Table 9. Hospital rooms' air distributions

Room	**Room's Name**	**Volume of room** [m^3]	**ACH**	**Fresh air** [m^3/h]	**Supply air** [m^3/h]	**Return Air** [m^3/h]	**Exhaust** [m^3/h]
101	Storage	32	-	-	-	-	-
102	Sister's room	32	6	360	360	306	-
103	Doctor's room	32	6	360	360	306	-
103a	WC	7	-	-	-		60
104	Reception	40	6	400	400	340	-
105	Patient's room	73	6	730	730	620.5	-
105a	WC	14	-	-	-		100
106	Patient's room	73	6	730	730	620.5	-
106a	WC	14	-	-	-		100
107	Sister's room	40	6	400	400	340	-
108	Storage	40	-	-	-		-
109	Operation room	126	15	1350	1890	1701	-
110	Bank	59	12	885	885	796.5	-
110a + 110b	Dirty filter	21	-	-	-		-
111	Laboratory	84	12	840	840	714	-
111a + 111b	Dirty filter	21	-	-	-		-
112	Andrologie	21	6	360	360	306	-

113	Embryology	85	6	850	850	722.5	-
113a	Cabin	-	-	-	-	-	-
114	Corridor	16	-	-	-	-	-
114 a b c	WC	32	-	-	-	-	200
115 a + 115b	WC	13	-	-	-	-	80
116a + 116b	WC	11	-	-	-	-	60
117a +117b	Corridor	219	3	-	650	650	
Total				**7265**	**8455**	**7425**	**600**

7.3 Heat Loss and Cooling Load

7.3.1 Heat loss

Calculations were done according to standard thermal conditions:

- Average temperature inside in winter: 20 °C
- Average temperature outside in winter: -15 °C

Heat Loss by Transmission

Heat loss by transmission can be calculated by using equation:

$$\boldsymbol{P = U_{value} * A * \Delta T} \quad (8)$$

Where:

P*= heat transferred per unit time [W]*

A *= area of the surface [m^2]*

ΔT = *temperature difference between temperature inside and outside [K]*

From that equation, we can calculate heat loss for each room as following:

Table 10. Heat loss by convection of each room

Num.	Room's Name	total Area of outside wall [m2]	total Area of windows [m2]	Area of the floor (roof) [m2]	Heat loss throw walls [W]	Heat loss throw window [W]	Heat loss throw floor [W]	Heat loss throw roof [W]
102	Sister's room	5.25	3.5	11.25	46.31	134.75	94.89	79.14
103	Doctor's room	5.25	3.5	11.25	46.31	134.75	94.89	79.14
104	Reception	5.25	3.5	14.4	46.31	134.75	121.46	101.30
105	Patient's room	11.34	7.56	31.10	100.02	291.06	262.36	218.82
106	Patient's room	11.34	7.56	31.10	100.02	291.06	262.36	218.82
107	Sister's room	5.25	3.5	14.4	46.31	134.75	121.46	101.30
110	Bank	20.16	0	21.2	*198.13*		178.80	149.12
111	Laboratory	17.10	11.396	29.96	150.77	438.75	252.67	210.73
114	WC	22.12	0	17.63	195.10	0.00	148.73	124.05
115	WC	4.45	0	3.54	39.20	0.00	29.89	24.93
101	Storage	21	3.5	11.25	185.22	134.75	94.89	79.14
108	Storage	21	3.5	11.25	185.22	134.75	94.89	79.14
109	Operation	44.94	0	44.94	*441.71*		379.10	316.18
113	Doctor's room	33.859	7.196	33.8726	298.64	277.05	285.72	238.29
	Corridor	-	-	80.3	-		677.33	564.91
				Total	**9870.15 [W]**			

Heat Loss by Ventilation

In the winter, the average temperature outside is -15°C, and we keep the temperature inside hospital around 20°C. So the difference temperature is:

$$\Delta\boldsymbol{\theta} = \boldsymbol{\theta}inside - \boldsymbol{\theta}outside$$

$$\Delta\boldsymbol{\theta} = 20 - (-15) = 35℃$$

In winter we work only with fresh air, therefore the heat loss by ventilation is only from fresh air:

$$\boldsymbol{P}_{FA} = \boldsymbol{V}_{FA} * \boldsymbol{\rho} * \boldsymbol{C}p * \Delta\boldsymbol{\theta}$$

$$\boldsymbol{P}_{FA} = \frac{7625}{3600} * 1.29 * 1.01 * 35$$

$$\boldsymbol{P}_{FA} = 96.5\ [kW]$$

From that we have:

Total heat loss = Heat loss by convection + Heat loss by ventilation

Total heat loss = 9.9 + 96.5 = **106.4 [kW]**

7.3.2 Cooling Load

In summer we will supply the air that has temperature 5°C lower than the temperature inside.

Average temperature outside is around 32°C, and we will keep temperature in hospital rooms around 25°C. Therefore, temperature of the supply air in summer will be 20°C.

Cooling load in summer is **35.8 kW** (calculated by software Stavební Fyzika).

8 DESIGN THE HVAC SYSTEM IN PARTICULAR CLEAN SPACE

The HVAC of this hospital is divided into 3 main sections:

- The Air Handling Units
- Piping systems
- Diffusers and Filters

The Air Handling Units

The Air handling system was design like the schema below:

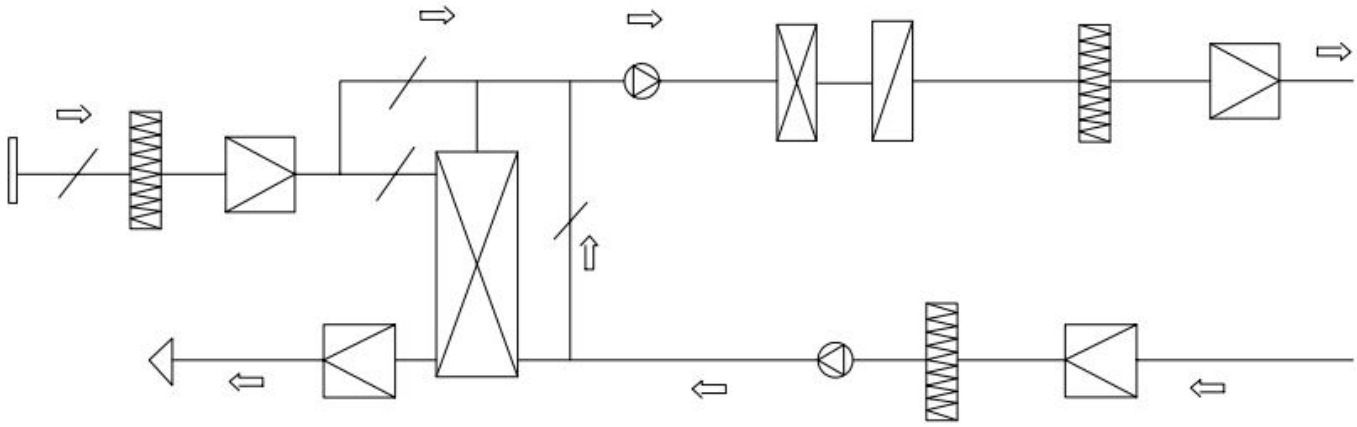


Fig. 8. Scheme of air handling units

In that:

Pump

Cooler

Heater

Filter

Heat Exchanger

Attenuator

The air handling units was designed to use re-circulate air, so we can save a lot of energy. The return air that come from inside will be used to mix with the fresh air to supply the hospital rooms. Only a part of return air is released outside, but before that, it will flow throw heat exchanger to preheat the fresh air, so more energy will be saved.

There is a valve that controls amount of the return air that will be mixed with fresh air, and amount of the return air that will be released outside.

Before the fresh air come into air handling systems, then it will flow throw attenuator, filter to heat exchanger. When temperature outside and inside is equivalent the fresh air will not flow into exchanger, but directly to heater.

Piping systems

Piping systems is divided to 3 parts:

- Supply air
- Return air
- Exhaust air

It is designed as following scheme:

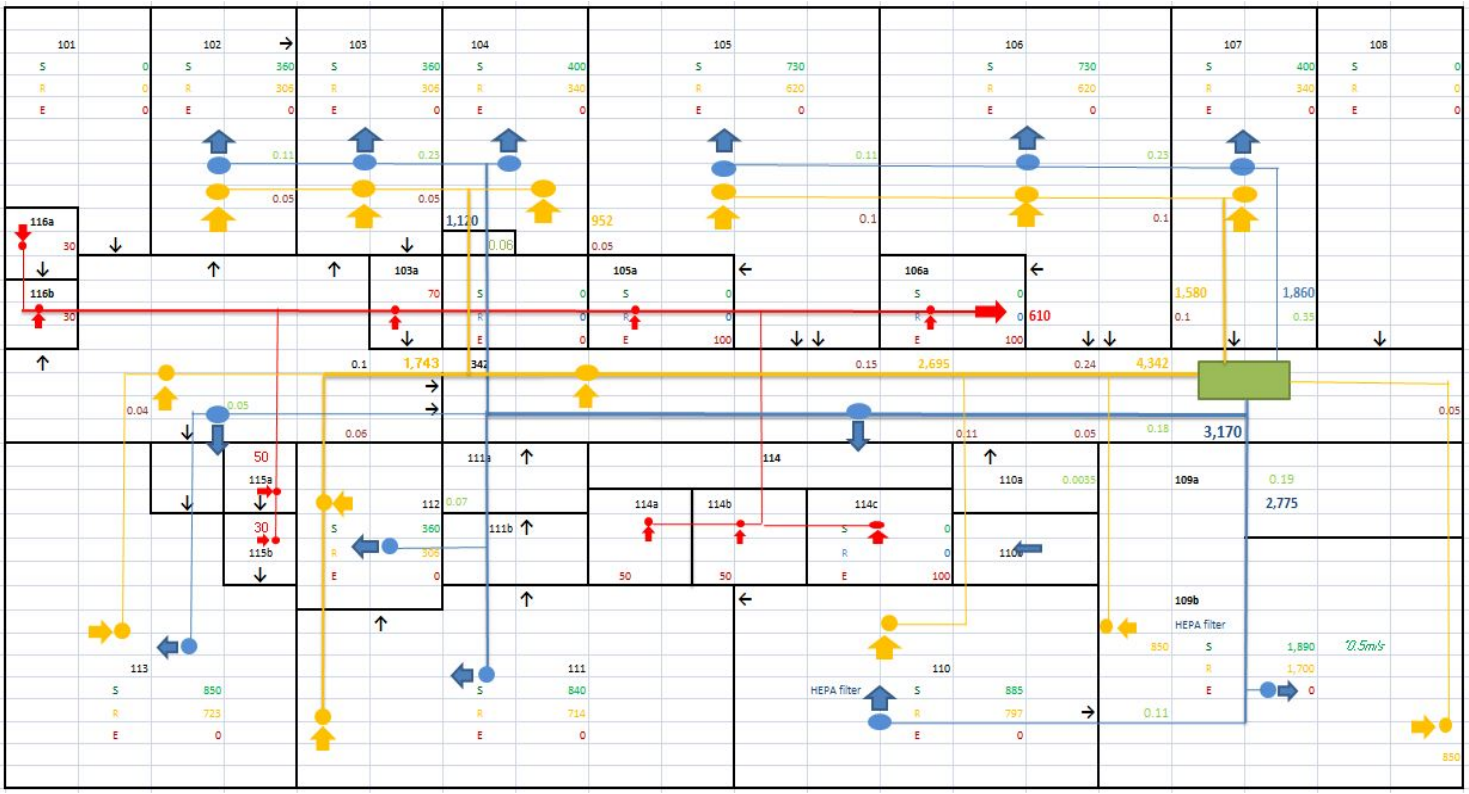


Fig. 9. Scheme of piping system

Piping systems are designed to satisfy: volume of required flow air, acceptable air flow speed, and fit area. Return air and Supply piping are connected by Air Handling systems. Exhaust air piping isn't connected with other piping, and it is used to suck the dirty air from WC directly out.

The main branch of return and supply air systems run along corridor, then it separates into smaller branches, going into the hospital rooms.

Detail of piping systems will be mentioned in the next part of this Thesis.

Diffusers and Filters

Diffusers are used to control direction and speed of the air that coming into the rooms. The operation room requires special diffusers to provide supply air, that flow at flow rate 5m/s in working place (1,5 meter from floor). Other rooms use the same diffuser.

Operation room also requires special filter. Before going inside operation room, supply air flow throw HEPA filter to provide highest class of clean air.

8.1 Heater

From parameters we calculated from II.2.1. we can draw H-x diagram for winter season:

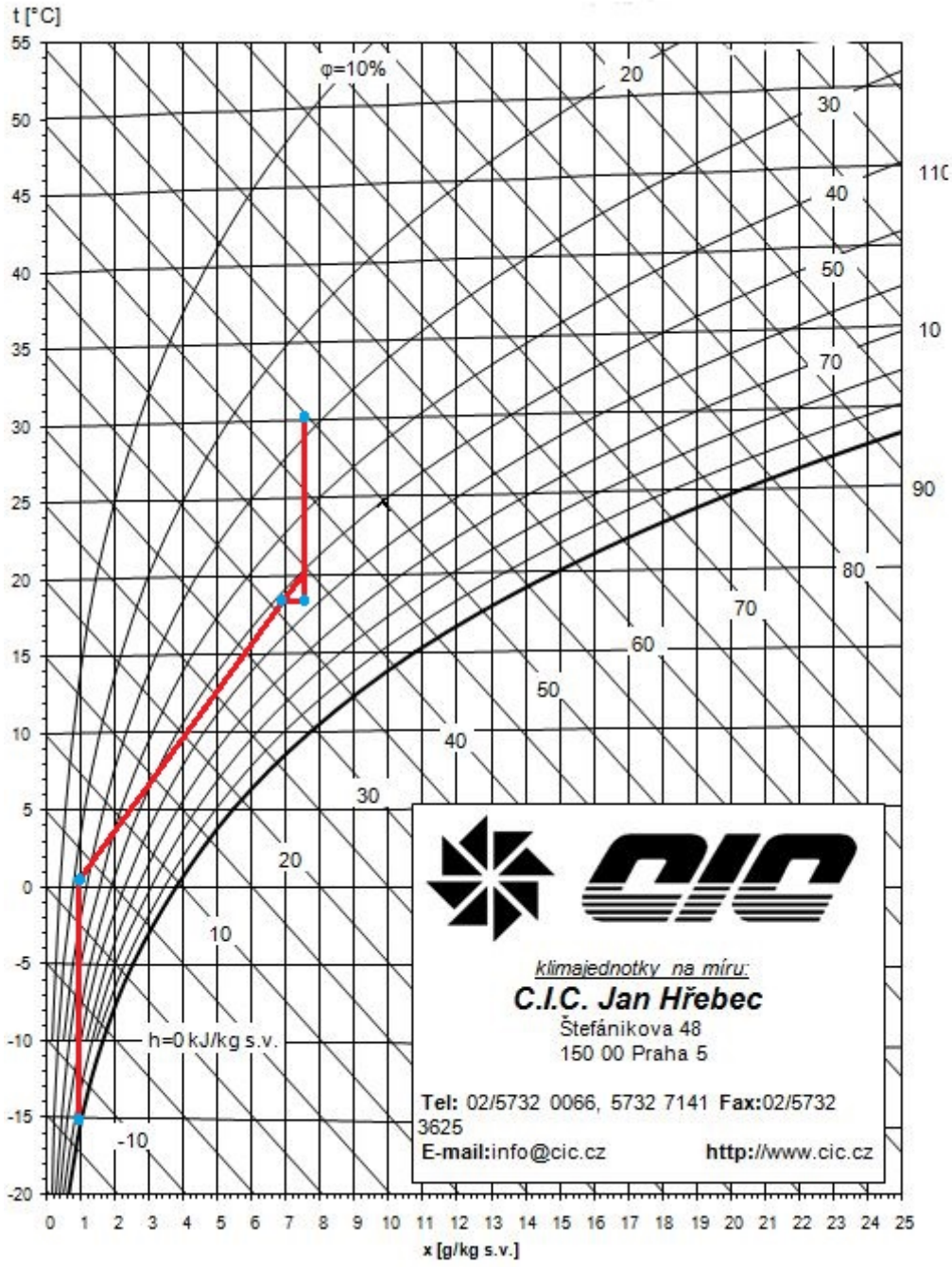


Fig. 10. H-x diagram for winter

From that we will calculate power of main heater:

$$\boldsymbol{P_{heater} = m_{SA} * \Delta h}$$

$$\boldsymbol{P_{heater} = V_{SA} * \ \rho * \Delta h}$$

$$\boldsymbol{P_{heate}} \ = \frac{8455}{3600} * \ 1.22 * (49.8 - 36.9)$$

$$\boldsymbol{P_{heater} = 37.0\ [KW]}$$

8.2 Cooler

From calculation of section II.2.2.we can draw H-x diagram for summer season:

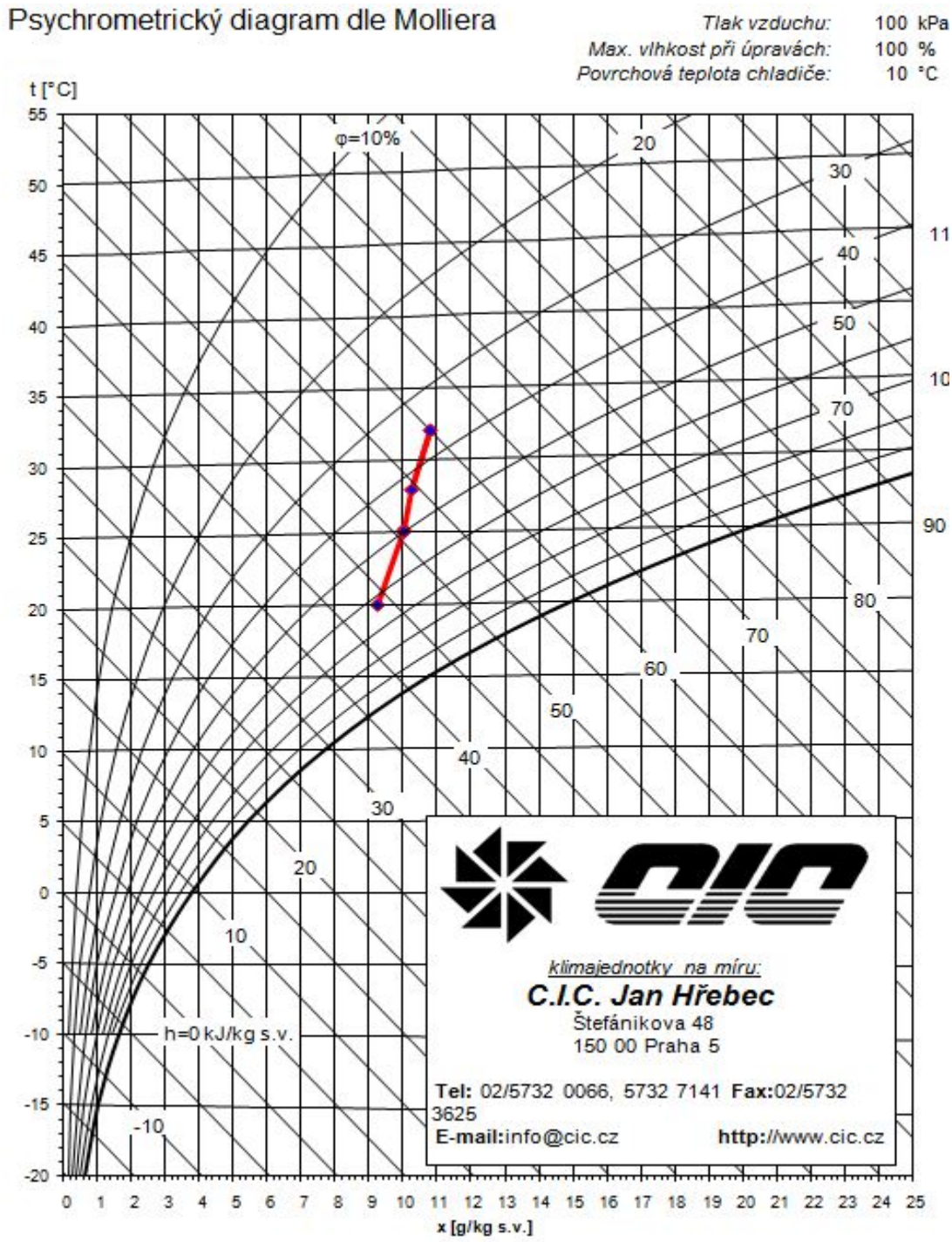


Fig. 11. H-x diagram for summer

From that we will calculate required power of cooler:

$$\boldsymbol{P_{cooler} = m_{SA} * \Delta h}$$

$$\boldsymbol{P_{cooler} = V_{SA} * \ \rho * \Delta h}$$

$$\boldsymbol{P_{cooler}} = \frac{8455}{3600} * \ 1.16 * (54.7 - 43.1)$$

$$\boldsymbol{P_{cooler} = 31.6\ [kW]}$$

Choosing Real Heater and Cooler

After calculating parameter of heater and cooler, I had chosen outdoor heat recovery device from Mitsubishi PURY-P500YSJM-A. This device can provide 63kW heating capacity, and 56kW cooling capacity. It can distribute surplus heat from cooling operations (and vice versa) to rooms where it is needed. This efficiency can result in energy savings of up to 30% over conventional systems. The picture and detail parameters of this device are showed below.

Fig. 12. Picture of Mitsubishi PURY-P500YSJM-A

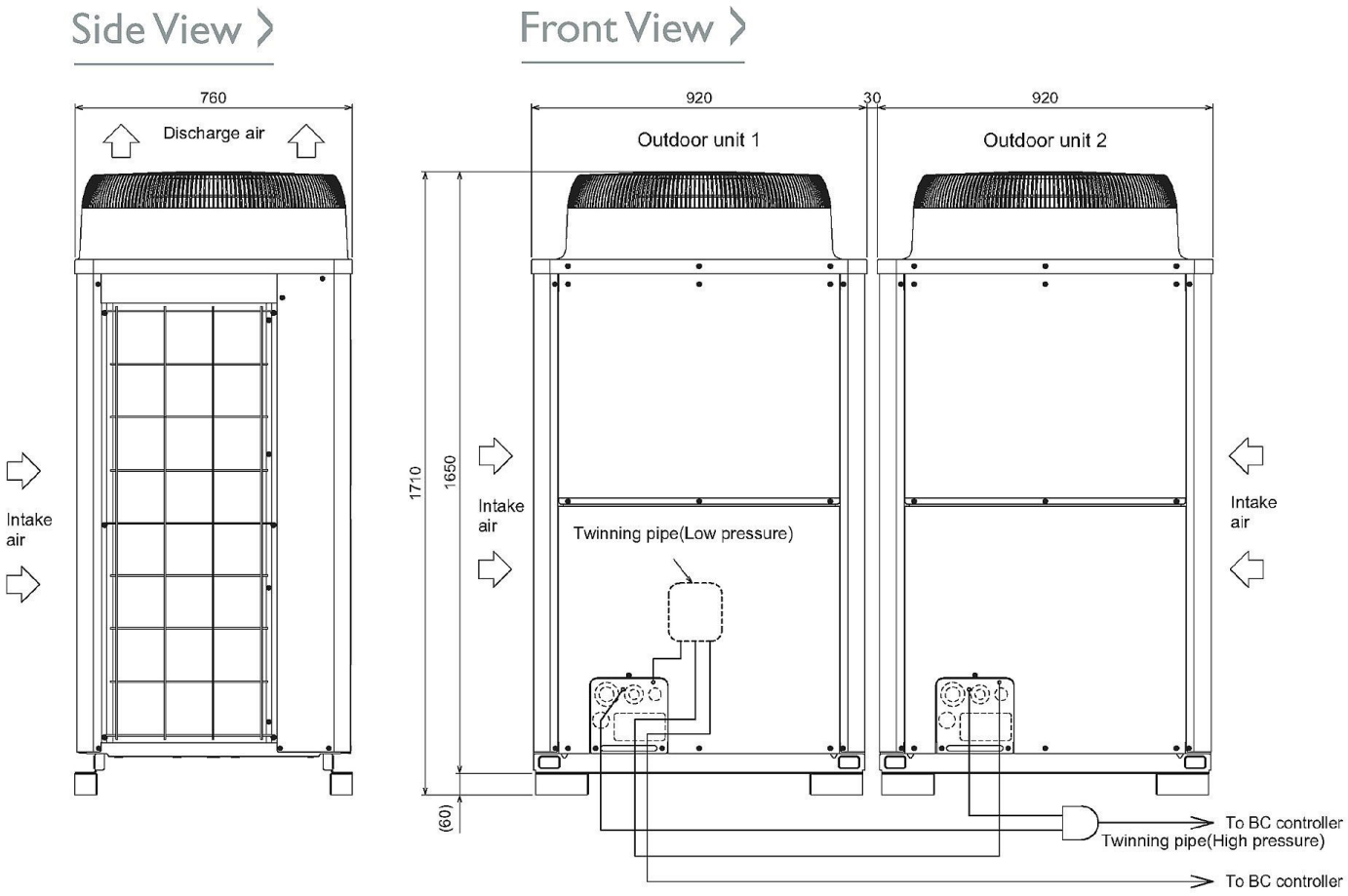


Fig. 13. Dimensions of Mitsubishi PURY-P500YSJM-A

Table 11. Parameters of Mitsubishi PURY-P500YSJM-A

Model name				PUHY-P500YSJM-A (-BS) connected with PFD series	
				Cooling	Heating
Capacity		*1	kW	56.0	63.0
Power source				3N ~ 380/400/415V 50/60Hz	
Power input			kW	13.60	13.20
Current			A	22.8/21.8/21.0	22.2/21.0/20.4
Set Model				PUHY-P250YJM-A(-BS)	PUHY-P250YJM-A(-BS)
Fan	Type × Quantity			Propeller fan x 1	Propeller fan x 1
	Airflow rate		m^3/min	170	170
	Motor output		kW	0.46 × 1	0.46 × 1
Compressor	Type			Inverter scroll hermetic compressor	Inverter scroll hermetic compressor
	Motor output		kW	6.8	6.8
	Crankcase heater		kW	0.035	0.035
Heat exchanger				Salt-resistant cross fin & copper tube	Salt-resistant cross fin & copper tube
Refrigerant/Lubricant				R410A/MEL32	R410A/MEL32
External finish				Pre-coated galvanized steel sheets (+ powder coating for -BS type) <MUNSEL 5Y 8/1 or similar>	
External dimension H×W×D			mm	1,710 (without legs 1,650) x 920 x 760	1,710 (without legs 1,650) x 920 x 760
Protection devices	High pressure protection			High pres. Sensor & High pres. Switch at 4.15MPa	
	Compressor			Over-heat protection	
	Fan			Thermal switch	
	Inverter circuit (COMP./FAN)			Over-heat protection, Over-current protection	
Refrigerant piping diameter	Liquid pipe			ø9.52 Brazed	ø9.52 Brazed
	Gas pipe			ø22.2 Brazed	ø22.2 Brazed
Sound pressure level		*2	dB(A)	61	
Net weight			kg	200	200

Note: *1. Cooling/Heating capacity indicates the maximum value at operation under the following condition.
<Cooling> Indoor: 27°CDB/19°CWB Outdoor: 35°CDB
<Heating> Indoor: 20°CDB Outdoor: 7°CDB/6°CWB
Pipe length: 7.5m Level difference: 0m

*2. It is measured in anechoic room.

** Installation/foundation work, electrical connection work, duct work, insulation work, power source switch, and other items shall be referred to the Installation Manual.

PFD-P500VM-E

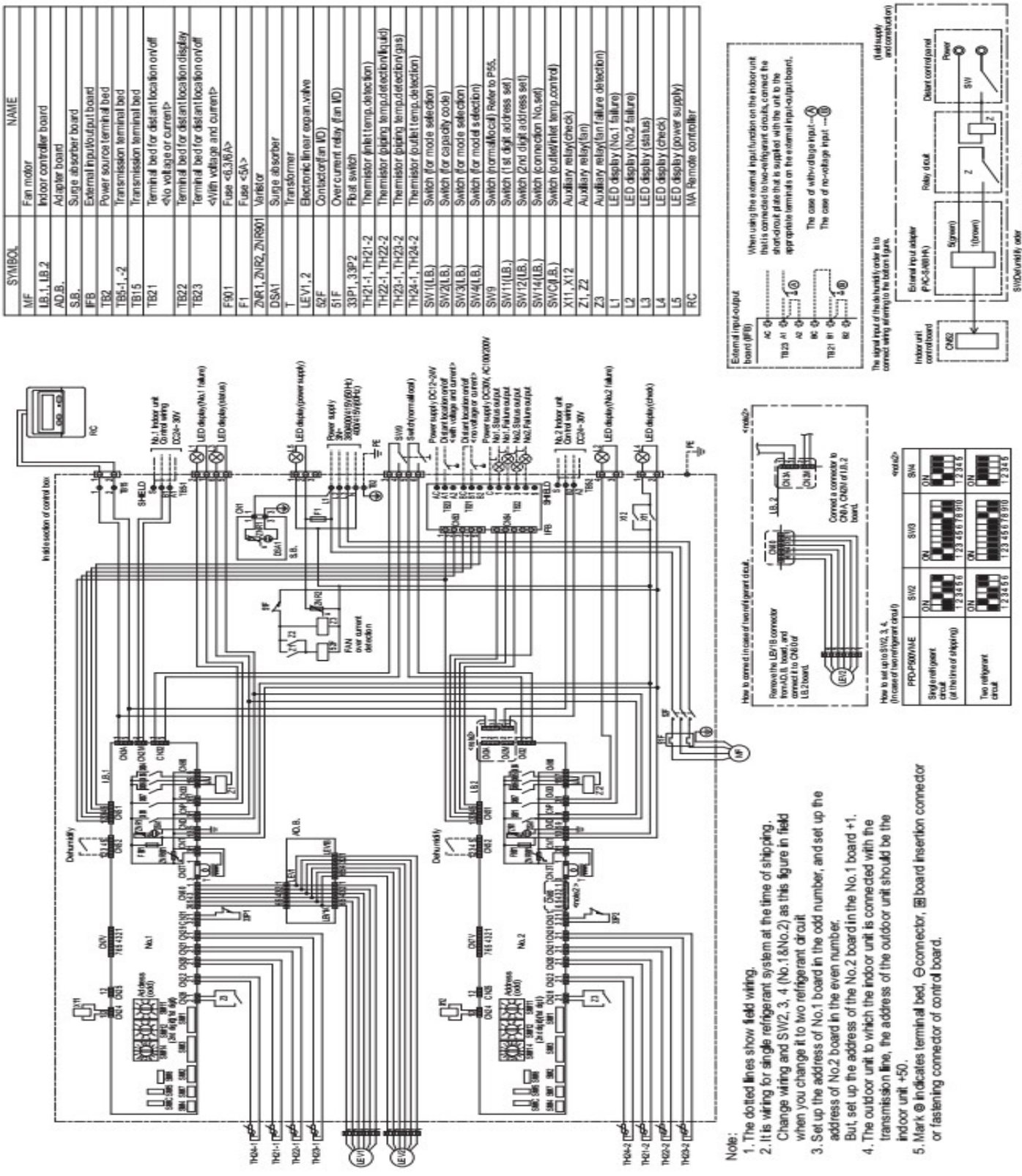


Fig. 14. Electrical wiring diagrams of Mitsubishi PURY-P500YSJM-A

8.3 Piping Systems

As I had mentioned before, the piping systems of this hospital is divided into 3 parts: supply air piping system; return air piping system and exhaust air piping system. The air flow speed is 5m/s to optimize air flow rate and noise.

8.3.1 Supply air piping system

The air is supplied for doctors', sisters' rooms and patients' rooms, bank, laboratory, and operation room. The main path of supply air runs along corridor, and then it separates into smaller branches, going into the rooms. The schema of supply air piping system is showed in following picture:

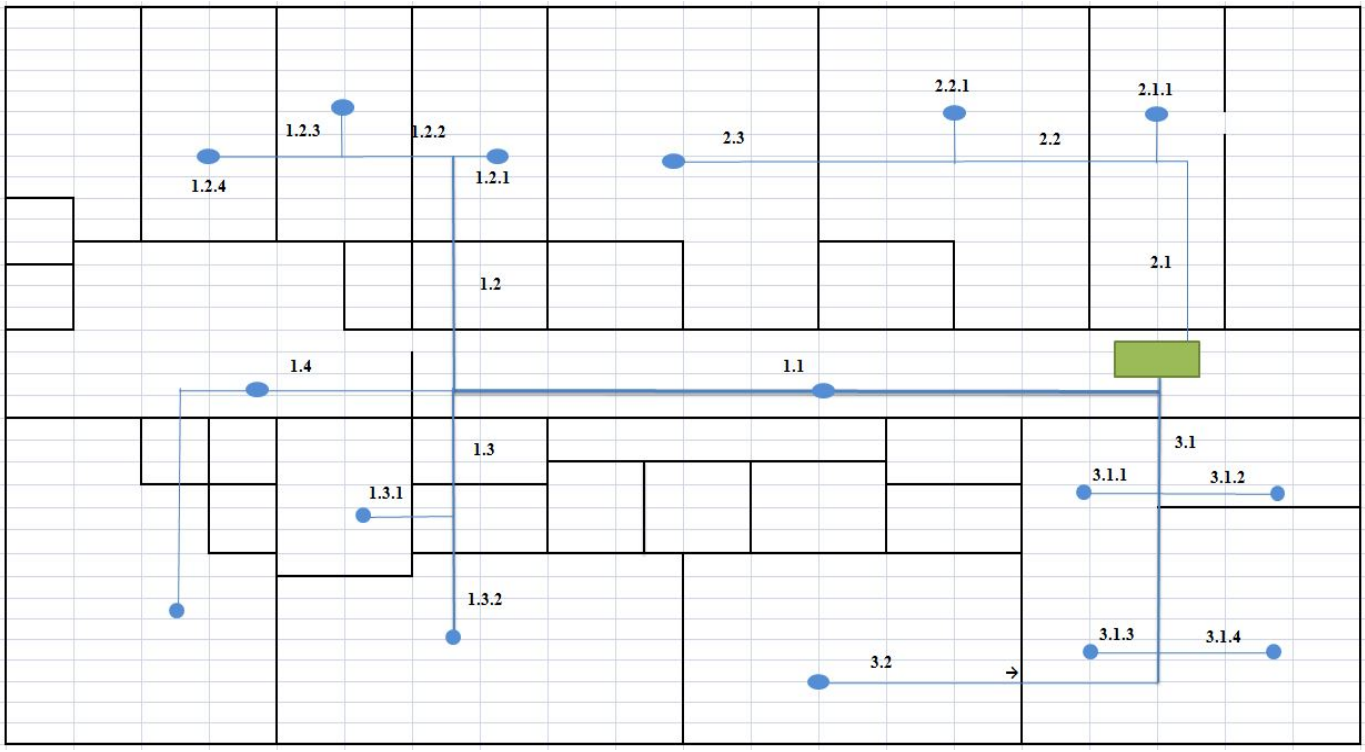


Fig. 15. Scheme of supply air piping system

The parameters of supply air piping system, such as volume of air flow, width (a), length (b), and length of pipe (l) are described in following table:

Table 12. Parameters of supply air piping system

Path num.	Volume of air flow $[m^3/s]$	a [m]	b [m]	Area of pipe $[m^2]$	l [m]
1.1	3820	0.5	0.4	0.2	14
1.2	1120	0.3	0.2	0.06	6.5
1.2.1	400	0.25	0.1	0.025	0.5
1.2.2	720	0.2	0.2	0.04	3
1.2.3	360	0.2	0.1	0.02	0.5
1.2.4	360	0.2	0.1	0.02	2.2
1.3	1200	0.3	0.2	0.06	2.2
1.3.1	360	0.2	0.1	0.02	1.5
1.3.2	840	0.25	0.2	0.05	3.5
1.4	850	0.25	0.2	0.05	10.5
2.1	1860	0.5	0.2	0.1	6.2
2.1.1	400	0.5	0.5	0.25	0.5
2.2	1460	0.4	0.2	0.08	5
2.2.1	730	0.2	0.2	0.04	0.5
2.3	730	0.2	0.2	0.04	5
3.1	2775	0.5	0.3	0.15	5.3
3.1.1	473	0.2	0.15	0.03	0.5
3.1.2	473	0.2	0.15	0.03	2.1
3.1.3	473	0.2	0.15	0.03	0.5
3.1.4	473	0.2	0.15	0.03	2.1
3.2	885	0.25	0.2	0.05	6

8.3.1.1 Pressure Loss in Supply Air Piping Systems

Equivalent Diameter (De)

The equivalent diameter is the diameter of a circular duct or pipe that gives the same pressure loss as an equivalent rectangular duct or pipe.The equivalent diameter of a rectangular tube or duct can be calculated as:

$$\boldsymbol{De} = \frac{4.a.b}{2(a+b)} \tag{9}$$

Pressure loss due to friction can be calculated as:

$$\Delta \boldsymbol{P_{\lambda}} = \frac{\boldsymbol{\lambda.v^2.\rho.l}}{\boldsymbol{De}} \tag{10}$$

Where:

- $\Delta \boldsymbol{P_{\lambda}}$*: Pressure loss due to friction [Pa]*
- $\boldsymbol{\lambda}$*: friction coefficient (0.16)*
- $\boldsymbol{v}$*: air velocity [m/s]*
- $\boldsymbol{\rho}$*: air density (1.204) [kg/*$\boldsymbol{m^3}$*]*
- $\boldsymbol{l}$*: length of pipe [m]*
- $\boldsymbol{De}$*: Equivalent Diameter* [m]

Velocity pressure can be calculated as:

$$\Delta \boldsymbol{P_{\xi}} = \frac{\boldsymbol{\xi.v^2.\rho}}{\boldsymbol{2}} \tag{11}$$

Where:

- $\boldsymbol{\xi}$*: Fitting loss coefficient*
- $\boldsymbol{v}$*: air velocity [m/s]*

- $\boldsymbol{\rho}$*: air density [kg/*$\boldsymbol{m^3}$*]*

Pressure loss of equipment (ΔP_{equip}) is pressure loss when air flow through equipments, such as filters, diffusers, heaters, coolers, etc.

Total Pressure:

$$\Delta P_{total} = \Delta P_{\lambda} + \Delta P_{\xi} + \Delta P_{equip} \quad (12)$$

From above equations, we can calculate pressure loss for each path of supply air piping systems:

Table 13. Pressure loss of supply air piping systems

Path Number	De [m]	v [m/s]	l [m]	ΔP_{λ} [Pa]	ξ [-]	ΔP_{ξ} [Pa]	ΔP_{equip} [Pa]	ΔP_{total} [Pa]	ΔP_{path} [Pa]
1.1	0.444		14	296.352	0	0		296.352	
1.2	0.240		6.5	254.800	0.75	22.05		276.850	
1.2.1	0.143		0.5	32.928	0.11	3.234	120	156.162	729.364
1.2.2	0.200		3	141.120	0.8	23.52		164.640	
1.2.3	0.133		0.5	35.280	0.11	3.234	120	158.514	896.356
1.2.4	0.133		2.2	155.232	0.05	1.47	120	276.702	1014.544
1.3	0.240		2.2	86.240	0.75	22.05		108.290	
1.3.1	0.133	5	1.5	105.840	0.66	19.404	120	245.244	649.886
1.3.2	0.222		3.5	148.176	0.57	16.758	120	284.934	689.576
1.4	0.222		10.5	444.528	0.57	16.758	120	581.286	**877.638**
2.1	0.286		6.2	204.154	0.75	22.05		226.204	
2.1.1	0.500		0.5	9.408	0.11	3.234	120	132.642	358.846
2.2	0.267		5	176.400	0.75	22.05		198.450	
2.2.1	0.200		0.5	23.520	0.11	3.234	120	146.754	571.408
2.3	0.200		5	235.200	0.05	1.47	120	356.670	781.324

3.1	0.375		5.3	132.966	0	0		132.966	
3.1.1	0.171		0.5	27.440	1.5	44.1	500	236.540	704.506
3.1.2	0.171		2.1	115.248	1.5	44.1	500	324.348	792.314
3.1.3	0.171		0.5	27.440	1.5	44.1	500	236.540	704.506
3.1.4	0.171		2.1	115.248	1.5	44.1	500	324.348	792.314
3.2	0.222		6	254.016	0.57	16.758	120	390.774	523.740

From that table, we can see the biggest pressure loss of supply air piping systems is:

$$\Delta P_{Path\ 1.4} = 877.64\ [Pa]$$

8.3.2 Return air piping system

The schema of return air piping system is showed in following picture:

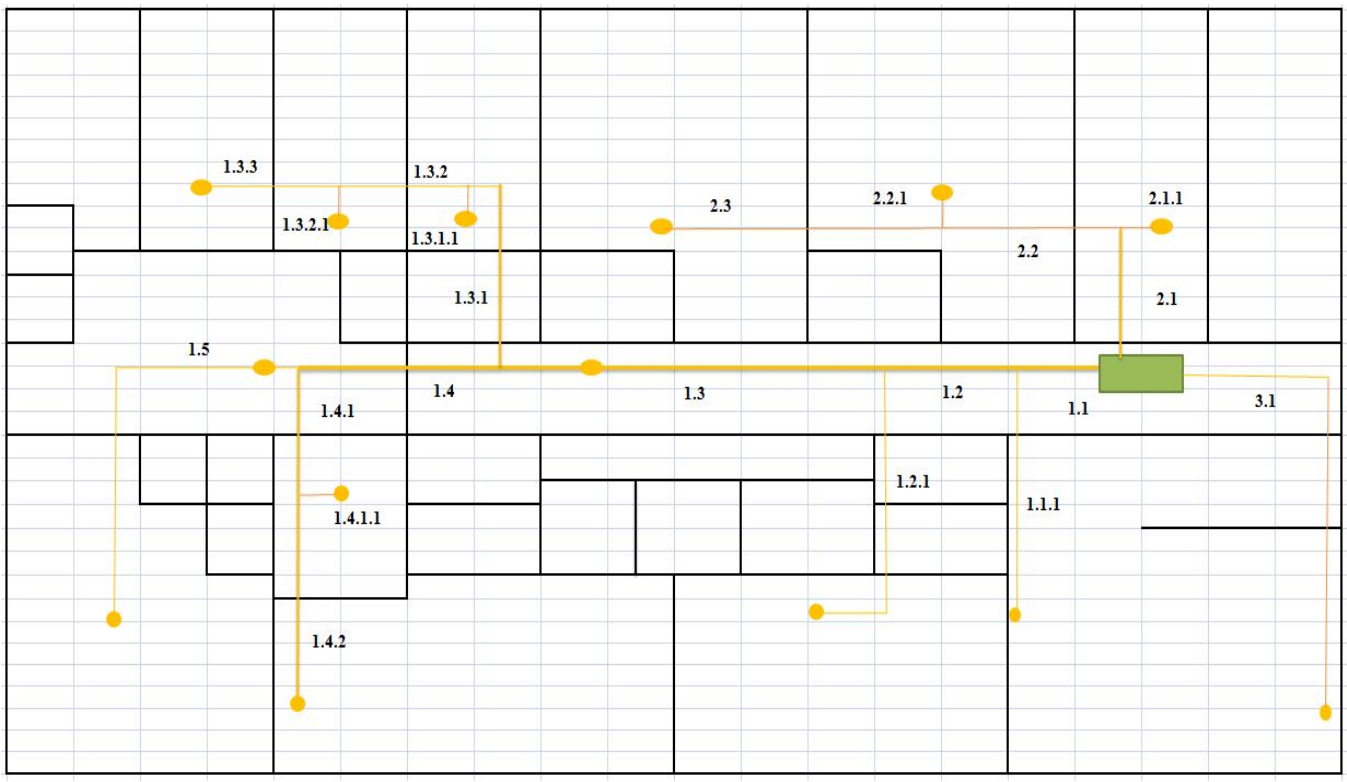


Fig. 16. Scheme of return air piping system

The parameters of return air piping system, such as volume of air flow, width (a), length (b), and length of pipe (l) are described in following table:

Table 14. Parameters of return air piping system

Path number	Volume of air flow $[m^3/s]$	a [m]	b [m]	Area of pipe $[m^2]$	*l* [m]
1.1	4992	0.7	0.4	0.28	2.53
1.1.1	850	0.25	0.2	0.05	4.5
1.2	4142	0.6	0.4	0.24	2.58
1.2.1	797	0.2	0.2	0.04	3.33
1.3	3345	0.5	0.4	0.2	7.44
1.3.1	952	0.25	0.2	0.05	4.5
1.3.1.1	340	0.2	0.1	0.02	0.5
1.3.2	612	0.2	0.175	0.035	2.64
1.3.2.1	306	0.15	0.1	0.015	0.5
1.3.3	306	0.15	0.1	0.015	2
1.4	1743	0.5	0.2	0.1	3.28
1.4.1	1020	0.275	0.2	0.055	2.78
1.4.1.1	306	0.15	0.1	0.015	0.75
1.4.2	714	0.2	0.2	0.04	4.78
1.5	723	0.2	0.2	0.04	9.6
2.1	1580	0.4	0.2	0.08	2.5
2.1.1	340	0.2	0.1	0.02	1
2.2	1240	0.3	0.2	0.06	4.2
2.2.1	620	0.2	0.172	0.035	4.7
2.3	620	0.2	0.175	0.035	5.5
3.1	850	0.25	0.2	0.05	11

8.3.2.1 Pressure Loss in Return Air Piping Systems

Following the same steps in II.3.3.1.a, we also have the table that shows the pressure loss for each path in return air piping systems:

Table 15. Pressure loss of return air piping systems

Path Number	De [m]	v [m/s]	L [m]	ΔP_{λ} [Pa]	ξ [-]	ΔP_{ξ} [Pa]	ΔP_{total} [Pa]	ΔP_{path} [Pa]
1.1	0.509		2.53	46.754	0	0	46.754	
1.1.1	0.222		4.5	190.512	1.1	32.34	222.852	269.606
1.2	0.480		2.58	50.568	0	0	50.568	
1.2.1	0.200		3.33	156.643	1	29.4	186.043	232.798
1.3	0.444		7.44	157.490	0.2	5.88	163.370	
1.3.1	0.222		4.5	190.512	1.75	51.45	241.962	
1.3.1.1	0.133		0.5	35.280	0.11	3.234	38.514	541.168
1.3.2	0.187		2.64	133.056	0.05	1.47	134.526	
1.3.2.1	0.120		0.5	39.200	0.11	3.234	42.434	679.614
1.3.3	0.120		2	156.800	0.05	1.47	158.270	**795.450**
1.4	0.286	5	3.28	108.004	0.06	1.764	109.768	
1.4.1	0.232		2.78	112.939	0.75	22.05	134.989	
1.4.1.1	0.240		0.75	29.400	0.66	19.404	48.804	444.485
1.4.2	0.400		4.78	112.426	0.57	16.758	129.184	634.633
1.5	0.400		9.6	225.792	0.57	16.758	242.550	613.010
2.1	0.533		2.5	44.100	0	0	44.100	
2.1.1	0.267		1	35.280	0.52	15.288	50.568	94.668
2.2	0.480		4.2	82.320	0.52	15.288	97.608	
2.2.1	0.370		4.7	119.542	0.52	15.288	134.830	276.538
2.3	0.373		5.5	138.600	0.05	1.47	140.070	281.778
3.1	0.444		11	232.848	0.44	12.936	245.784	431.182

From that table, we can see the biggest pressure loss of return air piping systems is:

$$\Delta P_{Path\ 1.3.3} = 795.45\ [Pa]$$

8.3.3 Exhaust Air Piping System

The schema of exhaust air piping system is showed in following picture:

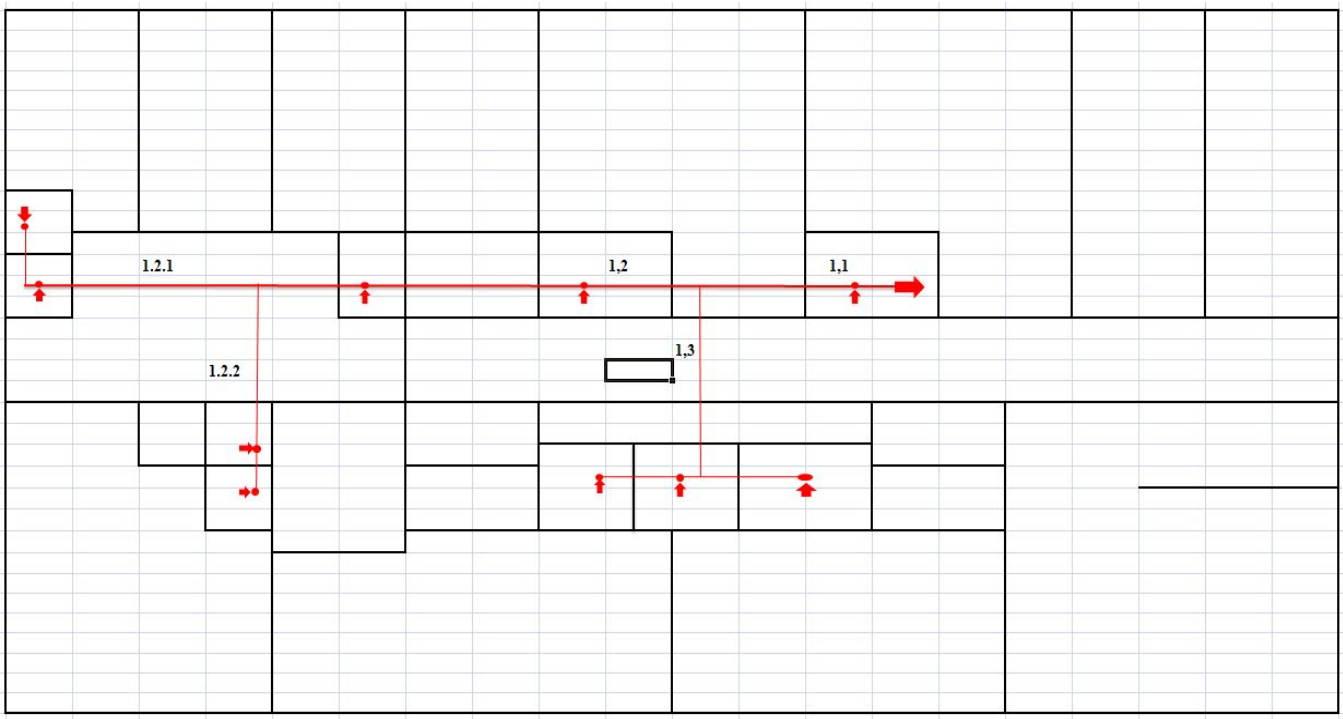


Fig. 17. Scheme of exhaust air piping system

The parameters of exhaust air piping system, such as volume of air flow, width (a), length (b), and length of pipe (***l***) are described in following table:

Table 16. Parameters of exhaust air piping system

Path number	Volume of air flow $[m^3/s]$	a [m]	b [m]	Area of pipe $[m^2]$	***l*** [m]
1.1	610	0.2	0.175	0.035	4
1.2	310	0.15	0.1	0.015	6.5
1.2.1	230	0.15	0.1	0.015	6
1.2.2	80	0.1	0.05	0.005	4
1.3	200	0.15	0.1	0.015	6.7

8.4 Diffusers

8.4.1 Diffusers of Operation Room

For operation room, I divided supply air to 4 smaller branches. Each branch has 1 diffuser and supply around 500 m^3 of air per hour. I had chosen diffusers from manufacture TROX, type PASS-Q-AKH-ZL-MO/623x500/9/0/RAL9010/0, in that:

- Q (Face plate): Square
- AKH (Plenum box): with plenum box
- ZL (Supply air): Supply air
- MO (Volume control damper): Spigot with volume flow rate control, without lip-seal
- 623x500 (Size): 623mm x 500mm
- 9 (Discharge possibility): Discharge possibility
- 0 Surface options for: Powder coated according to RAL 9010 matt, 25% brilliance
- RAL9010: Select the RAL-CLASSIC Color number: RAL 9010
- 0 Surface for Nozzles: nozzles in RAL 9010

Picture and parameters of this diffuser is showed below:

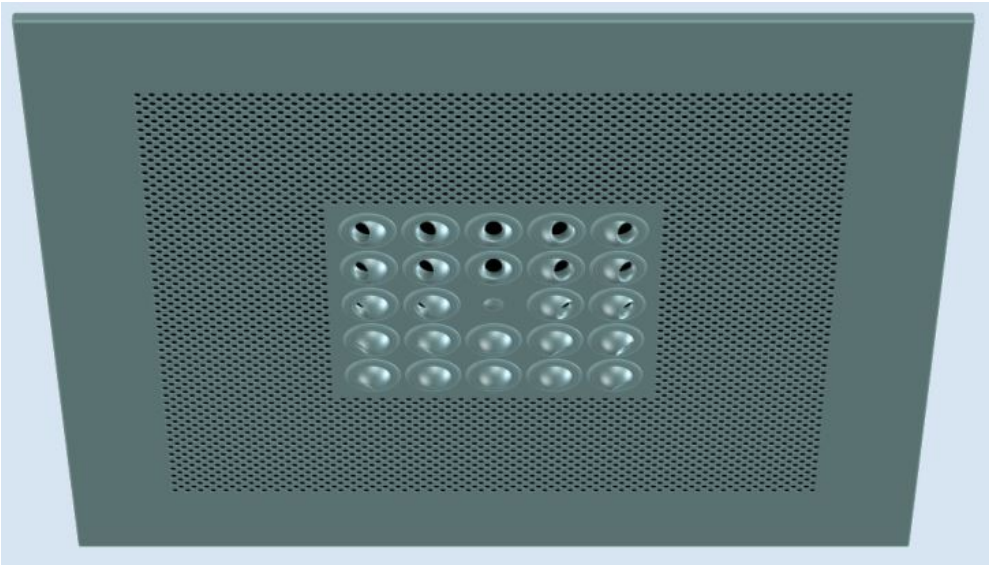


Fig. 18. Diffuser of operation room

Table 17. Parameters of diffuser type PASS-Q-AKH-ZL-MO/623x500/9/0/RAL9010/0

Air flow rate [m³/h]	500
Δp [Pa]	28
LWA [dB(A)]	38

This diffuser provides supply air with velocity of 0.4-0.5m/s at the working point (1.5 m from the floor). This air velocity is suitable for operation room.

8.4.2 Diffuser of Other Rooms

All rooms of the hospital, except operation room, use the same diffuser to supply 350 to 900 m^3 of air per hour. I had chosen diffuser from manufacture TROX, typeADLR-ZH-M/4/0/0/0/RAL 9010, in that:

- ZH: Connection: Side entry plenum, Supply air
- M: Volume control damper: Volume control damper with adjustment lever
- 4(Size): Size Diffuser
- 0 Sub-frame: without sub-frame

Picture and parameters of this diffuser is showed below:

Fig. 19. Diffuser type DLQ-AK-M/600/0/0/0/RAL 9010

Table 18. Parameters of diffuser type DLQ-AK-M/600/0/0/0/RAL 9010

Air flow rate[m³/h]	350 – 900		
Damper angle	0°	45°	90°
ΔPt [Pa]	9 - 56	12 - 79	26 – 174
LWA [dB(A)]	18 - 48	19 - 49	21 - 51

8.5 Attenuator

For this hospital, I used splitter sound attenuator type XKA:XKA200 / 700x560. In that:

- XKA: Energy-saving splitters
- 200: Splitter thickness [mm]
- 700: Width of attenuator [mm]

- 560: Length of attenuator [mm]

Total pressure drop when air flows throw this attenuator is: $\Delta \boldsymbol{p} = \mathbf{45}\ [\boldsymbol{PA}]$

Fig. 20. Attenuator XKA200-150-2

8.6 Filter

8.6.1 Filter of Air Handling Unit

Inside air handling unit, I used 4 filters from manufacturer Camfil France: Aeropleat Metal, size 740mm*620mm. Filter efficiency is G4 follow EN779:2012 standard, which means average dust arrestance is about 90%.

Maximum pressure drop when air flows throw 1 filter is $\mathbf{250\ Pa}$.

Fig. 21. Filter Aeropleat Metal

8.6.2 HEPA Filter for Operation Room

Operation room requires clean air at high level, so I had used 4 HEPA filters from manufacturer Camfil France: MD14-HL, size 610mm*610mm. Each of those filters can provide 600m^3/h of clean air, and can cover 9.5m^2. This filter is H14 compact filter-diffuser for clean room, which means efficiency of filters is ≥99.995% follow EN1822:2009 standard. They are ready to install.

Maximum pressure drop of this filter is 500 Pa**.**

Fig. 22. HEPA filter MD14-H

8.7 Fans

8.7.1 Fan for Supply Air

Total pressure drop for supply air is sum of pressure drop of supply air piping system, heat recovery device, 2Aeropleat Metal filters and 2 attenuators.

$$\Delta \boldsymbol{p_{supply}} = \Delta \boldsymbol{p_{piping}} + \Delta \boldsymbol{p_h} + \Delta \boldsymbol{p_{filter}} + \Delta \boldsymbol{p_{att.}}$$

$$\Delta \boldsymbol{p_{supply}} = 877.6 + 150 + 2.250 + 2.45$$

$$\Delta \boldsymbol{p_{supply}} = \mathbf{1617.6}\ [\boldsymbol{Pa}]$$

8.7.2 Fan for Return Air

Total pressure drop for return air is sum of pressure drop of return air piping system, 1 Aeropleat Metal filters and 2 attenuators.

$$\Delta \boldsymbol{p_{return}} = \Delta \boldsymbol{p_{piping}} + \Delta \boldsymbol{p_{filter}} + \Delta \boldsymbol{p_{att.}}$$

$$\Delta \boldsymbol{p_{return}} = 795.5 + 250 + 45$$

$$\Delta \boldsymbol{p_{return}} = \mathbf{1090.5}\ [\boldsymbol{Pa}]$$

9 DESIGN OF CONTROL AND MONITORING SYSTEMS

In order to simplify the operation and control of technical systems in the buildings, and to provide flexibility of their usage, and intelligent centralized control system was proposed. Control system offers a possibility of simple and user-friendly control of internal environmental parameters in the building. Any person without specific technical knowledge can manipulate it by simply setting parameters through control point (touch panel or PC).

Control system was realized using KNX building automation technologies. Being a universally accepted building automation protocol, it offers a broad variety of hardware suppliers, which can be combined in the optimal way from financial and functional points of view. Besides, a KNX system may be controlled by any functional control device, from microcontroller to a PC, depend on the scale of the project [15]. An important advantage is flexibility of KNX system. Architecture can be easily modified, elements can be added or remove without affecting the whole system. It is a low voltage system of 30V DC and current of 25 mA. Control bus is routed in parallel with 230V power circuit.

Control system connections of air handling units are presented in the following picture:

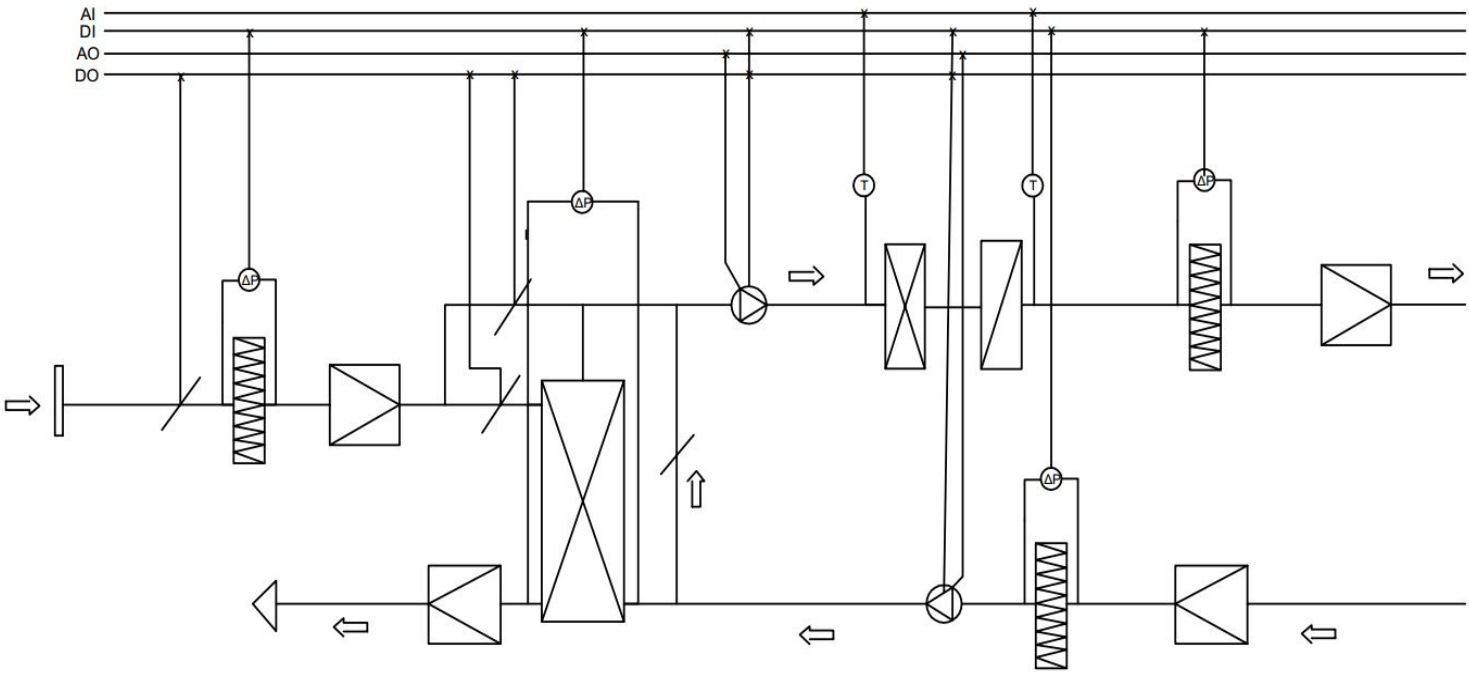


Fig. 23. Control system connections of air handling units

The inputs and outputs of control system include:

- Analog output on fans, which give us information that fans are running or not.
- Digital output and input of fans, which tell us the speed of fan, and it is used to control fan speed by alternating frequency of supplied current. Current value of shaft speed is measured by potentiometer or Hall sensor, and supplied as an output of the fans.
- Digital output of damper: it provide information that the damper is close or open.
- Pressure different between 2 sides of filter, from that we can know when the filter will be dirty and clean them.
- We also measure pressure different between 2 sides of heat exchanger to check when it is needed to be clean.
    - Analog input of the air temperate before and after heat recovery unit, which is used to control the heating or cooling power.

CONCLUSION

Scope oft his Thesis is designing a HVAC system for a hospital, with focus on supplying clean air, provision of comfort for doctors, nurses and patients. It also describes and uncovers thermal-technical parameters of the hospital to choose suitable equipments for HVAC system. Moreover, this Thesis explains the way of controls, monitoring and communication using and KNX-based inteligent control system.

Thesis is structured into two major parts: theoretical and practical. Theoretical part presents commont thermal properities, definition and classification of clean room, HVAC system and its components. It also includes requirements of building physics and HVAC system for hospital. While practical part is focus con calculation and actual design of HVAC and control system.

Frist chapter of theoretical part presents some basic thermal properities, suck as: thermal conductivity, thermal transmittance and thermal resistance. It also includes definition of heat loss and cooling load, and the way how they are calculated.

The second part presents overview of clean room, and its classification following standards FS 209E and ISO 14644-1. In that are some basic requirements of clean room and this part also mentions that HEPA filter is the main component in designing clean room.

The third and fourth chapters contain information about HVAC system, its components and how they are implemented. These two parts also include the information of ventilation, air distribution, air change rate requirement, and how room pressure can be controlled. In that are also the main components of HVAC system that are design for hospital: air handling units, HEPA filters, and air distribution pipes.

The requirements of building physics and HVAC system for clean room are mentioned in fifth chapter. Building physic requirements are following standards ČSN 73 0540, ČSN EN ISO 13788 and ČSN EN ISO 6946.

The seventh part is design of relevant hospital. This part presents the structures of outside walls, floor, roof, windows and their thermal properties. Details of air distribution ductwork are also mentioned. Based on building physic design and ventilation, heat loss and cooling load are calculated in this part.

HVAC system design and implementation are presented in eighth chapter, which is the main chapter of this Thesis. Parameters of each component of HVAC are calculated, and from that suitable equipments can be chosen. Piping systems design is also included in this part. This

eighth part shows us how the piping system can be implemented, the branches, size of duct and the positions, where the air is supplied. Diffusers for each rooms and HEPA filters selection can also be found on this part.

Last chapter deals with design of control and monitoring systems. In order to simplify the operation and control of technical systems in the buildings, and to provide flexibility of their usage, and intelligent centralized control system was proposed. Control system offers a possibility of simple and user-friendly control of internal environmental parameters in the building. Any person without specific technical knowledge can manipulate it by simply setting parameters through control point (touch panel or PC).

Overall, this Thesis presents definition of thermal properties, overview of clean room. It also presents HVAC system and its components. This Thesis provides solution for design and implementation a HVAC system for clean room, and specifically for hospital with specific requirements.

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