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T639KA

HEATING SYSTEM BASED ON HEAT PUMP TECHNOLOGY

Water Treatment Plant

Bachelor's thesis

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DESCRIPTION

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Heating system based on heat pump technology for water treatment plant

Abstract

The subject of this thesis was the deep studying of using heating systems based on heat pump technology and design of the heating system for water supply treatment plant building in Russian Federation. In spite of the big positive experience in European countries the heat pump technology is not widely spread as an alternative way of producing heat energy in Russian Federation. Moreover there is no experience of using this technology for such industrial buildings as water supply treatment plant, wastewater treatment plants and other industrial buildings where this technology might become one of the most suitable and economically reasonable. The role of this thesis is to show that this technology can be used as economically reasonable and environmentally friendly way of producing heat energy for the heating purposes for different types of industrial buildings.

The thesis presents the short heat pump history helps to understand the process of appearance and development of this technology, and those economical and environmental problems, which became a cause of appearance of heat pump technology. The needed calculations to design the heating system and to estimate capital and operation costs of installation and using this technology were made.

The calculations helped to prove that the heat pump technology in combination with good thermal insulation of the building envelope shows very good results of economical effectiveness of its using. And the existent problem of careless consumption of fossil fuel to produce heat energy in Russian Federation might be solved by reducing the fuel consumption as a result of using heat pump technology, which has high energy efficiency and much more lower consumption of primary fuel.

Subject headings, (keywords)

Heating system, Heat pump technology, Low potential heat, GSHP, Environmentally friendly, Fuel consumption, Industrial buildings, Energy piles, Trench collector, Underfloor heating.

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Remarks, notes on appendices

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CONTENT

LIST OF ABBREVIATIONS

GSHP – Ground source heat pump

GWHP – Ground water heat pump

GWP – Global warming potential

HCFC – Hydrochlorofluorocarbon

HHV – Higher heating value

IGSHPA – International Ground Source Heat Pump Association

JAMES – Joint Abbotsford Mission Environmental System

ODP – Ozone depletion potential

PE – Polyethylene

PER – Primary energy ratio

PFC – Perfluorocarbon

PVC – Polyvinyl chloride

SPF – Seasonal performance factor

SWHP – Surface water heat pump

TEWI – Total equivalent warming impact

WSTPB – Water supply treatment plant building

The list of abbreviations has been taken from [5] – "Ground Source Heat Pumps. A technology review", BSRIA, July 1999

INTRODUCTION

It’s known, that the biggest part of annual consumed fuel by humans is spent to produce heat with not very high temperature for domestic and industrial purposes. Fuel capacity on the Earth is very limited. According to calculations, the fuel consumption per day by humans is equivalent to fuel producing by the sun on the Earth per 1000 years. /2/ That’s why the problems of saving sources of natural fuel and finding new ways of economical and energy-effective heating systems become more and more important not only for countries with poor sources of fossil fuel but for all countries in the world.

In my Bachelor Thesis I want to make a design of the heat supply plant based on the heat pump technology. In my opinion, this technology is one of the most effective from the side of energy-efficiency and from the side of being environmentally friendly and fire safety. This project will be an example of a heating system for a water supply treatment plant building in Rahja settlement of Vsevologsk district, Leningrad region, Russian Federation.

There are several reasons to use this technology in Russian Federation for industrial premises and other buildings such as water supply and wastewater treatment plants. At first sight Russian Federation has a lot of different energy sources such as wood, coal, oil, natural gas, but all of these sources except of wood are nonrenewable. Fuel consumption, which is growing up from year to year, significantly exceeds its natural growth. Furthermore using the fossil fuel in Russian Federation is not only inefficient but also careless. Moreover the heat of the available fuel is not fully used. Significant part of this heat with insufficient temperature for conventional methods of using goes back to environment. That’s why in some decades; Russian Federation will face with the problem of lack of fuel resources. It will be one of the great problems for Russian people: nonrenewable fossil fuel sources will be empty or too deep to obtain the fuel; forests will be almost cut down; hydro- and atomic- energy plants won’t be able to supply enough energy and heat.

The conventional heating systems use very high quality fuel in boiler plants with the products of combustion, which are heated up to 1500˚C or, that’s much more careless, electricity to produce the energy in the form of low temperature heat (water with temperature lower than 100˚C and air with temperature lower than 50˚C). Only that part of the fuel is consumed thermodynamically efficiently which is burnt out at the heat and power plant. The high temperature heat of the products of combustion is used the most efficiently at these plants for the purposes of producing electrical energy. At the same time heat-transfer agent with temperature, which is near to the required temperature for heating needs, is used at the heat and power plant for the heat supply purposes.

However using the combined heat and power plants (CHP) may be inefficient in many cases. In the regions where consumers are divided into small groups, which are far from to each other, the main sources of heat supply are different types of boilers, boiler plants, stoves and electric heaters. The heat pump technology must be used to replace heating devices based on combustion and electric heaters.

At the same time, there are almost infinite sources in the surrounding nature, but the temperature of these sources is lower than we need. That’s why the problem of using low temperature heat has a big importance. One of the possible and hopeful ways to increase the temperature of heat-transfer agent is using a heat pump.

Increasing the temperature of low potential heat allows using new energy sources such as surrounding air, water energy, geothermal energy and others. All of these sources are renewable and almost infinite. The heat pump essentially expands the possibility of using the low potential energy thanks to consumption of a little part of energy, which is fully transformed into the work.

Unfortunately, this technology is very rarely used in Russian Federation. That’s why during my work I want to become familiar with the heating systems based on heat pump technology and become a good specialist in this area. I want to show my Russian colleagues that this technology is suitable and economically reasonable for heating purposes of water supply treatment plants, wastewater treatment plants and other industrial buildings because of it’s high heat energy efficiency and small electricity consumption.

For these purposes I’ll have to compare the benefits and limits of this method of heat producing with the conventional methods of heat producing such as small boiler plants, electric heaters, stoves, district heating systems from both technical and economical sides. Moreover I’ll need to make calculations of heat losses through the building envelope, the selection of suitable heat pump and defining its properties and other required actions. In the end of my work I’ll try to determine the area of using the heat pump technology, the cost of 1 MWh of heat energy and the coefficient of performance COP1 of heat pump.

1. HEAT PUMP THEORY

1.1 Basic concepts

The term "heat pump" can be broadly defined as a device where the heat-transfer agent with low potential increase it's potential to the required level by means of consumption of mechanical or other energy. More narrowly "heat pump" can be defined as piston compressor, rotation compressor or turbine compressor, which compresses steam or gas to increase its temperature.

In 1824 it was the first time when Carnot had used the thermodynamic cycle for description of the process, and this cycle, known as Carnot cycle, nowadays is staying a fundamental basic for comparison with it and evaluation of effectiveness of heat pumps.
The ideal Carnot heat pump cycle is presented on **figure 1**:

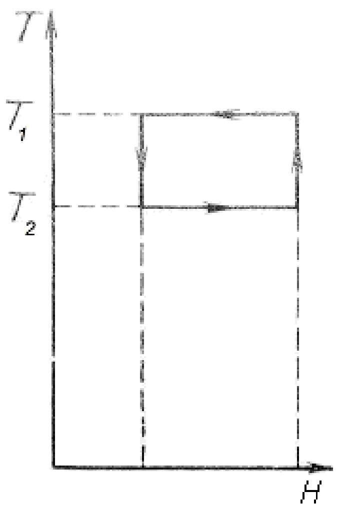

FIGURE 1. Carnot cycle. /1/

Abbreviations on **Figure 1** mean:
H – entropy;
T – temperature;

The heat pump can be presented as the backward (inverted) thermal engine (machine).
In thermal engine, by means of expanding of steam or gas, which has higher temperature than the environment, we get mechanical energy, and the temperature of steam or gas decrease very fast. In the heat pump we have an opposite situation. By means of consumption of mechanical energy the steam or gas is compressing and its temperature increase very much. That's why the heat pump is backward thermal engine. Entropy diagrams on **figure 2** and **figure 3** give us more visual presentation about nature of heat pump in comparison with

thermal engine. These figures show us working cycles of thermal engine and heat pump in the ideal machine, but with the real working substances.

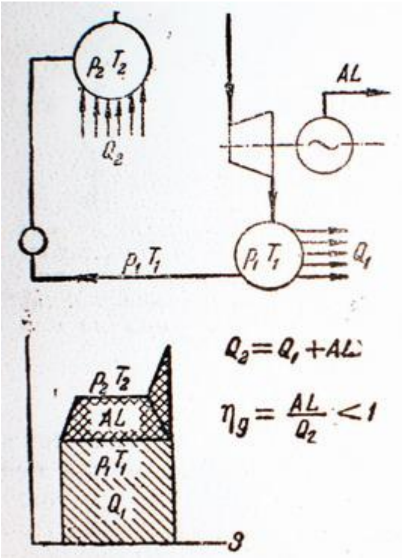

FIGURE 2. Steam thermal engine. /2/

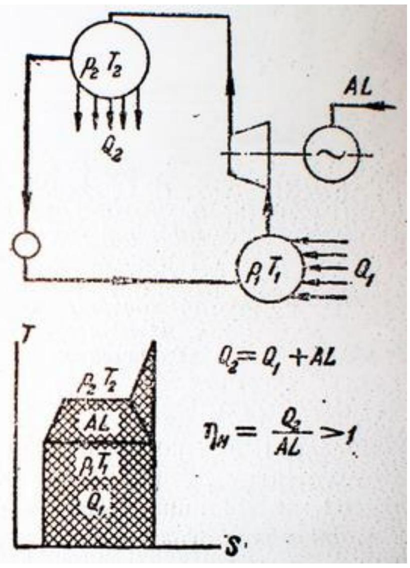

FIGURE 3. Steam heat pump. /2/,

Abbreviations on **Figure 2** and **Figure 3** mean:

S – entropy;

T_1, T_2 – temperature;

p_1, p_2 – pressure;

Q_1, Q_2 – heat;

AL - thermal equivalent of consumed mechanical energy.

On **figure 2** and **figure 3** the entropy diagrams of steam thermal engine and steam heat pump are presented. By means of supply (delivery) of low potential heat at the temperature $T_{1,}$ some working substance at the pressure p_1 evaporates in the evaporator and transforms from the liquid phase to the vapour. By means of compressors work the vapour is compressing under adiabatic conditions to the pressure p_2 and temperature T_2 that corresponds to this pressure. After this stage the vapour goes to the condenser where it loses superheating and starts to condensate. During the condensation process vapour gives back the heat with higher temperature into surrounding or to another substance. This produced condensate with the pressure p_2 goes to the throttling valve (expansion valve) where its pressure decrease to the pressure p_1 and its temperature becomes lower up to the temperature T_1. Under these conditions a small amount (small part) of condensate will evaporate, but the main part will go to the evaporator, where it will perceive the low potential heat and evaporate. The cycle will start again.

In this closed cycle, which is opposite to the power cycle of thermal engine, the heat Q_1 is perceived from the source of low potential energy, and the mechanical energy with thermal equivalent defined as area AL is consumed. The heat Q_2, which is defined as $Q_2 = Q_1 + AL$ gives back with higher temperature. It means that in heat pump in all cases the quantity (amount) of heat, which is given back, is higher than the thermal equivalent of mechanical energy, which was consumed.

In other words, the coefficient of efficiency of heat pump is:

$$\eta_{HP} = \frac{Q_2}{AL} > 1,$$ **Formula 1 /2/**

It is opposite to the coefficient of efficiency of thermal engine for which the useful output is AL and the heat Q_2 is consumed and less than 1:

$$\eta_{TE} = \frac{AL}{Q_2} < 1,$$ **Formula 2 /2/**

The ratio between the useful heat output Q_2 and the thermal equivalent of consumed mechanical energy AL is called the coefficient of performance COP_1. That's why

$$COP_1 = \frac{Q_2}{AL} > 1,$$ **Formula 3 /2/**

/2,p.4-5/

The Carnot cycle on **figure 1** shows us the working process of the ideal heat pump, which is working, in the given range (interval) of temperatures. Pointers show the direction of the process for heat pump. The heat is isothermically delivered at the temperature T_1 and isothermically withdrew at the temperature T_2. The compression and expansion are executed at the constant entropy, and the work is delivered from the external engine. Using the definition of the entropy and thermodynamic laws we can say that the coefficient of performance (COP_1) for Carnot cycle has the following view:

$$COP_1 = \frac{T_1}{(T_2 - T_1)} + 1 = \frac{T_2}{(T_2 - T_1)},$$ **Formula 4 /1/**

There are no heat pumps produced on the Earth have better characteristic than the characteristic of the Carnot cycle, and all practical cycles are trying to approach to its characteristic as close as it is possible. /1,p.16-17/

Usually value of COP_1 for heat pump is in range from 2,5 to 5.

1.2 Different types of heat pumps

"Heat pumps are used where geothermal water or ground temperatures are only slightly above normal, generally 10 to 35^{o}C. Conventional geothermal heating (and cooling) systems are not economically efficient at these temperatures. Heat pumps, at these temperatures, can provide space heating and cooling, and with a desuperheater, domestic hot water. Two basic heat pump systems are available, air-source and water- or ground-source”. /4/

Water- and ground-coupled heat pumps, referred to as geothermal heat pumps (GHP), have several advantages over air-source heat pumps:

1. They consume about 33% less annual energy;
2. They tap the earth or groundwater, a more stable energy source than air;
3. They do not require supplemental heat during extreme high or low outside temperatures;
4. They have a simpler design and consequently less maintenance. /4/

“The main disadvantage is the higher initial capital cost, being about 33% more expensive than air source units. This is due to the extra expense and effort to burying heat exchangers in the earth or providing a well for the energy source. However, once installed, the annual cost is less over the life of the system. The savings is due to the coefficient of performance (COP) averaging around 3 for GHP as compared to 2 for air-source heat pumps.” /4/

According to these advantages and disadvantages the decision to use the ground source heat pump for designing the heating system for water supply treatment plant building (WSTPB) was made.

1.3 Geothermal Heat Pump History

The basic principle of heat pump is described in the Carnot works and in the description of Carnot cycle, which has been published in his dissertation (thesis) in 1824. The practical heat pump system has been offered by Villiam Tomson (later – The lord Kelvin) in 1852. That system was called the “heat multiplier” and it shows how the cooling machine can be used for the heating purposes.   Look at the **figure 4**:

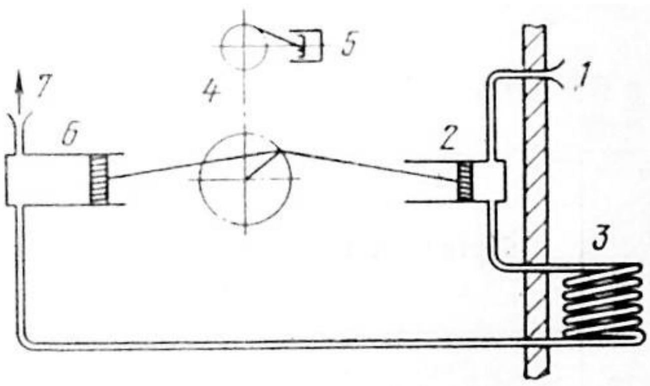

FIGURE 4. The scheme of Tomson’s “heat multiplier”

On the **Figure 4**: 1 – surrounding air; 2 – input cylinder; 3 – heat exchanger; 4 – drive; 5 – steam machine; 6 – output cylinder; 7 – heated room.

The main drawback of the “heat multiplier” was usage surrounding air as a working substance, what was the cause of lower effectiveness in comparison with ground source heat pump (see paragraph 2). /1,p.9/

“The earth offers a steady and incredibly large heat source, heat sink and heat storage medium for thermal energetic uses, like for the geothermal heat pump. A steady temperature in the underground first was scientifically proven in deep vaults beneath the Observatory in Paris. In 1838, very exact measurements of temperature in the ground started at the Royal Edinburgh Observatory in Scotland.” /6,p.5/ The diagram of measurement results is shown on **figure 5**.

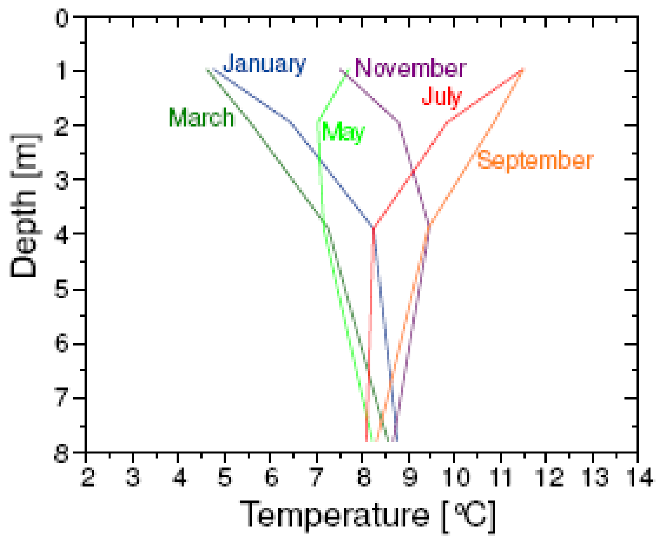

FIGURE 5. “Underground temperatures at the Royal Edinburgh Observatory, average 1838-1854 (after data from Everett, 1860)” /6,p.5/

The interesting fact is that this huge and almost infinite steady heat reservoir was used as a source of low potential heat energy for ground source heat pump only in the beginning of 20^{th} century, about 50 years later. “The first documented suggestion of using the ground as a heat source appears to be in 1912 in Switzerland in a patent filed by H. Zolly (Wirth, 1955) but at that time the efficiency of heat pumps was poor because of some technology problems caused by low technology level in that area.” /5,p.1/

“In the UK Sumner first used the ground as a source for a heat pump for space heating in a single-family house in the mid 1940s (Sumner, 1976). An underground horizontal collector laid at a depth of about 1m was used to supply heat via copper pipes buried in a concrete floor. A coefficient of performance (COP_1) of 2.8 was achieved.” /5,p.1/

“The first ground source heat pump in North America was installed in a house in Indianapolis in October 1945 (Crandall, 1946). This consisted of copper tubes buried at a depth of about 1.5 m in the ground with the refrigerant circulating directly through them.” /5,p.1/

On the **figure 6** can be seen the schematic illustration of first ground source heat pump in North America.

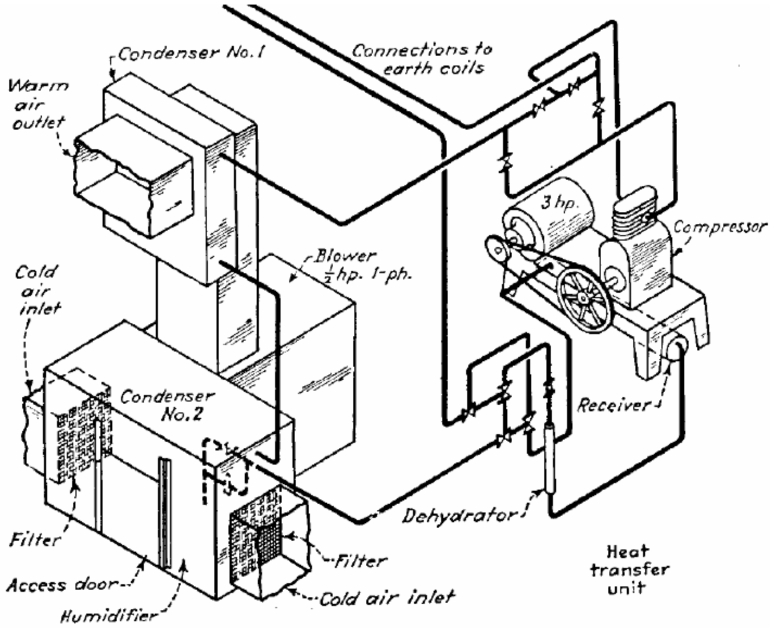

FIGURE 6. Schematic of first geothermal heat pump in North America, 1945 (from Crandall, 1946)

“Commercial use of the ground as a heat source/sink did not begin until after the first oil shock in 1973 but was well established by the end of the seventies by which time there were over 1000 ground source heat pumps installed in Sweden (Granryd, 1979).” /5,p.1/

“Today ground source heat pumps are an established technology with over 400,000 units installed worldwide (around 62% of which are in the US) and about 45,000 new units installed annually.” /5,p.2/

All in all using geothermal heat pump is effective way of producing heat energy because of the ground is a stable source of low potential heat energy during the whole year. That allows the heat pump to operate very close to its optimal design values.

1.4 Principle schemes of ground source of heat pumps collector systems

Ground source heat pumps, as it is already known, use the energy stored in the earth’s crust as a relatively stable temperature source of low potential heat energy during the whole year.

That's why it's very important to know more about different types of GSHP collector systems and factors which has essential impact on heat transfer from soil to ground collector.

1.4.1 Main factors affecting heat transfer to a ground collector

"The two main factors affecting heat transfer to a ground collector are its surface area (i.e. a factor of the pipe length and diameter) and the thermal properties of the ground, which will determine the length of heat exchanger needed to meet a given load." /5,p.4/

The **figure 7** shows us the seasonal variation of soil temperature through the year for two different depths 0,02m and 1m, which was measured in Belgium.

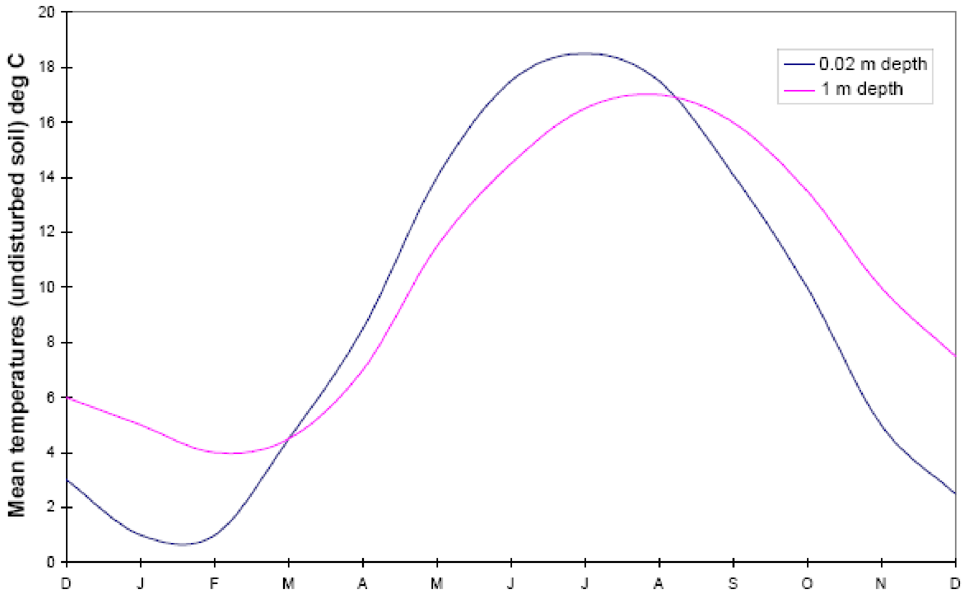

FIGURE 7. "Seasonal variations of soil temperatures at depths of 0,02m and 1,0m." /1,p.105/

Table 1 shows us the typical thermal properties of different types of soil for wet and dry conditions.

TABLE 1. "Typical thermal properties of soils." /5,p.5/

Material	Conductivity W/(m K)	Specific heat kJ/(kg K)	Density kg/m^3	Diffusivity m^2/day
Granite	2.1 to 4.5	0.84	2,640	0.078 to 0.18
Limestone	1.4 to 5.2	0.88	2,480	0.056 to 0.20
Marble	2.1 to 5.5	0.80	2,560	0.084 to 0.23
Sandstone				
Dry	1.4 to 5.2	0.71	2,240	0.074 to 0.28
Wet	2.1 to 5.2			0.110 to 0.28
Clay				
Damp	1.4 to 1.7	1.3 to 1.7		0.046 to 0.056
Wet	1.7 to 2.4	1.7 to 1.9	1,440 to 1,920	0.056 to 0.074
Sand				
Damp		1.3 to 1.7		0.037 to 0.046
Wet[1]	2.1 to 2.6	1.7 to 1.9	1,440 to 1,920	0.065 to 0.084

[1] Water movement will substantially improve thermal properties

“The moisture content of the soil has a significant effect on its thermal properties. When water replaces the air between particles it reduces the contact resistance. The thermal conductivity can vary from 0.25 W/mK for dry soil to 2.5 W/mK for wet soil. The thermal conductivity is relatively constant above a specific moisture threshold called the critical moisture content (CMC), but below the CMC conductivity drops rapidly.” /5,p.5/

1.4.2 Different systems of horizontal and vertical ground collectors

The ground collector for geothermal heat pump can be installed both vertically and horizontally. The choice of a vertical or horizontal direction of ground collector installation depends on the available space, local soil type and excavation costs. After describing main types of ground collector systems and studying more about their principal schemes, advantages and disadvantages, the principal scheme of GSHP ground collector for heating system of water supply treatment plant building based on heat pump technology will be chosen.

There are three main types of ground collector systems:

1. Open system
2. Closed horizontal system
3. Closed vertical system

According to the geological and hydro geological conditions on the water supply treatment plant site and the leakage of free space on the site there are two the most reasonable systems of ground collector:

1. Closed horizontal system with trench collector, which is presented on **figure 8**;

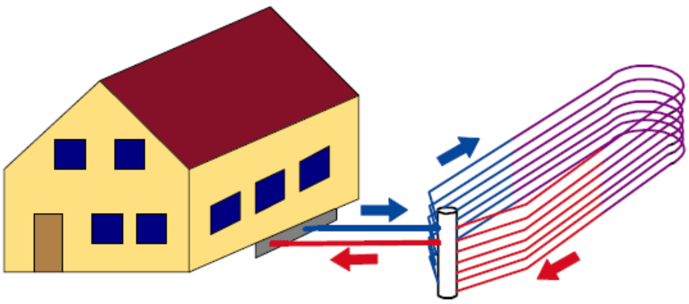

FIGURE 8. Trench collector

“Exploiting a smaller area at the same volume, these collectors are best suited for heat pump systems for heating and cooling, where natural temperature recharge of the ground is not vital. For the trench collector, a number of pipes with small diameter are attached to the steeply inclined walls of a trench some meters deep”. /6,p.9/

2. Closed vertical system with energy piles, which is presented on **figure 9**.

"A special case of vertical closed systems is „energy piles", i.e. foundation piles equipped with heat exchanger pipes. All kind of piles can be used (pre-fabricated or cuts on site), and diameters may vary from 40cm to over 1 m." /6,p.10/

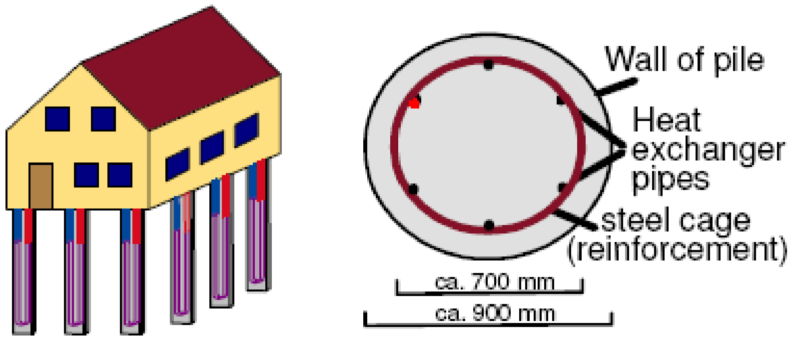

FIGURE 9. "Energy piles, cross-section of a pile with 3 loops." /6,p.10/

1.5 Refrigerants

This paragraph is about the most common refrigerants, which are used in heat pump technology and short overview about their characteristics.

"The dominant refrigerants today are fluorocarbons. However, as a result of the Montreal Protocol of 1987 on Substances that Deplete the Ozone Layer, the fully halogenate chlorofluorocarbons (CFCs) like R12 are being phased out because of their high ozone depletion potential (ODP). They also have high global warming potential (GWP)." /5,p.19/ According to the demands, which were listed in the Montreal Protocol, the research work was concentrated on investigating chlorine-free refrigerants and reducing their toxicity and inflammability. Environmental effects of refrigerants look at the **table 2**:

TABLE 2. "Environmental effects of refrigerants." /5,p.21/

Refrigerant	ODP	GWP [1]
CFC eg R12	0.25	1.0
HCFC eg R22	0.02 -0.05	1,000 - 2,800
HFC eg R134a, R407C, R410B	0	most significant (up to 3,800)
Ammonia NH_3 (R717)	0	0
Propane C_3H_3 (R290)	0	0

1. GWP is measured on the basis of a 100 year time horizon as defined in BS4434:1995.

"As a result of this problem, work in Europe on replacement refrigerants has concentrated on "natural" refrigerants, which have no ODP and no GWP instead of HFCs (ie hydrocarbons or perfluorocarbons (PFCs), ammonia)." /5,p.21/

1.6 Using warm water from the heat pump for underfloor heating and radiators: benefits and drawbacks.

"Vapour compression cycle heat pumps are well suited to low temperature heating systems (eg. underfloor heating) and systems with modern radiators, but they have poor COPs when used with conventional hydraulic heating systems with circulation temperatures of 70°C or higher. **Table 3** shows how the COP_1 of a water-to-water heat pump varies with the distribution/return temperature." /5,p.16/

TABLE 3. "Typical COPs for a water-to-water heat pump operating with various heat distribution systems." /5,p.16/

Heat distribution system (supply/return temperature)	COP [1]
Floor heating (30°C/35°C)	4.0
Modern radiators (35°C/45°C)	3.5
Conventional radiators (50°C/60°C)	2.5

1. Heat source 5°C, Carnot efficiency 50%.

To achieve COP_1 from 3,5 to 4,0 or more the underfloor heating and modern radiators as distribution systems for heating system for water supply treatment plant building will be used.

2. INITIAL DATA

2.1 Geographic situation

The site of water supply treatment plant is located in Rahja village of Vsevologsk district, Leningrad region, Russian Federation. The location is presented on **picture 1**.

PICTURE 1. Location of water supply treatment plant building.

2.2 Geological situation

The site of water supply treatment plant is characterized by smooth surface with elevations 30.90-31.15m. The platform is covered by well-decayed compacted peat from a surface to depth of 0,4-0,9m. Elevations of the peat base surface are 30,50-30,20 m. It gives the reasons to think that weak grounds were not fully replaced earlier. Below the layer of peat there is layer of fine brown-yellow water-saturated sands with the depth 0,4-1,9m and medium and coarse sands to the depth 2,4-2,8 м. Below sands layer there is basic layer of sandy loams and loams.

The depth of the water table is 0,7-1,0m from the day surface. The ground water concerning to the concrete is characterized as aggressive.

2.3 Weather conditions

The area climate is transitive from typically marine to moderate-continental, with cool summer and relatively soft winter.

Initial data has been taken from BUILDING CLIMATOLOGY. Building norms and rules of Russian Federation. /15/ Look at the **table 4**:

TABLE 4. Average monthly and annual air temperature, ^{o}C /15/

I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	year
-7.8	-7.8	-3.9	3.1	9.8	15.0	17.8	16.0	10.9	4.9	-0.3	-5.0	4.4

2.4 Initial data for heating system calculations

Heat-proofing characteristics of building envelopes allow reducing the expenses of energy resources. The building of water supply treatment plant will be heated by means of underfloor heating and radiators.

Calculation parameters of outdoor air:

For designing of the heating system - 26 °C

For designing of the ventilation system (winter) - 26 °C

For designing of the ventilation system (summer) +22 °C

Average temperature of the heating period - 1,8 °C

Duration of the heating period 220 days

Indoor air temperatures:

For office and accommodation rooms 18 °C

Reagent's room, technical room, tambour 16 °C

Pumping station of second lift, filtration hall 7 °C

Parameters of heat-transfer agent (water) for underfloor heating:

$T_1 = 35$ °C

$T_2 = 30$ °C

Ventilation

Supply and exhaust ventilation is designed for water supply treatment plant building. Mechanical supply and exhaust ventilation in pumping station of second lift is calculated for assimilation of heat gains from technical equipment. In the room of coagulant and flocculent the mechanical supply and exhaust ventilation is designed with the value of air exchange rate n=3 1/h. In the room of natrium hypochlorite the mechanical supply and exhaust ventilation is designed with the value of air exchange rate n=6 1/h. For other rooms the value of air exchange rate is accepted according to The National Building Code of Finland D2 "Indoor Climate and Ventilation of Buildings Regulations and Guidelines" 2003.

2.5 Existent technical conditions of water supply treatment plant site

Water supply of the occupied places by high-quality water has the important social and sanitary-and-hygienic value because it protects people from epidemic diseases and creates good conditions for life and economic activities. Moreover it is one of the major economic problems of the modern community for many countries. The Project of reconstruction of water supply treatment plant of Rahja village of Vsevologsk district, Leningrad region is very important for people who live not only in Rahja, but also for such nearest villages as Proba and Irinovka. In future it is planned to supply these villages with pure water from the water supply treatment plant of Rahja.

As you know, before making decisions designer need to know the location of the site in the village, geological and weather conditions in the sites region, the quality of water which is delivered from the water source, drinking water quality standards and many other additional conditions. All of these conditions are presented in previous paragraphs, but one more essential thing for each designer is the existing technical conditions of water supply treatment plant site. It's important for the designer because he needs to know which of existing constructions might be renovated and used as a part of new water supply treatment plant and which constructions should be destroyed because of their bad operating conditions.

After the technical inspection, which has been done by specialists from the Project and Research Bureau “Engineering Ecosystems”, the report about technical conditions of the water supply water treatment plant was made. There is some information from that report to make a short overview about the existing situation on the site.

Source of drinking water supply of the Rahja village is water of Ladoga lake which is delivered to water supply treatment plant by Ladoga conduit. Necessity of working out of the project of reconstruction and expansion water supply treatment plant of Rahja settlement is caused by the increased requirement of settlement for potable water (in the long term to 1600 m^3/day), and impossibility of maintenance of this requirement by existing water supply treatment plant. Because of the Ladoga water, especially in flood periods, contains a significant amount of a plankton, suspended solids and other pollutions, the sanitary-and-epidemiological service of Vsevolozhsk district repeatedly fixed cases of discrepancy of quality of potable water delivered to settlement to standard requirements, both on physical and chemical, and on bacteriological indicators. On existing water supply treatment plant disinfecting of water by chloric water before giving to the consumer is carried out only. Existing productivity of operating water treatment facilities is only 700-800 m^3/day.

The first view of site of existing water supply treatment plant and its building was extremely terrible. In one word, there was almost nothing from standard requirements for water supply treatment plant: no fence, no guards, no roads and so on (look at the **picture 2**)

PICTURE 2. The technical conditions of constructions on the site.

The metal land tank for reception of the Ladoga water has volume ~ 100 м^3. The "Luch-4" is installed on the tank for disinfecting of raw water. The second similar tank located on a site, is not completed and not placed in operation (look at the **picture 3**).

PICTURE 3. The technical conditions of raw water tanks on the site.

The main building of water supply treatment plant is made of a red brick with facing from a facade a silicate brick in the general thickness of walls 2,5 bricks. A constructive condition of a building is seems to be satisfactory. The internal stuffing of a building has become unfit for use: timber floors have rotted through, windows and doors require the replacement, plaster on 80 % of a surface has fallen off. The premises intended for laboratory, electro panel board, household premises are destroyed and cannot be maintained without building major repairs as it seen on the **picture 4**.

PICTURE 4. The technical conditions of the rooms of WSTPB.

According to the initial data and results of our inspection, was made the decision that the existing water supply treatment plant is not be able to be renovated because of economical and technical reasons. That’s why the design of a new compact water supply treatment plant with much more effective treatment technology must be done. New treatment plant should be installed on the site without destruction the existing building of water supply treatment plant. The destruction of the old building is too expensive for this region and the site of water supply treatment plant has very small size ~ 50x60m. It will be quite difficult task, but the new compact membrane filtration technology installed in fast-raised building will help us to solve this problem. Moreover it’s good chance to try to install innovative heating system based on heat pump technology into the new building of water supply treatment plant equipped by the newest water treatment technology.

3. HEAT LOSS CALCULATIONS AND DESIGNING THE HEATING SYSTEM FOR WSTPB

3.1 Calculations of U-values for elements of building envelope

"When the outdoor temperature, t_o (^oC), is lower than the room temperature, t_r (^oC), the building will lose heat to its surroundings. The cause of the loss is heat transmission and air infiltration through the outside walls, the windows and doors and the roof. The heat flow due to transmission through the different parts depends on their area, A (m^2), and their heat transmission coefficient, U W/(m^2·ºC)." /7,p.321/

In order to calculate heating demands for the water supply building plant, first of all U-values for elements of building envelope: walls, floor, roof should be calculated. After that U-values for windows and doors will be chosen from tables presented in C4 National Building Code of Finland "Thermal insulation. Guidelines 2003" and Building Regulations Approved Document L1 "Conservation of fuels and power in dwellings 2002". Choosing acceptable U-values is very important, because that has an essential impact on amount of heat losses, which will go through the elements of building envelope to the its surrounding. The less U-value of elements of building envelope, the less heat losses through them.

Besides the heat losses through the elements of building envelope the heat losses with leakage air flow and heat losses with ventilation air flow will be taken into account. There will no special chapter with calculations and designing of the ventilation system in Bachelor Thesis because there is no air heating system in the WSTPB. But the differences between designing temperatures of the building premises (e.g. the designing temperature for laboratory is +18ºC and for the filtering hall is only +7ºC) get us a possibility to calculate the addition heat gains from the ventilation system of more warm premises. The needed parameters of ventilation system such as air exchange rate and designed temperature will be presented as an initial data. For the topic, which is discussed in this Bachelor Thesis the ventilation system is already done and it operates separately from the heating system based on heat pump technology.

3.1.1 Calculations of U-value for walls of the WSTPB.

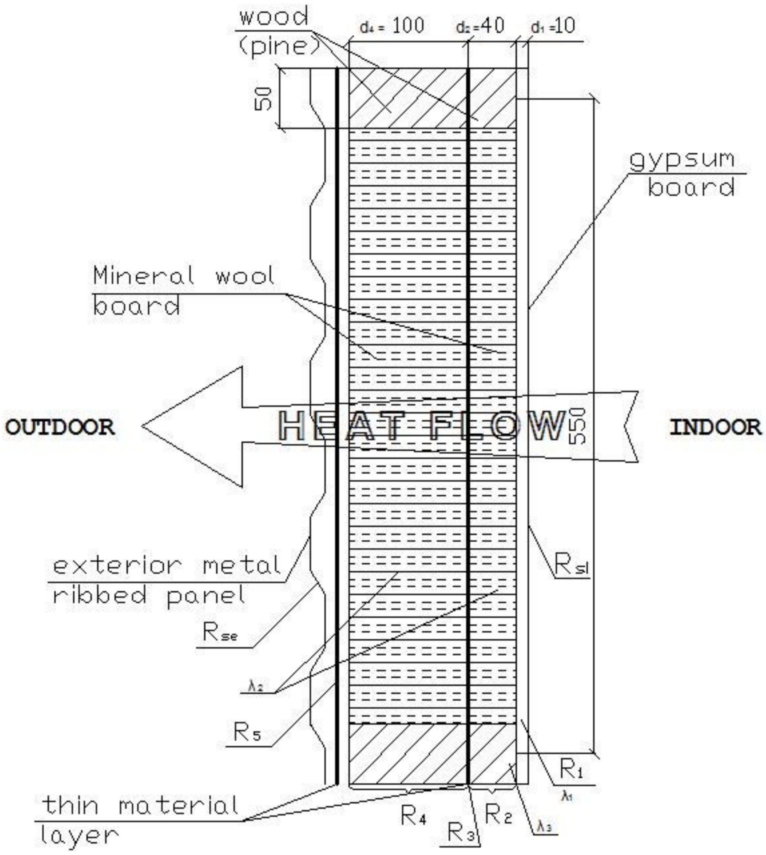

FIGURE 10. Components of wall with mineral wool board insulation.

The components of wall with wool board as an insulation material are presented on **figure 10.** The initial data of thermal conductivities, λ W/(m·ºC) and thermal resistances, R (m^{2}·ºC)/W for different materials of walls components were taken from the C4 National Building Code of Finland "Thermal insulation. Guidelines 2003" and have the following values /10,p.12-20/:

λ_1 = 0,20 W/(m·ºC) – gypsum board with thickness d_1=0,01m

λ_2 = 0,045 W/(m·ºC) – mineral wool board with thickness d_2=0,04m and d_4=0,1m

λ_3 = 0,10 W/(m·ºC) – pine timber with thickness d_2=0,04m and d_4=0,1m

R_3 = R_5 = 0,04 (m^{2}·ºC)/W – thin material layer

R_{si} = 0,13 (m^{2}·ºC)/W – internal surface resistance in horizontal direction

R_{se} = 0,04 (m^{2}·ºC)/W – external surface resistance in horizontal direction

The value of distance between pine timbers is defined as K=550mm

The U-value is calculated with using the **formula 5**:

$$U = \frac{1}{R_T},$$ **Formula 5 /10/**

Where:

U = U-value of building component (wall), W/($m^{2}\cdot$ºC)

R_T = total thermal resistance of a building component from one environment to another, ($m^{2}\cdot$ºC)/W

The total thermal resistance of a building component (wall) is defined as sum of thermal resistances of all material layers, and internal and external surface resistances, and will be calculated using **formula 6**:

$$R_T = R_{si} + R_1 + R_2 + ... + R_m + R_g + R_b + R_{q1} + R_{q2} + ... + R_{qn} + Rse\ ,$$ **Formula 6 /12/**

Where:

$R_1 = d_1/\lambda_1$, $R_2 = d_2/\lambda_2$ … $R_m = d_m/\lambda_m$

d_1, d_2, … d_m = thickness of material layer 1, 2, …m, m

λ_1, λ_2, … λ_m = design thermal conductivity of material layer 1, 2, …m, W/(m·ºC)

R_g = thermal resistance of air cavity in the building component, ($m^{2}\cdot$ºC)/W

R_b = thermal resistance of the ground, ($m^{2}\cdot$ºC)/W

R_{q1}, R_{q2}, … R_{qn} = thermal resistance of thin material layer 1, 2, …n, ($m^{2}\cdot$ºC)/W

R_{si} + R_{se} = sum of the internal and external surface resistances, ($m^{2}\cdot$ºC)/W

For the certain wall, which is used in the Bachelor Thesis, as a building component the **formula 6** will have a following view:

$R_T = R_{si} + R_1 + R_2 + R_3 + R_4 + R_5 + R_{se}$,

Where:

R_{si} = 0,13 ($m^{2}\cdot$ºC)/W

$R_1 = d_1/\lambda_1$,

$R_2 = \frac{f_1 \cdot d_2}{\lambda_2} + \frac{f_2 \cdot d_2}{\lambda_3}$, where f_1 and f_2 = proportion part of the total area of a material layer of homogeneous sub-area 1 and 2 in the inhomogeneous layer 2.

$f_2 = \frac{0,05}{K} = \frac{0,05}{0,55} = 0,091\ (9,1\%)$

$f_1 = 1 - 0,091 = 0,909\ (90,9\%)$

$R_3 = R_5$ = 0,04 ($m^{2}\cdot$ºC)/W

$R_4 = \frac{f_1 \cdot d_4}{\lambda_2} + \frac{f_2 \cdot d_4}{\lambda_3}$, where f_1 and f_2 = proportion part of the total area of a material layer of homogeneous sub-area 1 and 2 in the inhomogeneous layer 4.

$$f_2 = \frac{0,05}{K} = \frac{0,05}{0,55} = 0,091 \ (9,1\%)$$

$$f_1 = 1 - 0,091 = 0,909 \ (90,9\%)$$

$R_{se} = 0,04$ (m^2·ºC)/W

$$R_T = 0,13 + \frac{0,01}{0,20} + \frac{0,909 \cdot 0,04}{0,045} + \frac{0,091 \cdot 0,04}{0,10} + 0,04 + \frac{0,909 \cdot 0,10}{0,045} + \frac{0,091 \cdot 0,10}{0,10} + 0,04 + 0,04 = 3,26 \ (m^2 \cdot °C)/W$$

Than the U-value for a wall will be found using **formula 5**:

$$U = \frac{1}{R_T} = \frac{1}{3,26} = 0,307 \ W/(m^2 \cdot °C)$$

According to the C3 National Building Code of Finland "Thermal insulation in a building. Regulations 2007", "when a heated or especially warm space abuts to the outdoor air, to an unheated space or to the ground, the thermal transmittance U for building components must not exceed the following values" /9,p.5/:

Wall	0,24 W/(m^2·ºC)
Roof, base floor abutting to outside	0,15 W/(m^2·ºC)
Building component against the ground	0,24 W/(m^2·ºC)

The calculated U-value for wall is $U_{wall} = 0,307 \ W/(m^2 \cdot °C) > U_{wall}^{C3} = 0,24 \ W/(m^2 \cdot °C)$, thus this value is not acceptable. It means that the more effective insulating material must be chosen. The bigger thickness for insulating material layer can't be chosen because of sharp total dimension of the wall. That's why another type of insulating material should be chosen.

The expanded polystyrene as insulating material has λ_2 = 0,033 W/(m·ºC), than the thermal resistance will has the following value:

$$R_T = 0,13 + \frac{0,01}{0,20} + \frac{0,909 \cdot 0,04}{0,033} + \frac{0,091 \cdot 0,04}{0,10} + 0,04 + \frac{0,909 \cdot 0,10}{0,033} + \frac{0,091 \cdot 0,10}{0,10} + 0,04 + 0,04 = 4,28 \ (m^2 \cdot °C)/W$$

Than the U-value for a wall will be found using **formula 5**:

$$U = \frac{1}{R_T} = \frac{1}{4,28} = 0,234 \ W/(m^2 \cdot °C)$$

The calculated U-value for wall is $U_{wall} = 0,234 \ W/(m^2 \cdot °C) < U_{wall}^{C3} = 0,24 \ W/(m^2 \cdot °C)$, thus this value is acceptable. It means that the expanded polystyrene is acceptable insulating material for the wall.

3.1.2 Calculations of U-value for roof and floor of the WSTPB

The calculations of U-value for roof and floor were made in the same way as for walls:

The calculated U-value for roof is $U_{roof} = 0{,}15\ W/(m^2 \cdot °C) = U_{roof}^{C3} = 0{,}15\ W/(m^2 \cdot °C)$

The calculated U-value for floor is $U_{floor} = 0{,}22\ W/(m^2 \cdot °C) < U_{floor}^{C3} = 0{,}24\ W/(m^2 \cdot °C)$, thus these values is acceptable.

3.1.3 Choosing the U-value for windows and doors

The U-value for windows is chosen using the tables of U-values in the Building Regulations Approved Document L1 "Conservation of fuels and power in dwellings 2002" /11,p.23/.

The U-value for windows with triple glazing (low-E ε_n=0.05, argon filled) is $U_{window}^{L1} = 1{,}3\ W/(m^2 \cdot °C)$

3.2 Heat losses through the elements of building envelope

When the U-values for all elements of building envelope were calculated the heat losses through these elements might be calculated.

The total heat losses due to transmission can be calculated using **formula 7**:

$$\Phi_T = \sum U_i \cdot A_i \cdot (t_r - t_o)\,, \qquad \textbf{Formula 7 /7/}$$

Where:

Φ_T = total heat loss due to transmission, W

U_i = U-value of i-element of building envelope, W/(m^2·ºC)

A_i = area of i-element of building envelope, m^2

t_r = temperature of the room (indoor temperature), ºC

t_o = outdoor temperature, ºC

The heat loss due to air infiltration/leakage depends mainly on the tightness of the different elements, and how well they are fitted together. If the aif flow due to infiltration is q (m^3/s), the heat loss due to infiltration will be calculated using **formula 8**:

$$\Phi_{laf} = q \cdot \rho \cdot C_p \cdot (t_r - t_o) \qquad \textbf{Formula 8 /7/}$$

Where:

Φ_{laf} = heat loss due to leakage air flow, W

q = leakage air flow, l/s

ρ = density of air, kg/m^3

C_p = specific heat capacity of air, kJ/(kg·ºC)

t_r = t_{in} temperature of the room (indoor temperature), ºC

t_o = outdoor temperature, ºC

For the purposes of heat loss calculations the following initial data should be presented:
Average outdoor temperature of the heating period t_{av} = - 1,8 °C
Duration of the heating period Z_{hp} = 220 days
Designed outdoor temperature t_o = -26 ºC
Indoor temperature t_r = +7…+18 ºC (depends on the certain room temperature demands)
Density of air ρ = 1,213…1,261 kg/m^3 (depends on the temperature)
Specific heat capacity of air C_p = 1,0 kJ/(kg·ºC)
Air exchange rate n = 0,5…6,0 1/h (depends on the demand for special room)
Air flow rates for ventilation purposes (except special rooms) are chosen according to /8,p.32 Table 1/.
Coefficient of leakage n_{leak} = 0,16 1/h

There is an example of heat losses calculations for the Room №1. It is technical room, which has dimensions 3,0x3,5m and height 3,5m and designing temperature +16ºC.

$$\Phi_T = \sum U_i \cdot A_i \cdot (t_r - t_o)$$

$$\Phi^1_{wall} = U_{wall} \cdot A_{wall} \cdot (t_r - t_o) = 0,234 \frac{W}{m^2 \cdot °C} \cdot (3,0 \cdot 3,5 + 3,5 \cdot 3,5)m^2 \cdot (16 - (-26))°C = 223,6W$$

$$\Phi^1_{roof} = U_{roof} \cdot A_{roof} \cdot (t_r - t_o) = 0,15 \frac{W}{m^2 \cdot °C} \cdot (3,0 \cdot 3,5)m^2 \cdot (16 - (+7))°C = 14,2W$$

$$\Phi^1_{floor} = U_{floor} \cdot A_{floor} \cdot (t_r - t_o) = 0,22 \frac{W}{m^2 \cdot °C} \cdot (3,0 \cdot 3,5)m^2 \cdot (16 - (+7))°C = 20,8W$$

$$\Phi^1_{wind} = U_{wind} \cdot A_{wind} \cdot (t_r - t_o) = (1,3 - 0,234) \frac{W}{m^2 \cdot °C} \cdot (1,44 \cdot 3)m^2 \cdot (16 - (-26))°C = 193,4W$$

$$\Phi^1_{laf} = q_{leak} \cdot \rho \cdot C_p \cdot (t_r - t_o) = n_{leak} \cdot V_{room} \cdot \rho \cdot C_p \cdot (t_r - t_o) =$$
$$= 0,16 \cdot (3,0 \cdot 3,5 \cdot 3,5)m^3 \cdot 1,213 \frac{kg}{m^3} \cdot 1,0 \frac{kJ}{(kg \cdot °C)} \cdot (16 - (-26))°C / 3,6 = 83,2W$$

$$\Phi^1_{vent} = q_{vent} \cdot \rho \cdot C_p \cdot (t_r - t_{sup}) = 6,0 l/s \cdot 1,213 \frac{kg}{m^3} \cdot 1,0 \frac{kJ}{(kg \cdot °C)} \cdot (16 - (+17))°C = -7,3W$$

$$\Phi^1_T = \Phi^1_{wall} + \Phi^1_{roof} + \Phi^1_{floor} + \Phi^1_{wind} + \Phi^1_{laf} + \Phi^1_{vent} = 223,6W + 14,2W + 20,8W + 193,4W + 83,2W - 7,3W = 528W$$

The other heat losses calculations are presented in **table 5**:

TABLE 5. Results of calculations of heat losses through the building envelope with taking into account heat losses with leakage air and ventilation.

Room and temp.	barriering designs: Name	Orientation	Size, m	Area, m^2	t_{int}-t_{ext}, °C	n	U, W/(m^{2}*°C)	Φ, W	n, 1/h	V, m^3	q=n*V, L/s	Φlaf, W	q vent, L/s	t supply air °C	Φvent, W	∑Φ, W
Room №1 Technical room temp °C 16	W	-	3,0x3,5	10,5	42	1	0,234	103,2	0,16	36,8	1,6	83,2	6	17	-7	528
	W	-	3,5x3,5	12,3	42	1	0,234	120,4								
	window	-	1,2x1,2	4,3	42	1	1,3	193,4								
	roof	-	-	10,5	9	1	0,15	14,2								
	floor	-	-	10,5	9	1	0,22	20,8								
	∑Φ, W	-	-	-	-	-	-	452,0								
Room №2 WC temp °C 18	W	-	2,0x3,5	7,0	44	1	0,234	72,1	0,16	24,5	1,1	58,1	-	-	-	87
	window	-	1,2x1,2	1,4	44	1	1,3	67,5								
	roof	-	-	7,0	11	1	0,15	11,6								
	floor	-	-	7,0	11	1	0,22	16,9								
	∑Φ, W	-	-	-	-	-	-	28,5								
Room №3 Checkroom temp °C 18	W	-	2,0x3,5	7,0	44	1	0,234	72,1	0,16	24,5	1,1	58,1	12	17	15	173
	roof	-	-	7,0	11	1	0,15	11,6								
	floor	-	-	7,0	11	1	0,22	16,9								
	∑Φ, W	-	-	-	-	-	-	100,6								
Room №4 Chief room temp °C 18	W	-	3,5x2,4	8,4	44	1	0,234	86,5	0,16	29,4	1,3	69,7	6	17	7	265
	window	-	1,2x1,2	1,4	44	1	1,3	67,5								
	roof	-	-	8,4	11	1	0,15	13,9								
	floor	-	-	8,4	11	1	0,22	20,3								
	∑Φ, W	-	-	-	-	-	-	188,2								
Room №5 Electrolysis room temp °C 16	W	-	6,0x3,5	21,0	42	1	0,234	206,4	0,16	73,5	3,3	167,6	122,5	17	-149	553
	window	-	1,2x1,2	5,8	42	1	1,3	257,9								
	roof	-	-	21,0	9	1	0,15	28,4								
	floor	-	-	21,0	9	1	0,22	41,6								
	∑Φ, W	-	-	-	-	-	-	534,2								

Room №6 Storage room for reagents	W	-	3,0x3,5	10,5	42	1	0,234	103,2	0,16	36,8	1,6	83,8	30,625	17	-37	249
	window	-	1,2x1,2	1,4	42	1	1,3	64,5								
	roof	-	-	10,5	9	1	0,15	14,2								
temp °C	floor	-	-	10,5	9	1	0,22	20,8								
16	∑Φ, W	-	-	-	-	-	-	202,6								
Room №7 Dispatching office	W	-	4,6x3,5	16,1	44	1	0,234	165,8	0,16	47,5	2,1	113	6	17	7	650
	W	-	2,95x3,5	10,3	44	1	0,234	106,3								
	window	-	1,2x1,2	4,3	44	1	1,3	202,6								
temp °C	roof	-	-	13,6	11	1	0,15	22,4								
18	floor	-	-	13,6	11	1	0,22	32,8								
	∑Φ, W	-	-	-	-	-	-	529,9								
Room №8 Tambour 1	W	-	4,2x3,5	14,7	42	1	0,234	144,5	0,16	24,3	1,1	55	-	-	-	422
	W	-	1,65x3,5	5,8	42	1	0,234	56,8								
	W	-	1,65x3,5	5,8	42	1	0,234	56,8								
	door	-	1,04x2,1	2,2	42	1	0,8	51,9								
temp °C	roof	-	-	6,9	42	1	0,15	43,7								
16	floor	-	-	6,9	9	1	0,22	13,7								
	∑Φ, W	-	-	-	-	-	-	367,3								
Room №9 Duty personnel room	W	-	2,6x3,5	9,1	44	1	0,234	93,7	0,16	41,9	1,9	99,3	18	17	22	331
	window	-	1,2x1,2	1,4	44	1	1,3	67,5								
	roof	-	-	12,0	11	1	0,15	19,7								
temp °C	floor	-	-	12,0	11	1	0,22	28,9								
18	∑Φ, W	-	-	-	-	-	-	209,9								
Room №10 Tambour 2	W	-	1,86x3,5	6,5	44	1	0,234	67,0	0,16	12,4	0,5	29,5	-	-	-	164
	door	-	1,04x2,1	2,2	44	1	0,8	54,4								
	roof	-	-	3,5	11	1	0,15	5,8								
temp °C	floor	-	-	3,5	9	1	0,22	7,0								
16	∑Φ, W	-	-	-	-	-	-	134,2								

Room №11 Compressed air plant	W	-	3,17x3,5	11,1	0	1	0,234	0,0	0,16	53,8	2,4	0,0	-	-	-	51
	window	-	1,2x1,2	1,4	33	1	1,3	50,7								
	roof	-	-	15,4	0	1	0,15	0,0								
temp °C	floor	-	-	15,4	0	1	0,22	0,0								
7	∑Φ, W	-	-	-	-	-	-	50,7								
Room №12 Chemistry laboratory	W	-	5,25x3,5	18,4	44	1	0,234	189,2	0,16	97,4	4,3	231,0	27,1	17	33	769
	window	-	1,2x1,2	4,3	44	1	1,3	202,6								
	roof	-	-	27,8	11	1	0,15	45,9								
temp °C	floor	-	-	27,8	11	1	0,22	67,3								
18	∑Φ, W	-	-	-	-	-	-	505,1								
Room №13 Storage room	W	-	3,0x3,5	10,5	42	1	0,234	103,2	0,16	42,0	1,9	95,8	35,0	17	-43	261
	window	-	1,2x1,2	1,4	42	1	1,3	64,5								
	roof	-	-	12,0	9	1	0,15	16,2								
temp °C	floor	-	-	12,0	9	1	0,22	23,8								
16	∑Φ, W	-	-	-	-	-	-	207,6								
Room №14 Main WSTPB Hall	W	-	18,0x11,8	190,8	33	1	0,234	1473,4	0,16	6018,3	267,5	11128	267,5	17	-3256	15377
	W	-	28,2x11,8	5,8	33	1	0,234	44,6								
	W	-	18,0x11,8	194,0	33	1	0,234	1498,3								
	window	-	1,2x1,2	33,1	33	1	1,3	1165								
	door	-	3,2x6,0	38,4	33	1	0,8	717,2								
temp °C	roof	-	-	526,4	33	1	0,15	2605,7								
	floor	-	-	467,3	0	1	0,22	0,0								
7	∑Φ, W	-	-	-	-	-	-	7504,3								
Coridor and hall	roof	-	-	59,3	11	1	0,15	98	0,16	207,6	9,2	123,1	-	-	-	364
temp °C	floor	-	-	59,3	11	1	0,22	144								
18	∑Φ, W	-	-	-	-	-	-	241,4								
Area above rooms №1-10, cor and hall temp °C7	roof	-	-	163,3	33	1	0,15	808,3	0,16	483,4	21,5	893,8	-	-	-	1702
	∑Φ, W	-	-	-	-	-	-	808,3								
												Total heat losses, W			∑Φ =	21945

According to the results of heat loss calculations presented in **table 5** the value of total heat losses through the building envelope are:

$$\Phi_T = 21945\ W \approx 22\ kW$$

The value of degree days will be calculated using **formula 9**:

$D_d = (t_{in} - t_{av}) \cdot Z_{hp}$, **Formula 9 /15/**

Where:

D_d = degree days, ºC·day

t_{in} = indoor temperature, ºC

t_{av} = average outdoor temperature of the heating period, ºC

Z_{hp} = duration of the heating period, day

$D_d = (18\ ºC - (-1{,}8\ ºC) \cdot 220\ days = 4356\ ºC \cdot day$

The value of average heat loss through the building envelope will be calculated using **formula 10**:

$H_{av} = \Phi_T / (t_{in} - t_{out})$, **Formula 10 /15/**

Where:

H_{av} = average heat loss through the building envelope, W/ ºC

Φ_T = total heat losses through the building envelope, W

$H_{av} = 21945W / (18\ ºC - (-26\ ºC)) = 498{,}8\ W/\ ºC$

The value of annual energy losses will be calculated using **formula 11**:

$Q_{an} = H_{av} \cdot D_d$, **Formula 11 /15/**

Where:

Q_{an} = annual energy losses through the building envelope, kWh

$Q_{an} = 498{,}8\ W/\ ºC \cdot 4356\ ºC \cdot day \cdot 24h/day = 52146\ kWh \approx 52{,}1\ MWh$

The building of WSTPB is equipped by central hot and cold domestic water, that’s why heating demand for HDW won’t be taken into account.

In this case the value of total energy demand will be 52,1 MWh per year.

3.3 Calculation of total water flow for heating system

The total water flow will be calculated using **formula 12**:

$$Q_T = \frac{\Phi_T^B}{\rho \cdot C_p \cdot (t_f - t_r)},$$ **Formula 12**

Where:

Q_T = total water flow for heating system, l/s

Φ_T^B = total heat losses through the building envelope, kW

ρ = density of water, kg/m^3

C_p = specific heat capacity of water, kJ/(kg·ºC)

t_f = flow temperature, ºC

t_r = return temperature, ºC

For the purposes of water flow calculations the following initial data should be presented:

Total heat losses through the building envelope Φ_T^B = 22 kW

Density of water ρ = 1000 kg/m^3

Specific heat capacity of water C_p = 4,2 kJ/(kg·ºC)

Flow and return temperatures for underfloor heating:

Flow temperature t_f = +35 ºC

Return temperature t_r = +30 ºC

Flow and return temperatures for hydraulic heating system with modern radiators:

Flow temperature t_f = +45 ºC

Return temperature t_r = +35 ºC

The two variants of calculations of total water flow will be done. After comparison of two calculated water flows for underfloor heating and heating system with modern radiators the decision about choosing the distribution system for the WSTPB heating system will be made.

The total water flow for underfloor heating is calculated as follows:

$$Q_T = \frac{\Phi_T^B}{\rho \cdot C_p \cdot (t_f - t_r)} = \frac{22 \frac{kJ}{s}}{1000 \frac{kg}{m^3} \cdot 4,2 \frac{kJ}{kg \cdot °C} \cdot (35-30)°C} = 0,00105\ m^3/s = 1,05\ l/s = 3,77\ m^3/h$$

The total water flow for heating system with modern radiators is calculated as follows:

$$Q_T = \frac{\Phi_T^B}{\rho \cdot C_p \cdot (t_f - t_r)} = \frac{22 \frac{kJ}{s}}{1000 \frac{kg}{m^3} \cdot 4,2 \frac{kJ}{kg \cdot °C} \cdot (45-35)°C} = 0,00052\ m^3/s = 0,52\ l/s = 1,88\ m^3/h$$

The calculated values of total water flow for the underfloor heating and for the heating system with modern radiators are very small, so these values will not essential impact on heating system characteristics. Thus both the underfloor heating and the heating system with modern radiators are acceptable as distribution systems for the WSTPB heating system. It means that the distribution system for the WSTPB heating system might be based both on the underfloor heating and on the heating system with modern radiators or their combination where it is needed. That's why the underfloor heating as a distribution system is chosen.

It is known that the underfloor heating as a heat distribution system suits very well for heating system based on heat pump technology, because of low temperatures in flow water, comfortable temperature of the floor surface and optimal vertical gradient. But this distribution system has some limitations such as limits for output from the floor. The heat output per one squared meter of floor area is calculated using **formula 13**:

$$q = \frac{\Phi_{room}}{A_{room}} = (\alpha_r + \alpha_c) \cdot (\overline{T_s} - T_r) = \alpha_{r+c} \cdot (\overline{T_s} - T_r)\,, \qquad \textbf{Formula 13}$$

Where:

q = heat output per area, W/m^2

α_r = coefficient of heat radiation, $W/(m^2 \cdot {}^oC)$ ($\alpha_r = 6\ldots7\ W/(m^2 \cdot {}^oC)$)

α_c = coefficient of heat convection, W/(m2·ºC) (α_c = 4…5 W/(m2·ºC))

α_{r+c} = coefficient of total heat flow convection, W/(m2·ºC) (α_{r+c} = 10…12 W/(m2·ºC))

$\overline{T_s}$ = the mean temperature of floor surface, ºC

T_r = the room temperature, ºC

According to the **formula 13** rising the floor surface temperature with 1ºC, the heat output from floor increases about 10 $W/(m^2 \cdot {}^oC)$. But temperature difference should not be more than 6ºC, so the heat output from the surface is not more than 60…72W/m^2. Thus if the heat load is more than 60…72W/m^2, then the other additional heat emitter (e.g. radiator) must be installed.

Using to the calculations presented in **table 5** the needed heat out put per 1m^2 might be calculated for each room using **formula 13**. The example is shown below:

$$q_{room1} = \frac{\Phi_{room1}}{A_{room1}} = \frac{528W}{10,5m^2} = 50,3W/m^2$$

The values of needed heat output for other rooms are varying from 3,3W/m^2 (Room 11) to 60,9W/m^2 (Room 8). It means that the heat output from the floor surface will be enough for heating purposes.

The example of dimensioning the Room 1 and initial data are presented below:

The needed heat output for the room's area is $q_{room1} = 50,3W/m^2$

The room temperature is T_{room1} = +16ºC

The flow medium temperature is $T_f = +35ºC$

The return medium temperature is $T_r = +30ºC$

Thickness of the layer above the pipe $s_u = 45mm$

Thermal resistance of surface covering $R_{\lambda,B} = 0{,}02(m^2 \cdot ºC)/W$

The calculations were made using “Uponor. Heating and cooling solutions. Undefloor heating. Technical guidelines” /12/. In order to use the diagrams, which are presented, in /12/ the heating medium differential temperature ΔT_H must be calculated. The heating medium differential temperature ΔT_H is calculated using **formula 14**:

$$\Delta T_H = \left| \frac{T_f - T_r}{\ln \frac{T_f - T_i}{T_r - T_i}} \right|, \qquad \textbf{Formula 14}$$

/12/

Where:

ΔT_H = the heating medium differential temperature, K

T_i = designed indoor temperature, ºC

T_f = flow medium temperature, ºC

T_r = return medium temperature, ºC

According to the **formula 14** the heating medium differential temperature ΔT_H has the following value:

$$\Delta T_H = \left| \frac{T_f - T_r}{\ln \frac{T_f - T_i^{room1}}{T_r - T_i^{room1}}} \right| = \left| \frac{35-30}{\ln \frac{35-16}{30-16}} \right| = 16{,}4K$$

According to the diagram for PE-Xa pipes with dimensions 20x2 mm /12,p.10/ the distance between pipes should be $T_{pipe} = 300mm$ and the heat output from the floor surface will be $q_{floor} = 56W/m^2$. Thus $q_{room1} = 50{,}3W/m^2 < q_{floor} = 56W/m^2$. It means that these pipes (PE-Xa pipes 20x2 mm) and distance between them are acceptable. But the total length of the pipes for this room must be checked using **formula 15**:

$$L_T^{room} = \frac{A_{room}}{T_{pipe}}, \qquad \textbf{Formula 15}$$

Where:

L_T^{room1} = total pipe length for the certain room, m

A_{room} = total floor area of the certain room, m^2

T_{pipe} = the pipe spacing, m

The total length of the pipes for Room 1 will be as follows:

$$L_T^{room1} = \frac{A_{room1}}{T_{pipe}} = \frac{10,5m^2}{0,3m} = 35m < 100m$$

Thus the total length of the pipes for Room 1 is acceptable too.

According to the diagram /12,p.10/ the floor surface differential temperature is:

$$T_{surface}^{diff} = T_{surface}^{mean} - T_i = 5,3°C$$

The mean floor temperature in this case will be:

$$T_{surface}^{mean} = T_{surface}^{diff} + T_i = 5,3°C + 16°C = 21,3°C$$

The Room 14 (Main WSTPB hall) has very big area A_{room14} = 467,3m^2. If the distance between pipes is T_{pipe} = 400mm, the total length of pipes should be L_T = 1168m. This problem might be easily solved by dividing the total length into 16 smaller pieces. Thus the length of the pipes in each of 16 small circuits will be only 73m (<100m) that is acceptable.

According to the dimensions of the WSTPB the length of the longest circuit is about 200m. Pressure drop in pipes is about R = 100Pa/m = 0,01m/m. Thus the pressure drop in this circuit will be approximately:

$$H= R \cdot L \cdot Z_f = 0,01m/m \cdot 200m \cdot 1,3 = 2,6 \text{ m}$$

The value of water flow as calculated earlier is Q = 3,77 m^3/h

According to these parameters we can choose the circulation pump:

Wilo Star-RS 30/7: H = 3,5m; Q = 3,77m^3/h; P = 0,06 kW /18/

3.4 Choosing suitable heat pump

According to the heat loss calculations the total heat load for WSTPB is $\Phi_T^B = 22$ kW.

According to the manufacturer's catalog the suitable model of ground source heat pump is NIBE Fighter 1330-22kW 3-P 400V manufactured in Sweden by NIBE /13,p.7/.

The heat pump NIBE Fighter 1330-22kW 3-P 400V characteristics:

Dimensions: 1580x600x625mm (HxWxD)

Recorded capacity: 4,8 kW

Heat capacity: 23,1 kW

COP1: 2,5 – 3,5

Weight: 330 kg

The cost of this type of heat pump is 673200 rub /17/

4. ECONOMICAL CALCULATIONS

The economical calculations is one of very important chapters because the results of calculations presented in this chapter helps to estimate capital costs of heating system for water supply treatment plant building based on heat pump technology, it's operation costs, and pay back time. Moreover it gives a chance to describe effectiveness of using this technology for industrial buildings from the side of cost-effective way of using new technologies in 15 years period in comparison of convenient boiler heating system, which might be used, for the same purposes.

According to the plan of the WSTPB the total floor area is A_{floor} = 682 m^2

4.1 Estimation of capital costs

Capital cost – the cost of heading system, including main and additional equipment, distribution system, installation works and other associated works, which must be paid once before the heating system will be started up.

According to the information presented on the web site of one of the manufacturers /14/ the cost of each 1kW of heat pump energy include drilling or other associated works costs approximately from 800 € - 1000 €. So if we had an additional 22 kW of heat pump it would have increased the installation cost by c. 17600 € – 22000 €. Thus the estimated capital costs including heat pump installation, drilling and mounting the underfloor heating for WSTPB with heat load 22 kW will be $C_{cap\ ins}$ = 22000 €.

The cost of heat pump NIBE Fighter 1330-22kW 3-P 400V is 673200 rub.

According to the currency exchange rate 1 € = 41,9 rub the estimated capital cost will be:

$$C_{cap\ HP} = 22000\ € \cdot 41{,}9\ rub/€ + 673200\ rub = 921800\ rub + 673200\ rub = 1595000\ rub.$$

According to the information presented on the web site of one of the manufacturers the cost of the electrical boiler DACON PTE-22 is 37767 rub /16/.

As it known the electric boiler might be used as a heat source for heating system equipped with underfloor heating distribution system. In this case the distribution systems for the heat pump and for the electric boiler will be the same – underfloor heating.

Thus the estimated capital cost of heating system based on electrical boiler will be:

$$C_{cap\ EB} = 37767\ rub + 921800\ rub = 959567\ rub.$$

4.2 Estimation of operating costs

Operating cost – the cost of maintaining of heating system including annual service works, monthly/annual fuel cost, electricity consumption, repairing works, which must be paid monthly or annually during the whole period of using the heating system.

The information presented below will help to estimate the operating costs of using heating system for water supply treatment plant building based on heat pump technology per one year. The price for electricity in Saint-Petersburg is 2,59 rub/kWh = 0,06 €/ kWh

The total energy demand is Q_{Tyear} = 52,1 MWh/year.

The highest peak of electricity consumption of an electric boiler is 22 kW /16/
Power demand of the heat pump is 4,8 kW /13/
The electricity consumption of circulation pumps for the distribution systems in both cases is too low (about 0,06 kW) to take it into account.
The electricity consumption of circulation pump for ground loops for the heat pump is about Q_{circ} = 0,9 kW/h.
The time of operation is T_{oper} = 220day·24h/day = 5280 h/year
Thus the total electricity consumption for each of these systems will be:

1) $Q_{EB} = Q_{Tyear}$ = 52,1 MWh/year
2) $Q_{HP} = Q_{Tyear}/COP1 + Q_{circ}\cdot T_{oper}$ = 52100 kWh/year/2,5 + 0,9 kW/h· 5280h/year = = 20840 kWh/year + 4752 kWh/year = 25592 kWh/year ≈ 25,6 MWh/year

According to these calculations we can calculate the annual operating costs for both systems:

1) $C_{op\ EB}$ = 52100 kWh/year·2,59 rub/kWh ≈ 134939 rub/year
2) $C_{op\ HP}$ = 25600 kWh/year·2,59 rub/kWh ≈ 66304 rub/year

The delta of operation costs between the electric boiler and the heat pump is our savings:

$\Delta C_{op} = C_{op\ EB} - C_{op\ HP}$ = 134939 rub/year – 66304 rub/year = 68635 rub/year

These values help to estimate the pay back time for the heating system for WSTPB based on heat pump technology and heating system based on electrical boiler.

4.3 Calculations of cost effectiveness

The minimum lifetime for these systems is 10 years for the electric boiler and 15 years for the heat pump. Thus the calculations of the cost efficiency of these systems for the time period of n = 10 years and n = 15 years will be done.
The cash balance for each period will be corrected by using discount factor:

$D = 1/(1+i)^n$, **Formula 16 /19/**

Where:

D = discount factor

i = interest rate

n = number of periods

The interest rate for this type of project is 19 % /20/

The value of inflation in Russian Federation is 10 % for year 2010 /20/

The prediction of rising of price for electricity 15 %/year /21/

Nowadays the most popular indices of cost effectiveness are:

1. NPV – net present value
2. PP – payback period
3. IRR – internal rate of return
4. PI – profitability index

1. NPV – net present value

$$NPV = \sum_{n=t1}^{t2} D \cdot Inflow - \sum_{n=t1}^{t2} D \cdot Outflow ,$$ **Formula 17 /19/**

Where,

n = number of period of the project

t1, t2 = limits of calculations

The investor must choose from the projects, which have the value of NPV>0. If the value of NPV<0, it means that this project is ineffective.

2. PP – payback period

The payback period is the time of returning of the investments. There are two types of payback period: static payback period and dynamic payback period (discounted payback period.

- Static payback period

$$\sum_{n=t1}^{t2} Inflow - \sum_{n=t1}^{t2} Outflow = 0$$ **Formula 18 /19/**

- Dynamic payback period

$$\sum_{n=t1}^{t2} D \cdot Inflow - \sum_{n=t1}^{t2} D \cdot Outflow = 0 ,$$ **Formula 19 /19/**

Where:

n = number of period of the project

t1, t2 = limits of calculations

D = discount factor

3. IRR – internal rate of return

If IRR is calculated on the NPV base, it shows us the maximum loan interest rate for the total investment cost for the whole period of the project.

$$NPV(IRR)=\sum_{t=0}^{n}\frac{CF_t}{(1+IRR)^t}-\sum_{t=0}^{n}\frac{I_t}{(1+IRR)^t}=0,$$ **Formula 20 /19/**

Where:

CF_t = the inflow of money for period t

I_t = the sum of the investments for period t

n = the total number of periods t = 0,1,2,…, n

4. PI – profitability index

The profitability index shows us the relative profitability of the project or the discounted cost of the inflow during the operation of the project per one unit of the investments.

PI = NPV/I, **Formula 21 /19/**

Where:

NPV = net present value

I = investments

These formulas are very effective way of estimation the economical effectiveness of the whole project of heating system for WSTPB including the capital costs not only for the heat pump but for distribution system too.

In the case of this Bachelor Thesis the two system: heat pump and electric boiler will be compared from the side of capital costs of the heat pump and the electric boiler without taking into account the capital cost of distribution system.

The shortest way of comparison of two systems will be done with using the following formulas:

i = IL – Inf , **Formula 22**

Where:

i = real interest of loan, %

IL = interest of loan, %

Inf = the level of the inflation, %

$$a=\frac{i\cdot(1+i)^t}{(1+i)^t-1},$$ **Formula 23**

Where:

a = time discount factor

i = real interest of loan

t = operation period,

Case 1. Heat pump

The capital cost of heat pump is 673200 rub.

The operation period t = 15 years.

The real interest of loan is:

i = IL – Inf =19 % – 10 % = 9 % = 0,09

The time discount factor is:

$$a = \frac{0{,}09 \cdot (1+0{,}09)^{15}}{(1+0{,}09)^{15} - 1} = 0{,}124$$

The capital cost with using time discount factor is:

$$C^{a}_{HP} = a \cdot C_{HP} = 0{,}124 \cdot 673200 = 83477 \text{ rub}$$

Case 2. Electric boiler

The capital cost of heat pump is 37767 rub.

The operation period t = 10 years.

The real interest of loan is:

i = IL – Inf =19 % – 10 % = 9 % = 0,09

The time discount factor is:

$$a = \frac{0{,}09 \cdot (1+0{,}09)^{10}}{(1+0{,}09)^{10} - 1} = 0{,}156$$

The capital cost with using time discount factor is:

$$C^{a}_{HP} = a \cdot C_{HP} = 0{,}156 \cdot 37767 = 5892 \text{ rub}$$

According to the **formulas 22** and **23** the estimation of the economical effectiveness of the heating systems for WSTPB based on heat pump technology and the electrical boiler is done. The calculations were made with using the program Excel 2003. The results of calculations are presented in the **table 6**:

TABLE 6. Results of economical effectiveness calculations.

Year of operation period	1	2	3	4	5	6	7	8
Case 1. Capital cost	83477	83477	83477	83477	83477	83477	83477	83477
Case 1. Operation cost	66304	76250	87687	100840	115966	133361	153365	176370
Case 1. Cumulative cost	149781	159727	171164	184317	199443	216838	236842	259847
Case 2. Capital cost	5892	5892	5892	5892	5892	5892	5892	5892
Case 2. Operation cost	134939	155180	178457	205225	236009	271411	312122	358940
Case 2. Cumulative cost	140831	161072	184349	211117	241901	277303	318014	364832
Savings from using HP	-8950	1345	13185	26800	42458	60465	81172	104985

Year of operation period	9	10	11	12	13	14	15
Case 1. Capital cost	83477	83477	83477	83477	83477	83477	83477
Case 1. Operation cost	202825	233249	268237	308472	354743	407954	469148
Case 1. Cumulative cost	286302	316726	351714	391949	438220	491431	552625
Case 2. Capital cost	5892	5892	5892	5892	5892	5892	5892
Case 2. Operation cost	412781	474699	545904	627789	721957	830251	954789
Case 2. Cumulative cost	418673	480591	551796	633681	727849	836143	960681
Savings from using HP	132371	163864	200082	241732	289629	344712	408056

The values of the indices of the project effectiveness are:

1. The time when the cumulative costs in Case1 and Case2 become equal is 1,87 years
2. Payback period of Heat Pump is PP = 9,3 years
3. The total savings from using HP for operation period of Heat Pump after payback period is 1428706 rub

These values of indices of the project effectiveness and other calculations let us to conclude that the heating system for WSTPB based on heat pump technology is more cost effective than the heating system based on electric boiler.

According to the technical and economical calculations the heating system for WSTPB based on heat pump technology was chosen as the most suitable for this building. The scheme of the system might be described as the heat pump system with the energy piles as a system of obtaining a low potential geothermal energy, the NIBE Fighter 1330-22kW 3-P 400V heat pump as a device for transformation of low potential energy to the energy with the required potential level for heating purposes, and underfloor heating as the heat distribution system.

The main parameters of designed system are presented below:

Heat output of the heating system: 23,1 kW

Electricity consumption: 4,8 kW

COP1 of heat pump: 2,5 – 3,5

Total electricity consumption during the heating period: 25,6 MWh/year

Total heat energy demand during the heating period: 52,1 MWh/year

The cost of 1MWh of heating energy produced by this heating system: 1272,6 rub/MWh

The annual cost of heating energy: 66304 rub/year

The cost of heating energy per month after pay back period: 5525 rub/month

Payback period PP = 9,3 years

Approximate operation period: 15 years

These characteristics can be estimated as very good for such new industrial building as water supply treatment plant building.

CONCLUSION

To achieve the aims, which have been formulated in the beginning, the big work was done. For these purposes the comparison of the benefits and limits of the heat pump method of heat producing with the conventional method (electric boiler) of heat producing was done from technical and economical reasons. Moreover the calculations of heat losses through the building envelope, the selection of suitable heat pump and defining its properties, and other required actions were made.

According to the calculations and comparison of benefits and limits of the heating system based on ground source heat pump technology, the designed heating system demonstrates very good results and economical efficiency. Moreover this heating system besides the high economical efficiency has some environmental benefits.

One of the most important environmental benefits is the possibility to reduce the energy consumption, which is used for space heating and preparation of domestic hot water.

Thus this technology in the combination with good thermal insulation of the building envelope is suitable, environmentally friendly and economically reasonable for heating purposes for water supply treatment plants, and especially wastewater treatment plants, which have warm effluents during the whole year and other industrial buildings in Russian Federation because of its high heat energy efficiency and small electricity consumption.

All in all this heating system for industrial buildings based on heat pump technology must be not only deep studied by the Russian specialists but this technology must be taken in use for heating purposes of these buildings as fast as possible.

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