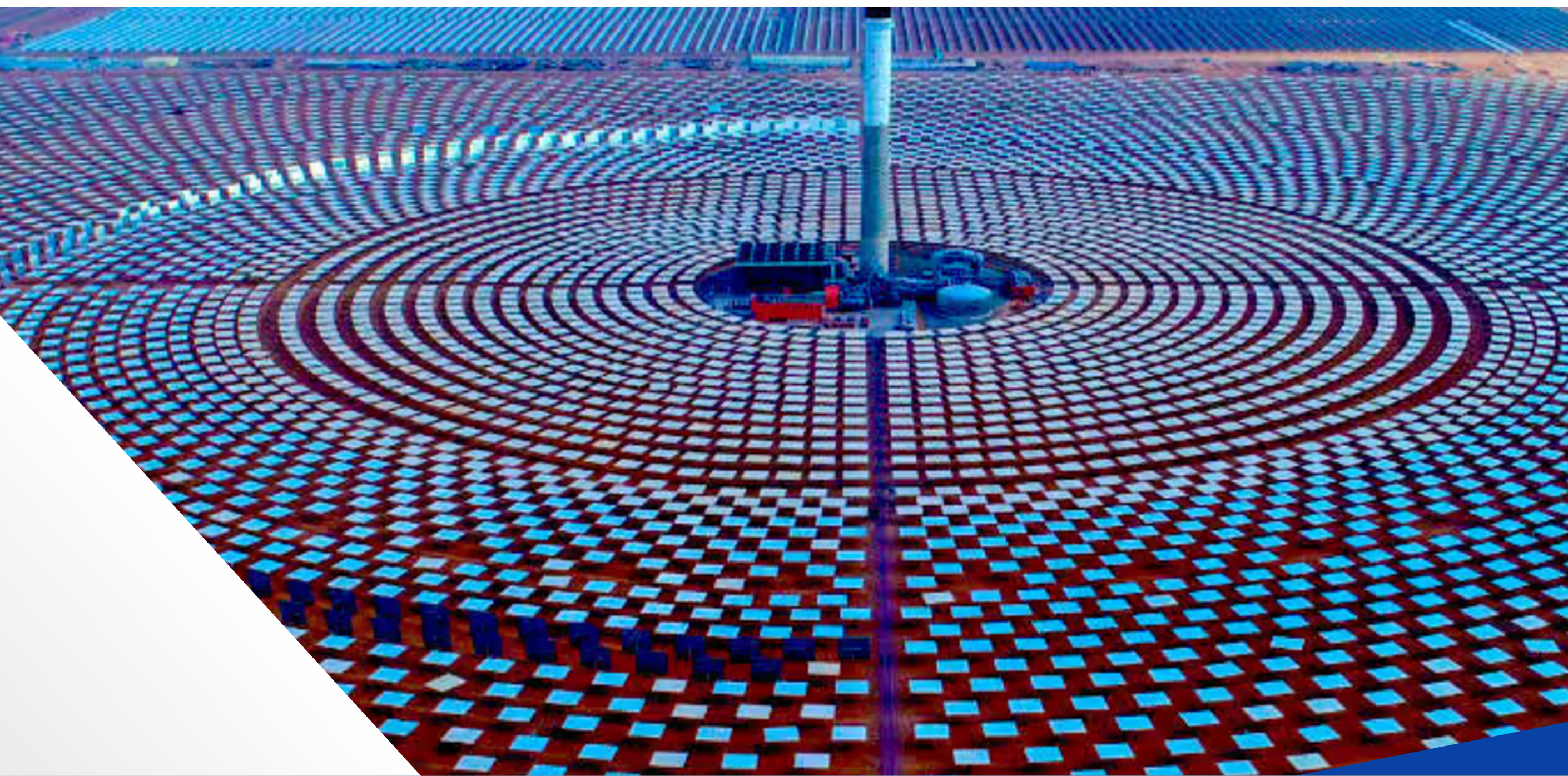




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SOLAR THERMAL DEVICES AND SYSTEMS

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Tashkent, 2025. – 270 pages.

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The textbook is based on a new program based on the analysis of international curricula and programs for higher education. The textbook *Solar Thermal Devices and Systems* provides a comprehensive foundation in the principles, design, modeling, and practical application of solar thermal technologies. The course begins with the fundamentals of solar radiation, including solar geometry, spectral distribution, atmospheric effects, and methods for estimating solar energy availability. It then introduces the theory of heat transfer—conduction, convection, and radiation—as applied to solar energy systems. Special emphasis is placed on different types of solar thermal collectors, such as flat-plate collectors, evacuated tube collectors, and concentrating collectors. The textbook explains their operating principles, thermal efficiency analysis, performance parameters, and energy balance equations. Advanced topics include thermal energy storage methods, phase change materials, and system optimization techniques.

The textbook was prepared as one of the practical results of the Project: “Development of the targeted Educational program for Bachelors in Solar Energy in Uzbekistan” (101128871 - DEBSEUz - ERASMUS-EDU-2023-CBHE) within the framework of the European Union’s “Erasmus+” program.

Annotation

This textbook titled “Solar thermal devices and systems” provides a comprehensive and systematic analysis of the principles, technologies, and applications of solar energy conversion to thermal energy. It is aimed at equipping students and practitioners with a comprehensive understanding of solar thermal systems, how they are designed, operated, and applied to the environment.

The content is organized around a theme in six chapters covering:

Chapter I presents the composition of the Sun, the nature of solar radiation, its spectral character, and measurement methods. It highlights crucial subjects such as solar constants, regimes of radiation, and geographical variations of solar intensity.

Chapter II addresses the physics of heat transfer — conduction, convection, and radiation — by applying mathematical models and differential equations that provide the basis for energy conversion processes.

Chapter III involves various types of solar collectors and concentrators like flat-plate collectors, vacuum tube collectors, and air heating units. It discusses optical properties, heat loss mechanisms, and high-temperature device thermal efficiencies.

Chapter IV is dedicated to solar heating systems, such as passive, active, and hybrid technologies. Detailed comparisons are given in terms of design, efficiency, environmental performance, and economic feasibility. Static and dynamic models, and energy-efficient architectural solutions are encompassed.

Chapter V addresses heat storage systems, considering different strategies such as sensible, latent, and thermochemical storage. Material selection and determination of storage capacity needed to design the system efficiently is explained.

Chapter VI illustrates actual solar thermal hardware like solar water heaters, purifiers, chillers, and dryers. Emphasis is on heat and mass transfer operations, system design, and performance evaluation methods.

One of the strongest features of this textbook is its contextual focus on Uzbekistan's climate and sun resources, showing the nation ready to integrate solar technology into its energy sector. Using actual systems and active national programs, the book bridges theory with reality.

It is written for undergraduate and graduate students majoring in renewable energy, environmental and mechanical engineering, researchers, instructors, and professionals in the solar energy sector. It is both an academic textbook and a resource for the design and utilization of sustainable solar thermal technologies.

INTRODUCTION

Today, the share of traditional fuels in the total volume of energy generated and used in various industries in the world is 86%, hydropower - 7%, nuclear fuel - 6%, and other types of resources - 1%. With the ever-increasing demand for traditional fuels, their sources, on the contrary, are decreasing, in other words, energy production lags behind its consumption. The impact of the technosphere on the environment is rapidly increasing.

All these are fundamentally interconnected problems. The development of the economies of the world's countries, including China, India, Russia, and similar large countries, requires the extraction of large quantities of natural resources from the Earth's interior, which leads to the release of a large amount of toxic substances into the biosphere during their extraction, transportation, and use. According to experts, about 200 million tons of solid particles, 200 million tons of sulfur dioxide, 700 million tons of carbon, and 150 million tons of nitrogen oxides are released into the Earth's atmosphere annually. As a result, various anomalous phenomena ("acid rains", "climate warming", "depletion of the ozone layer", "desertification", etc.) and disasters occur in nature, as well as an increase in diseases associated with the penetration of toxic chemicals into the human body due to biological metabolism.

Scientists have repeatedly emphasized that it is more rational to spend oil on the production of vital products than on oil combustion. Perhaps, in time, it is possible to ban the use of oil as a vital raw material, at least for heating purposes, but this should be done not through political means, but through common sense.

The standard of living of humanity is becoming directly dependent on the level of its energy supply. Wasting energy is a crime, and the search for new types of energy is becoming an important task.

It is important that every member of society understands that excessive energy consumption and associated environmental pollution can, over time, undermine the progress of society as a whole. Every step taken on the path to "cleansing" energy must be supported.

Thus, the energy problem in the world as a global problem requires bold steps to improve the current system of energy use and the use of renewable and environmentally friendly energy sources. In particular, the large-scale development of technologies for the use of inexhaustible and environmentally friendly solar energy serves as a powerful lever in ensuring the continuity and quality of energy supply, contributing to the preservation of environmental cleanliness.

Whatever achievements have been made throughout human history are the result of the intelligence of people who lived in different eras. Therefore, the intelligence and enthusiasm of our contemporaries are a reliable tool for finding and implementing more

efficient ways to use environmentally friendly energy sources, primarily solar energy, that can fully meet our energy needs.

It can be said that the 21st century has brought a new look and a spirit of hope to solar energy around the world. This field is developing rapidly in various directions. In more than 70 countries around the world, from Northern Scandinavia to the hot deserts of Africa, scientific research and practical work are being carried out on the basis of solar energy programs. Solar devices are used to heat and cool buildings, desalinize water, generate electricity, and carry out various technological processes.

It is worth noting that Uzbekistan has great potential for the use of solar energy. In our republic, along with fundamental scientific research on the conversion of solar energy into heat and electricity, obtaining high temperatures and heliomaterials, large-scale practical work is being carried out. The fact that the largest and most unique solar furnace in the world is operating and efforts to build solar power plants, as well as the establishment of the International Center for the Use of Solar Energy, are evidence of our opinion.

This manual was written to familiarize students with the general issues of converting solar energy into thermal energy. It contains information on the structure and source of energy of the Sun, the spectral composition of solar radiation, types and methods of their measurement, and the determination of the radiation power falling on solar devices located in different directions. Information is provided on the climatic conditions and radiation regime of Uzbekistan and the existence of wide opportunities for the use of solar energy. As typical examples of solar thermal devices, detailed information is provided on solar water heaters, water purifiers, dryers, concentrators and furnaces, which are gaining wide popularity not only in the future, but also today in various countries of the world.

CHAPTER-I. THE SUN AND ITS STRUCTURE. THE NATURE OF SOLAR RADIATION

1.1. The Sun and its source of energy

The Sun is one of billions of stars in the vast universe, it is the main cause of the emergence and development of life on Earth, the primary source of all types of energy on it. At the same time, it is the central body of a system that is constantly moving in space and is called the Solar System. The Earth is located in the Solar System in such a way that (on average 149.5 million km from the Sun) this location has endowed it with life, which does not exist on any other planet.

The Sun is an average representative of the stars in the Universe in terms of size and temperature. Since it is millions of times closer to the Earth than other stars, it is visible from a much larger angle (32°). The Sun has a diameter of 1.395 million kilometers, and a volume of $1.4 \cdot 10^{18} \text{ km}^3$, that is, 1.3 million times larger than the volume of the Earth. 99.87% of the mass of the Solar System is accounted for by the mass of the Sun, which is 330 thousand times greater than the mass of the Earth, or $2 \cdot 10^{27}$ tons. While the average density of the Sun is 1410 kg/m^3 , the density in the central part is extremely high $\sim 10^5 \text{ kg/m}^3$. The Sun, together with its “family”, moves along its orbit at an average speed of 250 km/s. The acceleration of gravity on its surface is 273.8 m/s^2 .

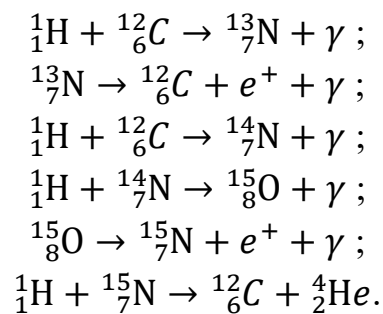
Each planet around the Sun receives a certain amount of energy depending on the size of its cross-section and its distance from it. In particular, the amount of energy falling on the Earth every second is 2.2 billion times less than all the energy coming from the Sun, and is $17.4 \cdot 10^{17}$ Joules. 36% of this energy is reflected by the atmosphere and 17% is absorbed. Only the rest reaches the Earth’s surface. Even if we take into account that half of the energy that arrives is spent on evaporating sea and ocean waters and 1% is consumed by the plant world, the remaining amount of energy is $35 \cdot 10^{17}$ kW-hours per year. This is about 39 thousand times more than the total energy consumed by all mankind in one day.

Astrophysicists predict that the Sun is about 5 billion years old and that it will continue to shine for at least that long. There are various opinions and dozens of scientific hypotheses about how the Sun, which has been continuously radiating energy for such a long time, can compensate for the energy lost by the Sun. The most reliable of these scientific conclusions was the idea put forward in 1938–1939 by astrophysicists A. Eddington, K. Weissecker and G. Boeth in which they concluded that the energy source of stars, including the Sun, is nuclear reactions occurring at very high temperatures. Theoretical calculations show that the temperature at the center of stars (including the Sun) is several million degrees, providing favorable conditions for a thermonuclear reaction.

The continuous repetition of the reaction in the center of the Sun, in which four protons fuse to form a single helium nucleus, replenishes the energy that is radiated into space by its radiation. The energy released in the formation of a single helium nucleus is $4.3 \cdot 10^{-12}$ J, calculated using Einstein's famous formula expressing the relationship between energy and mass.

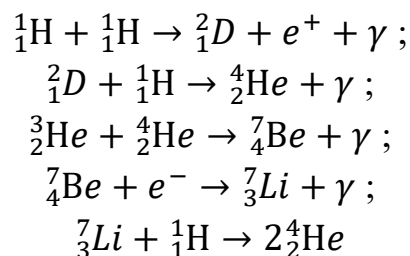
Calculations show that the power released from the center of the Sun in this way per second is $4 \cdot 10^{26}$ W, that is, the energy it loses per second. Currently, two sequence reactions are known for the formation of a helium nucleus from four protons: the first type of thermonuclear reaction is the reaction in the carbon-nitrogen cycle.

This reaction occurs in the Sun's core, which has a temperature of 20 million °C and a density of $115 \cdot 10^3$ kg/m³, in the following order:

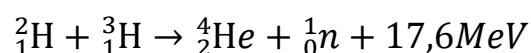


During this reaction, four hydrogen nuclei are lost to form one helium nucleus, with ${}^{12}_6\text{C}$ acting as a catalyst. The conversion of 1 kg of hydrogen into helium releases heat equivalent to the energy from burning 15,000 tons of coal.

A thermonuclear reaction can also proceed in the proton-proton cycle in the following order:



If the reaction in the carbon-nitrogen cycle occurs in the core of the Sun, then the reaction in the proton-proton cycle can occur throughout its entire volume. One of the most likely thermonuclear reactions is the process of converting deuterium and tritium nuclei (five protons) into helium, that is, one helium nucleus and a neutron; the sum of the masses of the helium nucleus and the proton is less than five proton masses, which means that the mass “lost” during the reaction is converted into energy:



At the center of the Sun, gravity has compressed matter to such a high density and temperature that fusion reactions are possible. These reactions are the source of the energy that the Sun continuously radiates into space.

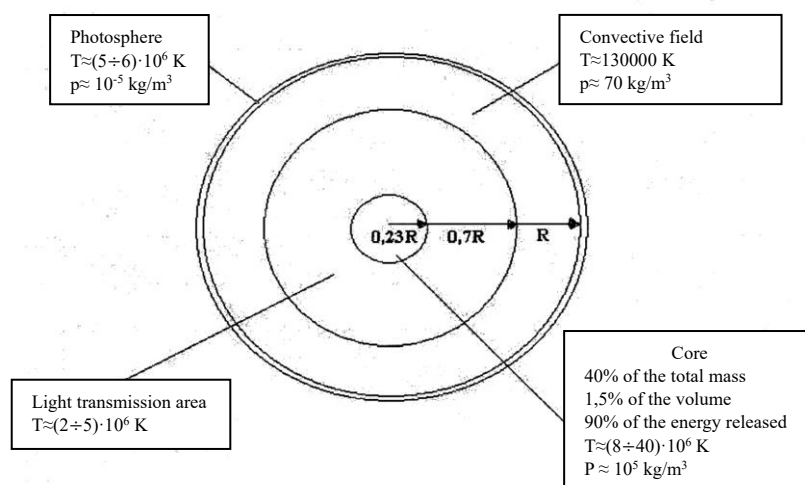


Figure-1.1.1. A schematic diagram of the structure of the Sun.

The core, which occupies only one-sixty-fourth of the Sun's total volume and extends to 0.23 of its radius, contains 40% of its mass and produces 90% of its energy. According to various estimates, the temperature in the inner parts of the Sun's center is approximately $8 \cdot 10^6$ – $40 \cdot 10^6$ K, and the pressure reaches $250 \cdot 10^9$ atm. Although the core of the Sun's atoms is thousands of times denser than that of metals, the high temperature keeps it in a gaseous state. There is no crystalline structure typical of solids here, and electrons and “naked” nuclei move as free particles. During thermonuclear reactions, 0.007 grams of each gram of hydrogen that reacts is converted into energy.

To date, 74 different solar elements have been clearly proven. There is no doubt that there are also those controlled by the Sun, but they are so few that it is impossible to determine them by spectral analysis. According to calculations, the composition of the Sun, by the number of atoms, is 91% hydrogen and 9% helium. By mass, hydrogen makes up 70%, helium 27%, since the nucleus of a helium atom is four times heavier than the nucleus of a hydrogen atom. Surprisingly, helium was first discovered on the Sun based on the spectra of solar radiation, and it was also detected on Earth. The name of this element is also derived from the Greek word “Helios” - Sun.

The energy generated by thermonuclear reactions travels from the Sun's core to its surface in the form of radiation, and naturally, in this process, the temperature, density, pressure, and average energy of the photons decrease. In the part called the light energy transfer zone (in the part extending from 0.23R to 0.7R), the process of “crushing” of quanta occurs, that is, high-energy gamma rays emanating from the core first turn into X-rays, then into far ultraviolet, and finally into the most characteristic part of solar energy - visible light radiation. In the depths of the Sun, 630 million tons

of hydrogen are converted into helium every second, and during this time it loses more than 4 million tons of mass due to radiation. The energy generated by thermonuclear reactions is distributed through the photosphere into space in the form of electromagnetic radiation. The intensity of radiation emitted from the Sun's surface, which has a temperature of 6000 °C, is 70–80 thousand kW/m². It is undeniable that the Sun, in addition to fulfilling its role as the main cause of life on Earth, serves as an inexhaustible and environmentally friendly source of energy for humanity.

1.2. Spectral composition of solar radiation

The effects of sunlight on the Earth's flora and fauna, as well as inorganic substances, vary. Therefore, studying the spectral composition of sunlight is of practical importance. This is especially important in the practical use of solar energy.

Most of the energy released in the Sun as a result of the “burning” of hydrogen is distributed to the Universe through the photosphere as radiation. The photosphere is considered a source of electromagnetic radiation, and the radiation emitted by it includes everything from ultraviolet to infrared rays. In other words, the Sun emits rays of various colors, like any solid or liquid object that is heated. The radiation of the photosphere can be compared to the radiation of an absolutely black body at a temperature of 5800 K. An absolutely black body is an object that absorbs all the radiation falling on it, regardless of its spectral composition. Such an object can be used as a source of radiation with known properties, that is, as a light standard (at the solidification temperature of pure platinum).

The electromagnetic radiation of the Sun has a long spectrum, lying in the range from 0.1 μm to 30 m. The chromosphere and corona of the Sun are sources of radiation lying outside the visible part of the spectrum (0.4 - 0.76 μm). Since the temperature of the gases that make up the corona is about 10⁶ K, the main part of its radiation is X-rays and ultraviolet rays (10⁻³ - 0.2 μm). The radiation of the corona and chromosphere has a significant impact on the formation of the Earth's ionosphere.

More than 99 percent of the energy of solar radiation is in the wavelength range from 0.28 to 6.0 μm (0.28–0.40 μm–ultraviolet, 0.40–0.76 μm–visible, and 0.76–6.0 μm–infrared). At the same time, 0.5 percent and 0.4 percent of solar radiation energy fall on the wavelength range from 10⁻² μm to 0.28 μm and from 6 μm to 5·10² μm, respectively. The colors of solar radiation by wavelength (λ) are given in table-1.2.1.

The radiation near the surface of the Sun does not have time to reach thermal equilibrium, so the spectrum of the Sun does not completely match the spectrum of balanced thermal radiation. Each element atom emits light at its own frequency. Due to the selective absorption of light by the relatively colder gas atoms on the surface of the Sun, the spectrum contains Fraunhofer dark lines (about 20 thousand) formed by

all the substances in the Sun's atmosphere (these lines are named after the German physicist Josef Fraunhofer, who first studied the phenomenon).

Table-1.2.1

Color characteristics of sunlight

Color	Wavelength range, μm	Typical λ , μm	Color	Wavelength range, μm	Typical λ , μm
Ultraviolet			Yellow-green	0,55-0,57	0,56
Vacuum	10^{-3} -0,20	-	Yellow	0,57-0,58	0,58
Far	0,20-0,30	-	Orange	0,58-0,67	0,60
Near	0,30-0,40	-	Red	0,62-0,76	0,64
Purple	0,40-0,45	0,43	Infrared		
Blue	0,45-0,48	0,47	Near	0,76-25,0	-
Cerulean	0,48-0,50	0,50	Far	$25,0-10^4$	-
Green	0,50-0,55	0,53			

The spectral analysis method allows us to determine a lot of information about the surface of the Sun. For example, the wavelength and intensity corresponding to the Fraunhofer lines can be used to determine the chemical composition of the gas, and the width of the lines can be used to determine the temperature of the absorbing medium, etc. Unfortunately, it is impossible to observe the solar spectrum as it really is from the Earth, since it is partially modified by the Earth's atmosphere.

Spectrum of Solar Radiation (Earth)

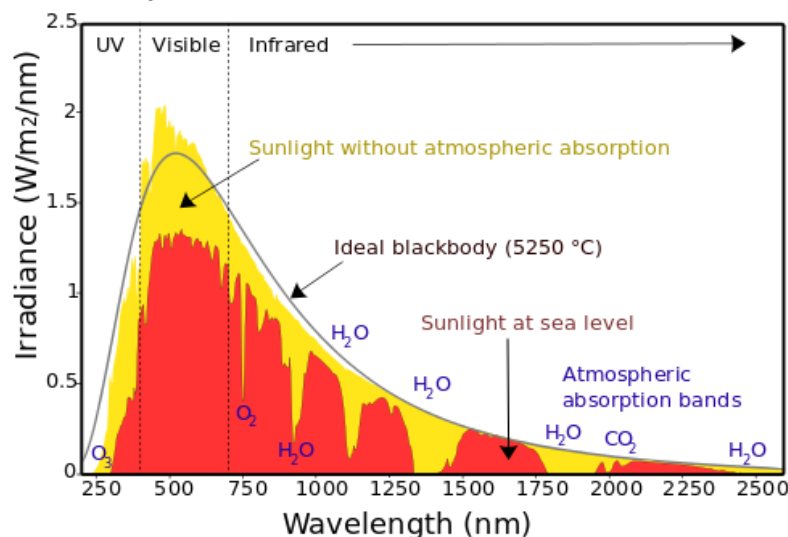


Figure-1.2.1. Spectral distribution of solar radiation energy flux.

First of all, it shortens the length of the lines in the spectrum and forms absorption lines in it. In particular, the effect of the atmosphere is more noticeable on red and yellow rays. In other words, the Earth's atmosphere, due to its optical properties, is a selective light filter, changing the parameters of cosmic solar radiation. If the radiation flux passes through the atmosphere along the normal and reaches a point on

the Earth's surface at sea level, its optical path is taken to be equal to 1 atmospheric mass (AM1).

The optical path length of rays incident at an angle to the Earth's surface is defined relative to the optical length AM1. If the radiation is not attenuated at all in the atmosphere, the optical mass of the atmosphere is zero (AM0).

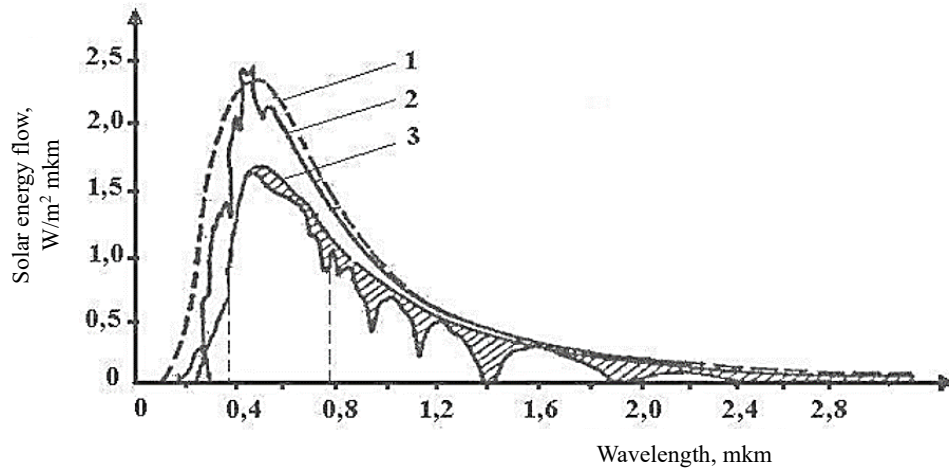


Figure-1.2.2. Spectral distribution of solar radiation energy flux. 1 - the radiation flux density of a black body at a temperature of 6000 K; 2 and 3 - the flux density of solar radiation at the upper and lower limits of the atmosphere (at sea level), respectively.

The fraction of the solar radiation incident on the Earth's surface that is reflected back into the surrounding space is called the albedo of the Earth's surface and is on average 0.34. The spectrum of the reflected radiation is similar to the spectrum of sunlight incident on the Earth's surface when the atmospheric mass is 1.

The power of sunlight falling on a unit area is called its intensity (W/m^2). At the upper limit of the Earth's atmosphere, the intensity of solar energy is equal to $1365 \text{ W}/\text{m}^2$, which is called the solar constant. To study the solar constant and the spectral distribution of solar radiation in depth, studies have been conducted on all continents of the Earth. The main data on this subject were collected by the Smithsonian Institution and the US Air Force Research Laboratory in cooperation with the National Bureau of Standards.

The studies were carried out at different heights above the Earth's surface and using instruments installed on specially launched rockets. The measurements confirm the results of theoretical calculations of the determined value of the solar constant. It was found that the radiation flux density at perihelion (January) can be 3–4 percent higher than the solar constant and 3–4 percent lower at aphelion (July). The density of direct solar radiation at the upper and lower limits of the Earth's atmosphere is presented in Figure 2. For comparison, the radiation flux density of a black body at a temperature of 6000 K is also presented.

Various impurities entering the atmosphere reduce its clarity. The dry air in the layers near the Earth's surface contains 78.03% nitrogen, 20.94% oxygen, 0.94% argon,

0.03% carbon dioxide, 0.01% hydrogen, 0.0012% neon, and 0.0004% helium. At the same time, the atmosphere also contains varying amounts of water vapor, ozone (mainly in the upper layers), and other gases. Sunlight is mainly absorbed by water vapor, ozone, carbon dioxide, and dust. Figure-1.2.3 shows the absorption curve of sunlight passing through the atmosphere by water vapor. Ozone in the atmosphere (stratosphere) plays an extremely important role by absorbing the short-wave part of sunlight that has the property of destroying living cells in the body. Ozone absorbs approximately 1.5 to 3% of solar radiation, depending on its amount in the atmosphere and the altitude of the Sun. An increase in dust in the atmosphere leads to greater scattering and absorption of solar radiation and, ultimately, to a decrease in its flow to the Earth's surface.

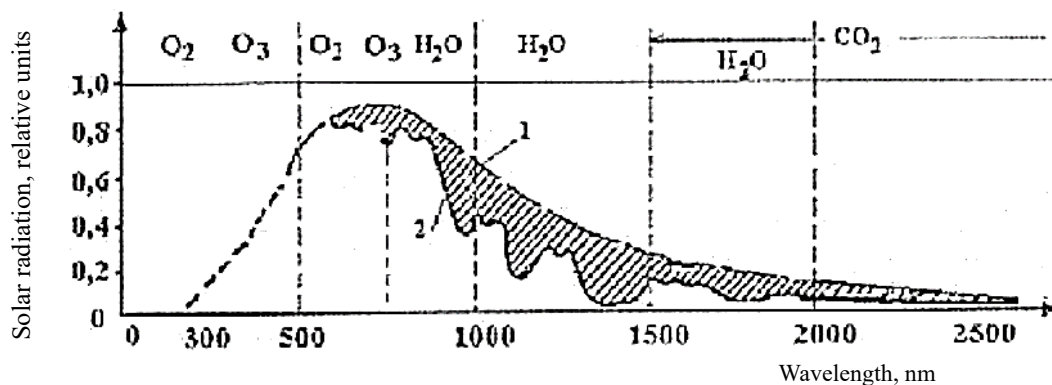


Figure-1.2.3. Dependence of the absorption of sunlight by water vapor on wavelength: 1 and 2—distribution of solar radiation in dry air and water vapor (the hatched part indicates the degree of absorption of sunlight by water vapor).

Transparent glass and polymer films used in solar installations transmit light with wavelengths from 300 nm to 2800 nm. Studies have shown that practically all the energy of the scattered and reflected solar radiation flux falls into the short wavelength range (200–5000 nm), and most of it falls into the visible and near-infrared parts of the spectrum. Thus, solar installations receive energy mainly from short-wavelength sunlight.

1.3. Sources of solar radiation and its characteristics. Basic concepts and quantities

Solar radiation, often referred to as solar energy or simply sunlight, is a general term for the electromagnetic radiation emitted by the sun. Solar radiation can be captured and converted into useful forms of energy, such as heat and electricity, using various technologies. However, the technical feasibility and economic viability of these technologies at a given location depend on the available solar energy.

Every place on Earth receives sunlight for at least part of the year. The amount of solar radiation reaching any point on the Earth's surface varies depending on:

- Geographic location;

- Time of day;
- Season;
- Local landscape;
- Local weather.

Because the Earth is round, the sun hits the Earth's surface at different angles, from 0° (just above the horizon) to 90° (directly overhead). When the sun's rays are vertical, the Earth's surface receives all the energy it can. The more oblique the sun's rays are, the further they are scattered through the atmosphere, becoming more diffuse and scattered. Because the Earth is round, the cold polar regions never get high sunlight, and because the axis of rotation is tilted, these areas receive no sunlight at all for part of the year.

The Earth orbits the Sun in an elliptical orbit, and is closer to the Sun for part of the year. When the Sun is closer to the Earth, the Earth's surface receives slightly more solar energy. When it is summer in the Southern Hemisphere and winter in the Northern Hemisphere, the Earth is closer to the Sun. However, the presence of vast oceans moderates the hot summers and colder winters that can be expected to occur in the Southern Hemisphere as a result of this difference. The tilt of the Earth's axis of rotation, at 23.5° , is a more important factor in determining the amount of sunlight that reaches Earth at a given location. The tilt causes the days to be longer in the Northern Hemisphere from the vernal (spring) equinox to the autumnal (fall) equinox, and longer in the Southern Hemisphere for the remaining 6 months. On the equinoxes, which occur on March 23 and September 22 each year, the days and nights are exactly 12 hours long.

Countries located at mid-latitudes, such as the United States, receive more solar energy in the summer, not only because the days are longer, but also because the sun is almost overhead. During the short days of the winter months, the sun's rays are more oblique. Cities such as Denver, Colorado (near latitude 40°) receive almost three times as much solar energy in June as in December.

The rotation of the Earth is also responsible for the hourly variation in sunlight. In the early morning and late afternoon, the sun is lower in the sky. Its rays travel further through the atmosphere than at noon, when the sun is at its highest point. On clear days, the greatest amount of solar energy reaches the solar collector at noon.

Diffuse and direct sunlight

When sunlight passes through the atmosphere, some of it is absorbed, scattered, and reflected:

- Air molecules;
- Water vapor;
- Clouds;
- Dust;

- Pollutants;
- Forest fires;
- Volcanoes.

This is called diffuse solar radiation. Solar radiation that reaches the Earth's surface without being scattered is called direct solar radiation. The sum of diffuse and direct solar radiation is called global solar radiation. Atmospheric conditions can reduce direct radiation by 10% on clear, dry days and by 100% on thick, cloudy days.

Figure 1.3.1 shows the geometry of the Sun-Earth relationship schematically. The eccentricity of the Earth's orbit is such that the distance between the Sun and the Earth varies by 1.7%. At a distance of one astronomical unit, 1.495×10^{11} m, the average distance between the Earth and the Sun, the Sun is at an angle of $32'$. The radiation emitted by the Sun and its spatial relationship with the Earth result in an almost constant intensity of solar radiation outside the Earth's atmosphere. The solar constant G_{sc} is the solar energy received per unit area of a surface perpendicular to the direction of radiation propagation outside the atmosphere at the average Earth-Sun distance. Before the advent of rockets and spacecraft, it was necessary to calculate the solar constant by measuring solar radiation from the ground, after solar radiation had been transmitted through the atmosphere and thus partially absorbed and scattered by the constituents of the atmosphere.

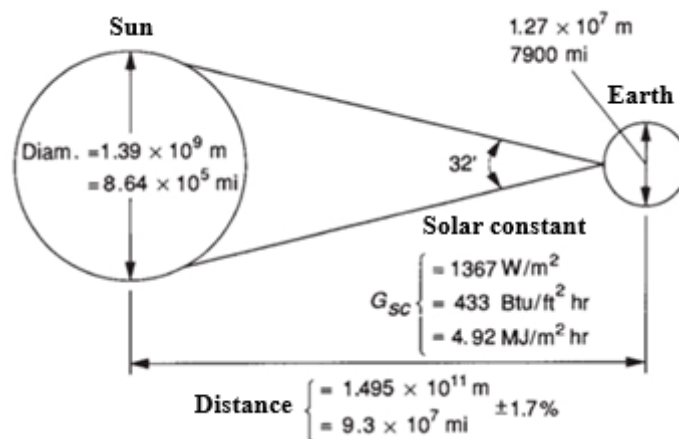


Figure-1.3.1. Sun-Earth connection.

Extrapolations from ground-based measurements from high mountains were based on estimates of atmospheric transmittance in various parts of the solar spectrum. Pioneering studies were carried out by C.G. Abbott and his colleagues at the Smithsonian Institution. These studies and subsequent rocket measurements were summarized by Johnson (1954); Abbott's value of the solar constant, 1322 W/m^2 , was revised upwards by Johnson to 1395 W/m^2 . The availability of very high-altitude aircraft, balloons, and spacecraft has made it possible to directly measure solar radiation outside most or all of the Earth's atmosphere. These measurements were made using a variety of instruments in nine separate experimental programs. They resulted

in a solar constant G_{sc} of 1353 W/m^2 with an estimated error of $\pm 1.5\%$. This standard value was adopted by NASA (1971) and the American Society for Testing and Materials (2006). The data based on the 1353-W/m^2 value were revised by Frohlich (1977) and reduced to a new pyrheliometric scale based on comparisons of instruments with absolute radiometers. Data from the Nimbus and Mariner satellites were also included in the analysis, and from 1978 Frohlich recommended a new value for the solar constant G_{sc} of 1373 W/m^2 , with an error of 1 to 2%. This was 1.5% higher than the previous value and 1.2% higher than the best determination of the solar constant by combining spectral measurements. Additional space-based measurements were made, with Hickey et al. (1982) reporting 1373 W/m^2 and Wilson et al. (1981) reporting 1368 W/m^2 . Three rocket flight measurements reported by Duncan et al. (1982) were 1367, 1372, and 1374 W/m^2 . The World Radiation Center (WRC) accepted a value of 1367 W/m^2 , with an uncertainty of the order of 1%. From these, it is clear that the uncertainties in solar radiation measurements over most land areas are larger than those in G_{sc} .

1.4. Types of solar radiation and their measurement

When using solar energy, it is of great practical importance to know how much time a selected place is illuminated during the year, how much energy falls on a unit of surface per unit of time, or in other words, the radiation regime of the place.

The branch of meteorology that deals with measuring the flow of radiant energy (radiation) and studying the radiation regime of surfaces that receive light is called actinometry. In actinometry, there are concepts of direct, diffuse and total radiation of the Sun. The flow of mutually parallel rays of the Sun directed to the Earth's surface is called its direct (S), the flow scattered from the atmospheric layer is called diffuse (D), and the total flow reaching the Earth's surface is called total radiation (Q).

The amount of solar energy is determined by its intensity. The power of sunlight falling on a unit surface is called its intensity (W/m^2). At the upper limit of the Earth's atmosphere, the intensity of solar energy is equal to 1365 W/m^2 , which is called the solar constant.

Part of the sunlight falling on the Earth or on the surface of objects is reflected back. The ratio of the radiation flux reflected from the surface to the flux incident on it, expressed in percentage, is called the *albedo* of this surface. For example, the albedo is 0.5% for black velvet, 15–35% for dry sand, 75% for white tile, 85–88% for mirror, 85–90% for aluminum, and 50–60% for steel.

There are calorimetric, photoelectric, photographic and visual methods for measuring solar radiation energy.

Calorimetric measurements use the direct conversion of light energy absorbed by a black surface into thermal energy. In this case, the amount of solar radiation flux

falling on the receiving surface of the device is determined by recording the amount of heat released on this surface or the increase in temperature.

The phenomena of the photoelectric effect and the photochemical action of light were used to develop photoelectric and photographic methods for measuring radiation energy. Currently, the photoelectric method of recording using photocells, photomultipliers and photoresistors is widely used in the study of short-wave spectral currents. In visual measurements, the human eye, which is sensitive to the effects of the visible region of the spectrum, serves to assess radiation.

The main task of actinometric measurements is to determine the integral flux (intensity) of short- and long-wave radiation. For this purpose, the calorimetric method is more suitable than all other methods. The heat received by water flowing through the calorimeter serves as a measure of the flux of radiant energy. Water pyrheliometers of various designs designed to measure direct solar radiation operate on this basis.

It is convenient to use actinometric instruments to directly determine the heat flux of solar radiation. These instruments, which are based on determining the heat flux by measuring the temperature difference between the light-receiving surface and the environment, have a wide range. This difference is determined by the thermoelectric method, depending on the amount of current that appears in the circuit formed by the serial connection of thermocouples, the hot junction of which is attached to the light-receiving surface, and the cold junction of which is maintained at a constant temperature. Such instruments are considered relative instruments and require calibration, meaning their readings must be compared with those of absolute instruments.

An absolute method of measuring radiant energy has been developed using the compensation method, the essence of which is to heat the cold junction temperature to the same temperature as the hot junction. The amount of heat expended to heat the cold junction is an absolute measure of the heat absorbed by the incident surface of the device. Based on this principle, Angstrom's absolute pyrheliometer was constructed, which serves to measure the absolute amount of direct solar radiation flux. This instrument is the standard for actinometric instruments.

Currently, the Savinov-Yanishevsky thermoelectric actinometer is mainly used to measure the direct solar radiation flux (Fig. 1.4.1). The main part of the instrument is a tube 5 mounted on a tripod 9 with 1 screw and a thermopile light absorber placed in a box 7 at its end. The horizontal axis of the tube is tilted in the meridian plane in accordance with the geographical latitude of the location. Using screws 3 and 4, the tube is directed towards the Sun so that the sunlight passing through 2 small holes in the outer diaphragm falls on the point 6 opposite this hole, so that the direct solar radiation falls exactly perpendicular to the surface of the thermopile. A GSA-1 galvanometer is connected to the thermopile. When the instrument is not in use, the

tube is closed with a cover 10 to protect the disk on which sunlight falls from dust. Inside the tube are 5 diaphragms that protect the light-receiving surface of the instrument from the effects of wind and scattered and reflected light. The hot junctions of the manganin-constantan thermopile, known as a thermostar, are attached to the back of a thin silver disk with a blackened light-receiving surface, and the cold junctions are attached to a copper ring clamped to the actinometer case, insulated with cigarette paper.

When solar radiation falls on the surface of the disk, the hot junctions heat up, a thermocurrent is generated in the thermopile, which is measured by a galvanometer. The temperature difference between the hot and cold junctions depends on the incident radiation flux, and the thermocurrent generated, in turn, depends on the temperature difference. To find the radiation intensity, the number shown by the galvanometer must be multiplied by the transmission coefficient, which is known by comparing it with the Angstrom pyrheliometer. Typically, the sensitivity of the actinometer is 4–7 millivolts per 696 W/m^2 .

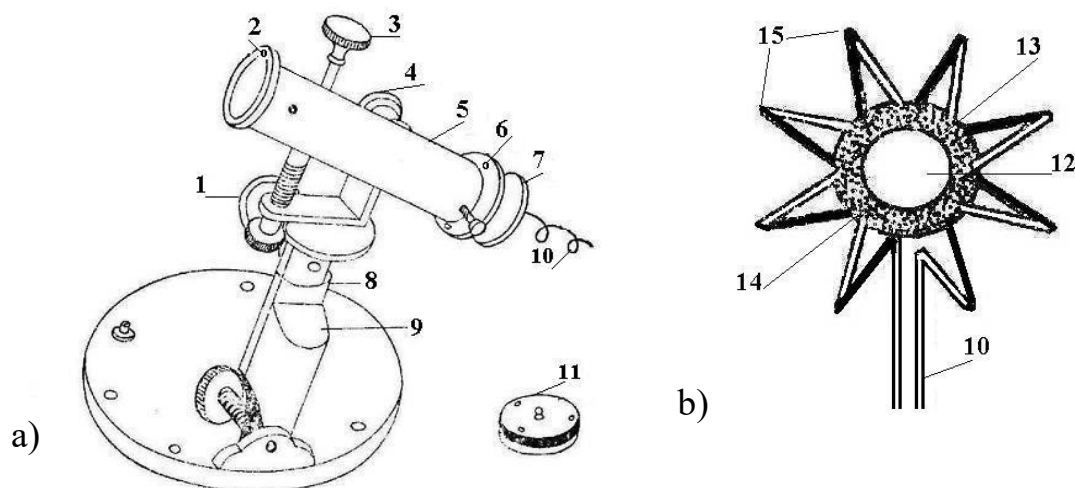


Figure-1.4.1. (a) – General view of the Savinov-Yanishevsky actinometer; (b) – scheme of the actinometer thermopile (thermostar). 1, 3 and 4—screws; 2— hole for light transmission; 5—tube; 6—spot of light passing through 2 holes; 7—box with thermopile; 8—scale; 9—stand; 10—conductors for connecting to the galvanometer; 11— cap; 12—silver disk; 13—copper ring; 14 and 15—hot and cold junctions of the thermocouple, respectively.

To record the collected and scattered radiation, the Yanishevsky thermoelectric pyranometer is widely used (Fig. 1.4.2). The main part of the instrument is a thermoelectric battery 2 mounted on a metal base 1, which consists of a large number of manganin and constantan particles, often arranged in a checkerboard pattern. All even joints are whitened with magnesium, and odd joints are blackened with soot. A glass cover 3 protects the thermoelectric battery from dust, wind, and long-wave radiation. To measure scattered radiation, the light-receiving part of the instrument is

shaded by a blackened metal screen 4. For this, an umbrella screen with a diameter equal to the diameter of the glass cover is attached to the upper end of a rod 6 mounted on a metal base (usually the length of the rod is 5–7 times greater than the diameter of the screen). In both cases, the thermocurrent generated by the temperature difference between the black and white thermocouples is measured using a GSA-1 galvanometer. The galvanometer needle deviates in proportion to the intensity of solar radiation. The radiation intensity is determined using special tables in accordance with the galvanometer readings. On cloudy days, the total radiation is equal to the scattered radiation.

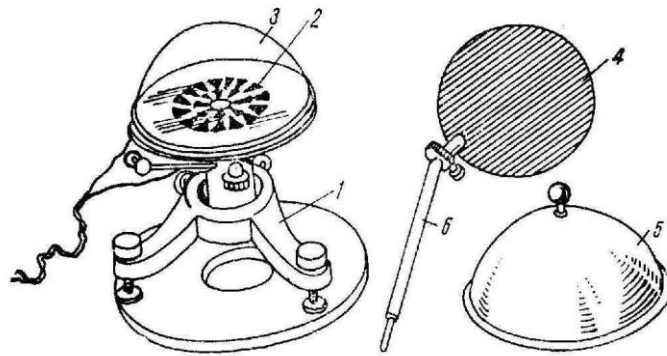


Figure-1.4.2. Yanishevsky's thermoelectric pyranometer. 1–metal base; 2–thermoelectric battery; 3–glass cover; 4–umbrella screen; 5–metal cover; 6–screw.

In order for the actinometer to record accurate solar radiation quickly and continuously, it is mounted on a heliostat with a clock mechanism that automatically adjusts to the Sun (Figure 1.4.3). The current generated in the actinometer circuit is automatically recorded using an electronic potentiometer.

The heliograph automatically records the time when the sun is above the horizon and not blocked by clouds (Figure 1.4.3). The operation of the instrument is based on the thermal or chemical effects of sunlight. Its main part is made up of two heavy glass spheres that act as collecting lenses. The spherical lens is mounted on a base with a device for changing the angle of deviation of its axis and a special holder for placing a paper strip divided into hours. The incident sunlight is collected at the focus of the glass sphere and burns the paper strip. The length of the burn on the paper strip indicates the time when the sun is shining. It is worth noting that the ultraviolet rays of the Sun reach the Earth's surface in a small amount (about 3%) mixed with the visible and infrared regions of the spectrum. Therefore, before measuring it, it must be separated, in other words filtered.

For this purpose, ultraviolet light filters are used, for example, glass stained with nickel and cobalt salts, which is called Wood's black glass. Such glasses do not transmit visible light.

One of the devices used to block direct ultraviolet radiation with a wavelength of 300 nm to 400 nm falling on surfaces perpendicular to the rays is the quartz solar monochromator of A.N. Boyko. This device is compact and convenient for conducting observations at actinometric and biological stations, as well as in field conditions. Using a monochromator, it is possible to measure the intensity of direct ultraviolet radiation in 15 parts of the solar spectrum for 2–3 minutes.

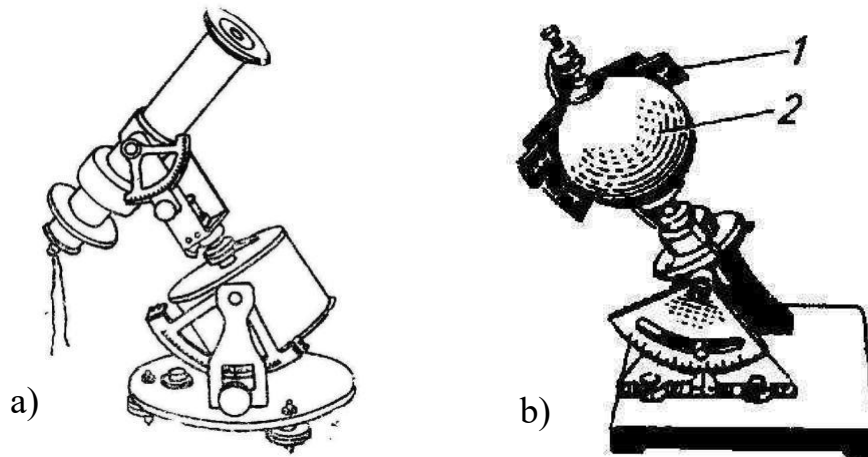


Figure-1.4.3. (a) Heliostat for actinometer; (b) General view of heliograph: 1– paper strip divided into hours; 2–spherical lens.

The optical concentration of ultraviolet rays with a wavelength between 288 nm and 400 nm in a monochromator thermopile is 1000 times higher than that of natural rays. A needle galvanometer with a sensitivity of about 10^{-7} Å serves as a measuring instrument.

1.5. Daily total indicators of direct, diffuse and total solar radiation falling on horizontal and inclined surfaces.

Radiation data are the best source of information for calculating average incident radiation. In the absence of these or similar data from nearby locations with similar climates, empirical relationships can be used to calculate radiation from sunshine or cloud hours. Data on average hourly sunshine or average percentage of probable sunshine hours are widely available from many hundreds of stations in many countries, usually based on data obtained with Campbell-Stokes instruments. Examples are shown in Table 2. Cloud cover data (i.e. cloudiness), although also widely available, are based on visual estimates and are probably less useful than sunshine hours.

The original Angstrom-type regression equation relates the monthly average daily radiation to the filtered radiation at the location in question and the average fraction of possible sunshine hours:

$$\frac{\bar{H}}{\bar{H}_c} = a' + b' \frac{\bar{n}}{\bar{N}} \quad (1.5.1)$$

where, $\bar{H}$ = Monthly average daily radiation on the horizontal surface; H_c = Average clear sky daily radiation for the requested location and month; a' , b' = empirical constants; $\bar{n}$ = monthly average daylight hours bright sunlight; $\bar{N}$ = monthly average light is the maximum daily hours of sunlight (i.e., the length of the average day of the month).

Table-1.5.1

Examples of monthly average hours of sunshine by latitude and altitude

Location Latitude Height, m	Hong kong, 22° E, Sea level	Paris, France, 48° N, m	Mumbai, India, 19° E, Sea level	Sokoto, Nigeria, 13° E, 107 m	Pert, Australia, 32° S, 20 m	Madison, Wisconsin, 43° E, 270 m
January	4.7	2.1	9.0	9.9	10.4	4.5
February	3.5	2.8	9.3	9.6	9.8	5.7
March	3.1	4.9	9.0	8.8	8.8	6.9
April	3.8	7.4	9.1	8.9	7.5	7.5
May	5.0	7.1	9.3	8.4	4.7	9.1
June	5.3	7.6	5.0	9.5	4.8	10.1
July	6.7	8.0	3.1	7.0	5.4	9.8
August	6.4	6.8	2.5	6.0	6.0	10.0
September	6.6	5.6	5.4	7.9	7.2	8.6
October	6.8	4.5	7.7	9.6	8.1	7.2
November	6.4	2.3	9.7	10.0	9.6	4.2
December	5.6	1.6	9.6	9.8	10.4	3.9
Annual	5.3	5.1	7.4	8.8	7.8	7.3

The main difficulty with equation 1.5.1 lies in the uncertainty of the terms $\bar{n} / \bar{N}$ and H_c . The former is an instrumental problem (the records of the sunlight recorders are open to interpretation). The latter arises from the uncertainty in the definition of a clear day. Page (1964) and others modified it to base it not on the apparent irradiance but on the surface irradiance at a horizontal surface:

$$\frac{\bar{H}}{\bar{H}_o} = a + b \frac{\bar{n}}{\bar{N}} \quad (1.5.2)$$

where, $\bar{H}_o$ - is the radiation at the Earth's surface for a location averaged over the time period in question, where a and b are constant depending on location. $\frac{\bar{H}}{\bar{H}_o}$ - is called the monthly average accuracy index.

Example.

Calculate the monthly averages of solar radiation on a horizontal surface for Madison, Wisconsin, latitude 43°, based on the average duration of the sunshine hour data in Table 1.5.2.

Solution.

The calculations are based on equation 2.7.2 using the constants $a = 0.30$ and $b = 0.34$ from Table 2.7.2. The values of H_o are obtained from the table or from Figure 1.10.1 and equation 1.5.3 from the day lengths, each for a mean day of the month. The desired estimates are obtained from the following table, which shows daily H in MJ/m². (For comparison, TMY3 data for Madison are shown and were determined using the constants a and b for Blue Hill in the latest Madison radiation calculations.)

$$N = \frac{2}{15} \cos^{-1}(-\tan \phi \tan \varphi) \quad (1.5.3)$$

Table-1.5.2

Month	$\overline{H_o}$ (MJ/m ²)	$\overline{N}$ (h)	$\overline{n} / \overline{N}$	Rated $\overline{H}$ (MJ/m ²)	Measured $\overline{H}^a$ (MJ/m ²)	Rated $\overline{H}^b$ (MJ/m ²)
January	13.36	9.2	0.49	6.3	6.9	6.2
February	18.80	10.3	0.55	9.2	9.7	9.3
March	26.01	11.7	0.59	13.0	13.1	13.4
April	33.75	13.2	0.57	16.6	16.9	17.0
May	39.39	14.5	0.63	20.2	21.0	21.0
June	41.74	15.2	0.67	22.0	23.4	23.1
July	40.52	14.0	0.66	21.2	22.2	22.2
August	35.88	13.8	0.73	19.6	19.6	20.9
September	28.77	12.3	0.70	15.5	14.5	16.4
October	20.89	10.8	0.67	11.0	9.7	11.6
November	14.61	9.5	0.44	6.6	6.2	6.4
December	11.90	8.8	0.44	5.4	5.6	5.3

Although the constants a and b for Madison are taken from a different database than the measured data, the agreement between the measured and calculated radiation is reasonably good.

If we didn't have constants for Madison and had to choose a climate close to Madison, Blue Hill would be a reasonable choice. The averages calculated using the Blue Hill constants are shown in the last column. Trends are shown, but the agreement is not very good.

This is more typical when using equation 1.5.2. The data are also available in C , the average monthly cloud cover, expressed as the fraction of the sky covered by clouds.

Empirical relationships were derived to relate the monthly average daily radiation H to the monthly average cloud cover C .

These are usually of the form

$$\frac{\bar{H}}{H_o} = a'' + b''\bar{C} \quad (1.5.4)$$

Calculating open sky radiation

The effect of the atmosphere on scattering and absorption of radiation varies with time as atmospheric conditions and air masses change. It is useful to define a standard “clear” sky and calculate the hourly and daily radiation received by a horizontal surface under these standard conditions.

Table-1.5.3

Correction factors for climate types

Weather type	r_0	r_l	r_k
Tropical	0.95	0.98	1.02
Midlatitude summer	0.97	0.99	1.02
Subarctic summer	0.99	0.99	1.01
Midlatitude winter	1.03	1.01	1.00

Hottel (1976) presented a method for calculating the specific atmospheric transmittance, taking into account the zenith angle and altitude, for a standard atmosphere and four climate types. The atmospheric transmittance for a radiant intensity τ_b is G_{bn}/G_{on} (or G_{bT}/G_{oT}), and is given by:

$$\tau_b = a_0 + a_1 \exp\left(\frac{-k}{\cos \theta_z}\right) \quad (1.5.5a)$$

For a standard atmosphere with a visibility of 23 km, the constants a_0 , a_1 and k are found from a_0^* , a_1^* and k^* , respectively. These are given for altitudes below 2.5 km.

$$a_0^* = 0.4237 - 0.00821(6 - A)^2 \quad (1.5.5b)$$

$$a_1^* = 0.5055 + 0.00595(6.5 - A)^2 \quad (1.5.5c)$$

$$k^* = 0.2711 - 0.01858(2.5 - A)^2 \quad (1.5.5d)$$

where A is the observer's altitude in kilometers.

Correction factors are applied to a_0^* , a_1^* and k^* to allow for variations in climate types. The correction factors $r_0 = a_0/a_0^*$, $r_1 = a_1/a_1^*$ and $r_k = k/k^*$ are given in table-1.5.3. Thus, this standard atmospheric transmittance for radiant energy can be determined for any zenith angle and any altitude up to 2.5 km. Clear sky radiation is then normal radiation

$$G_{cnb} = G_{on}\tau_b \quad (1.5.6)$$

Clear sky horizontal radiant radiation

$$G_{cb} = G_{on}\tau_b \cos \theta_z \quad (1.5.7)$$

Clear sky horizontal radiation over one hour

$$I_{cb} = I_{on} \tau_b \cos \theta_z \quad (1.5.8)$$

Example: Calculate the transmittance for the radiant radiation of a standard clear atmosphere at Madison (altitude 270 m) on August 22 at 11:30 AM solar time. Calculate the intensity of the radiant radiation and its component on a horizontal surface at that time.

Solution:

On August 22, $n = 234$, the declination is 11.4° , the cosine of the zenith angle is 0.846.

The next step is to find the coefficients of equation 1.5.5. First, the values for the standard atmosphere are obtained from equations 1.5.5b to 1.5.5d for an altitude of 0.27 km:

$$\begin{aligned} a_0^* &= 0.4237 - 0.00821(6 - 0.27)^2 = 0.154 \\ a_1^* &= 0.5055 + 0.00595(6.5 - 0.27)^2 = 0.736 \\ k^* &= 0.2711 - 0.01858(2.5 - 0.27)^2 = 0.363 \end{aligned}$$

The climate type correction factors for midlatitude summer are taken from Table 1.5.3. Equation 1.5.5a becomes

$$\tau_b = 0.154 \times 0.97 + 0.736 \times 0.99 \exp\left(-0.363 \times \frac{1.02}{0.846}\right) = 0.62$$

The radiation at the Earth's surface is 1339 W/m^2 . The radiation after

$$G_{cnb} = 1339 \times 0.62 = 830 \text{ W/m}^2$$

Component in the horizontal plane

$$G_{cb} = 830 \times 0.846 = 702 \text{ W/m}^2$$

It is also necessary to calculate the clear sky diffuse radiation on a horizontal surface to obtain the total radiation. Liu and Jordan (1960) developed an empirical relationship between the transmission coefficients for radiation and the diffuse radiation for clear days:

$$\tau_d = \frac{G_d}{G_o} = 0.271 - 0.294\tau_b \quad (1.5.9)$$

where τ_d is the ratio of diffuse radiation G_d/G_o (I_d/I_o) to ground-level (radiant) radiation in the horizontal plane. The equation is based on data for 3 stations. The data used by Liu and Jordan were used by Hottel (1976) and may not be fully consistent with it; until better data are available, equation 2.8.5 is proposed to be used to estimate diffuse net sky radiation. This can be combined with the Hottel method predicted radiant radiation to obtain a precise hourly total. These calculations can be repeated for each hour of the day, based on the midpoints of the hours, to obtain a standard net daily radiation H_c .

CHAPTER-II. PHYSICAL BASIS OF CONVERSION OF SOLAR ENERGY INTO HEAT

2.1. Heat transfer. Principles of heat transfer. Thermal conductivity. (convective heat transfer)

Thermal conductivity is the molecular transfer of heat in a closed medium due to a temperature difference.

This method of heat transfer occurs mainly within a solid body, as well as between two solid bodies in contact with each other. Heat conduction can also occur through a layer of liquid or gas, but in general, liquids and gases (with the exception of liquefied metals) are very poor conductors of heat.

Let's look at the heating of a homogeneous isotropic body. Bodies that have the same physical properties in all directions are called isotropic bodies. When such a body is heated, the temperature at different points in it changes with time, and heat spreads from the region of higher temperature to the region of lower temperature.

The sum of the temperature values at all points of the space under consideration at the same time is called the temperature field. The temperature field is expressed by the following equation:

$$t=f(x,u,z,\tau) \quad (2.1.1)$$

where, x,u,z – point coordinates; τ - time.

If the temperature of an object is a function of coordinate and time, then the temperature field is nonstationary:

$$t=f(x,u,z,\tau); \partial t/\partial \tau \neq 0 \quad (2.1.2)$$

If the temperature of an object is a function of the coordinate only and does not change over time, then the temperature field is stationary.

$$t=f(x,u,z); \partial t/\partial \tau = 0 \quad (2.1.3)$$

The temperature field can be a function of three, two, and one coordinate and is called three-, two-, and one-dimensional, respectively. A surface at all points of which the temperature is the same is called an isothermal surface.

Since it is impossible to have two different temperatures at the same point in space, different isothermal surfaces never intersect. They all end at the surface of the body or lie entirely within it. The temperature of the body changes only in the directions intersecting the isothermal surfaces (Fig. 2.1.1). In this case, the greatest change in temperature per unit length occurs in the direction n normal to the isothermal surface.

The ratio of the temperature change Δt to the distance Δn along the normal to the isotherm is called the temperature gradient:

$$\lim\left[\frac{\Delta t}{\Delta n}\right]_{\Delta n \rightarrow 0} = \frac{\partial t}{\partial n} = \text{grad} t \quad (2.1.4)$$

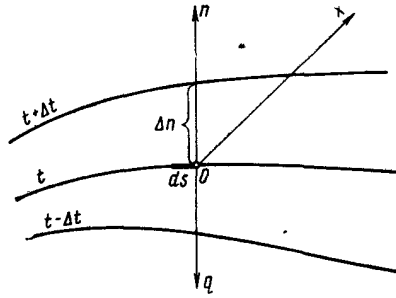


Figure-2.1.1. Isotherms. About the concept of temperature gradient.

The temperature gradient is a vector directed along the normal to an isothermal surface. Its direction in the direction of increasing temperature is considered positive. Like other types of heat transfer, the process of heat conduction occurs only when the temperature at different points of the body is not the same, that is, the gradient $t \neq 0$. The amount of heat passing through an arbitrary surface per unit time is called the heat flux Q . The vector of the heat flux is always directed in the direction of decreasing temperature. The French scientist Fourpe, studying heat conduction processes in solids, determined the relationship between the amount of heat dQ passing through a unit area dF per unit time $d\tau$ and the temperature gradient.

$$dQ = -\chi dF \text{ grad } t \, d\tau = -\chi dF \, d\tau (\partial t / \partial n) \quad (2.1.5)$$

Equation (2.1.5) expresses the basic law of heat transfer and is called Furpe's law. The minus sign in this equation indicates that the heat flow and temperature gradient vectors are directed in opposite directions.

The coefficient of proportionality χ in expression (2.1.5) is called the heat transfer coefficient. The amount of heat that passes through a unit of isothermal surface per unit of time is called the heat flux density.

$$q = -dQ / (dF d\tau) \text{ yoki } q = -\chi (\partial t / \partial n) \quad (2.1.6)$$

The vector of heat flux density q is always directed in the direction of decreasing temperature. The amount of heat passing through an arbitrary surface F per unit time is defined as:

$$Q = \int_0^\tau \int_F \chi dF d\tau (\partial t / \partial n) \quad (2.1.7)$$

The units of the quantities studied above are as follows:

temperature gradient – grad/m;

heat flow – W;

heat flux density – W/m².

The unit of the thermal conductivity coefficient is determined from the expression (8.7):

$$\chi = -\frac{Q}{F(\partial t/\partial n)} = \frac{Vt}{m \cdot \text{grad}} \quad (2.1.8)$$

Thus, the value of the thermal conductivity coefficient is numerically equal to the specific heat flux passing through a unit layer of the wall when the temperature difference is 1°C.

For different substances, χ has a certain value, which depends on the structure, density, pressure and temperature of the substance.

The value of the thermal conductivity coefficient χ is found experimentally for each body. For most materials, the dependence of χ on temperature can be expressed as follows:

$$\chi = \chi_0[1 + b(t - t_0)],$$

where, χ_0 – thermal conductivity coefficient at temperature; t – temperature, °C; b – experimentally determined temperature coefficient.

Metals conduct heat best, with χ_3 to 458 W/(m·degree). The thermal conductivity of pure metals (except aluminum) decreases with increasing temperature. Lightweight porous materials conduct heat poorly, since their pores are filled with air. If $\chi < 0.2$ W/(m·degree), such materials are called thermal insulation materials.

The thermal conductivity of such materials increases with increasing temperature. Humidity has a significant effect on the thermal conductivity. Water has poor thermal conductivity, but the thermal conductivity of a wet material is much higher than its thermal conductivity in a dry state.

This is because water conducts heat almost 20 times better than air, so filling the pores of the body with water dramatically reduces its thermal insulation properties.

As the temperature increases, the thermal conductivity of liquid droplets decreases, while that of gases increases. The χ of water increases as the temperature increases from 0°C to 127°C, and then decreases as the temperature increases. Table 2.1.3 lists the thermal and thermal conductivity coefficients of some materials.

Table-2.1.3

Thermal and temperature conductivity coefficients of some materials

Name of materials	ρ , kg/m ³	t , °C	λ , Wt/(m·grad)	S kJ/(m·grad)	$Q \cdot 10^6$ m ² /sec.
Asbestos	770	30	0,11163	0,816	0,186
Concrete	2300	20	0,279	1,13	0,622

Wet soil	1700	17	0,657	2,01	0,192
Baked brick	1800	0	0,768	0,879	-
Ice	920	0	2,25	2,26	1,08
Dry sand	1500	20	0,326	0,795	2,74
Glass	2500	20	0,744	0,67	0,444
Aluminum	2670	0	204	0,921	86,7
Copper	8800	0	384	0,381	112,5
Nickel	9000	20	58	0,461	17,8
Silver	10500	0	458	0,234	170
Carbon steel	7900	20	45	0,461	14,7
Water	999,9	0	0,5513	4,212	0,131
Air (dry)	1,293	0	0,0244	1,005	18,8
Oxygen	1,429	0	0,0247	0,915	18,8

Differential equation of heat transfer. Boundary conditions

To determine the amount of heat passing through an isothermal surface dF in time $d\tau$, equation (2.1.5) must be integrated with respect to F and τ , that is, the temperature field inside the body must be known. To solve this problem, a differential equation of heat conduction is derived.

The following conditions are assumed when deriving the equation: the body is homogeneous and isotropic; its physical parameters do not change. According to the law of conservation of energy, the sum of the amount of heat dQ_1 supplied to the elementary volume of the body by heat conduction from the outside and the amount of heat released by the internal heat source dQ_2 during a time τ must be equal to the change in the internal energy of the body $dQ=dU$:

$$dQ_1+dQ_2=dQ \quad (2.1.9)$$

To determine the terms of this equation in the Cartesian coordinate system, we can isolate a parallelepiped with sides dx , dy , and dz in the solid (Figure 2.1.2).

Where, dQ_x , dQ_y , dQ_z – heat being brought. dQ_{x+dx} , dQ_{y+dy} , dQ_{z+dz} – heat being transported. Then, according to Fourier's law (2.1.5) for the edge $dy \cdot dz$:

$$\begin{aligned} dQ_x &= -\chi \frac{\partial t}{\partial x} dydzd\tau; \quad dQ_{x+dx} = -\chi \left[\left(\frac{\partial t}{\partial x} + \frac{\partial}{\partial x} \left(\frac{\partial t}{\partial x} \right) dx \right) dydzd\tau = \right. \\ &= \left. -\chi \left(\frac{\partial t}{\partial x} + \frac{\partial^2 t}{\partial x^2} dx \right) dydzd\tau. \right. \end{aligned}$$

The difference between these quantities gives the amount of heat remaining in the parallelepiped:

$$dQ_{x1} = dQ_x - dQ_{x+dx} = \chi \frac{\partial^2 t}{\partial x^2} dx dy dz d\tau$$

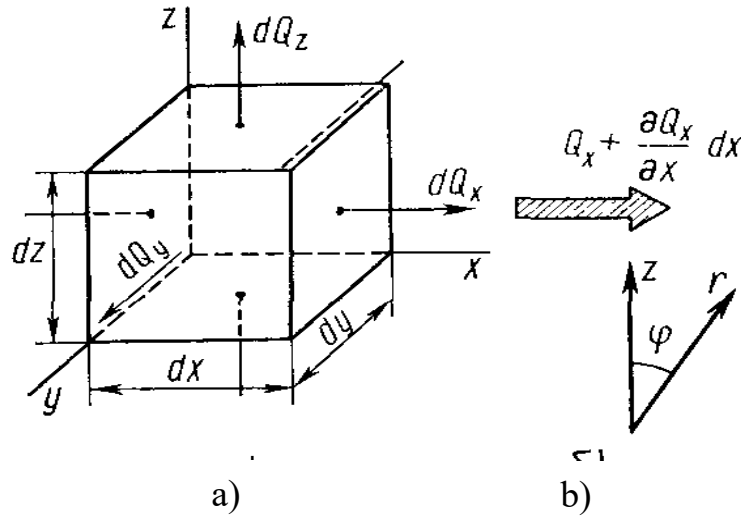


Figure-2.1.2. Differential equation for heat transfer. In Cartesian (a) and cylindrical (b) coordinates.

A similar division can be made for the other two edges. In this case, the total amount of heat supplied to and retained by the body is equal to:

$$dQ_1 = \lambda \left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} \right) dx dy dz dr$$

If we define the specific heat capacity of an internal heat source as q_v (J/m³):

$$dQ_2 = q_v dx \cdot dy \cdot dz \cdot d\tau$$

The change in the internal energy of a body during time $d\tau$

$$dQ = c_\rho \frac{\partial t}{\partial \tau} dx dy dz d\tau$$

Substituting dQ_1 , dQ_2 , and dQ into equation (2.1.9) and making some modifications, we obtain the following:

$$c_\rho \frac{\partial t}{\partial \tau} = \lambda \left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} \right) + q_v$$

or

$$\frac{\partial t}{\partial \tau} = \frac{\lambda}{c_\rho} \left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} \right) + \frac{q_v}{c_\rho} = a \nabla^2 t + \frac{q_v}{c_\rho} \quad (2.1.10)$$

Where, $a = \frac{\lambda}{c_\rho}$ - thermal conductivity coefficient;

$$\nabla^2 = \left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2} \right) - \text{Laplace operator.}$$

In this case, the differential equation for heat transfer can be expressed as:

$$\frac{\partial t}{\partial \tau} = a \nabla^2 t + \frac{q_v}{c\rho} \quad (2.1.11)$$

In the cylindrical coordinate system (Figure-2.1.2, b), equation (2.1.11) takes the following form.

$$\frac{\partial t}{\partial \tau} = a \left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r} \frac{\partial t}{\partial r} + \frac{1}{r^2} \frac{\partial^2 t}{\partial \varphi^2} + \frac{\partial^2 t}{\partial z^2} \right) + \frac{q_v}{c\rho} \quad (2.1.12)$$

where, r – radius; φ - angle.

For stationary conditions $\partial t / \partial \tau = 0$, then equation (2.1.11) can be written as follows:

$$Q \nabla^2 + q_v / (c\rho) = 0; \quad \text{or} \quad \nabla^2 t + q_v / x = 0 \quad (2.1.13)$$

Without an internal heat source:

In stationary condition

$$\nabla^2 t = 0 \quad (2.1.14)$$

In a non-stationary state

$$\partial t / \partial \tau = a \nabla^2 t \quad (2.1.15)$$

Equations (2.1.11) and (2.1.12) define the relationship between the time and spatial variations of temperature at any point in a body where the heat transfer process occurs.

The differential equation of heat transfer (Fourier equation) describes heat transfer by the heat transfer method in the most general way.

When applying this equation to specific cases, it is necessary to know the temperature distribution in the body at the initial moment of time and the initial conditions.

In addition, the following must be known: the geometric shape and size of the body, the physical parameters of the environment and the body, and the boundary conditions that determine the temperature distribution on the surface of the body.

All of the above properties, together with the differential equation, fully describe a specific heat transfer process and are called uniformity conditions or boundary conditions. Usually, the initial temperature distribution is given for time $\tau = 0$.

Boundary conditions can be given in three different ways. In the first type of boundary conditions, the temperature distribution on the surface of the body is given for any given time.

In the second type of boundary conditions, the heat flux density at any point on the surface of the body is given for any given time.

In the third type of boundary conditions, the temperature of the environment surrounding the body and the laws of heat transfer between the surface of the body and the environment are given. Solving the differential equation of heat transfer based on the uniformity conditions allows us to determine the temperature field over the entire volume of the body at any given time.

2.2. Convective heat transfer

The process of heat transfer when gas or liquid macroparticles move from one place to another is called convection. Convection (from Latin *canvectia* - to bring) is manifested in the irregular movement of particles of layers of dispersed, liquid and gaseous substances. Therefore, convection can occur only in a medium whose particles move easily. Heat exchange due to the combined action of convective and molecular heat transfer is called convective heat exchange. In other words, convective heat exchange is carried out simultaneously by two methods: convection and heat conduction. Convective heat exchange between a moving medium and its boundary surface with another (solid, liquid or gas) is called heat transfer.

The main task of the theory of convective heat transfer is to determine the amount of heat transferred through a solid body through which a current flows. The final flow of heat is always directed in the direction of decreasing temperature.

In practical calculations of heat transfer, Newton's law is used.

$$Q = \alpha F(t_c - t_{deg}) \cdot \tau \quad (2.2.1)$$

This equation was derived by I. Newton in 1701 and is called Newton's law of convective heat transfer. According to this law, the amount of heat transferred from a fluid to a wall or from a wall to a fluid Q is proportional to the surface area F involved in the heat transfer, the temperature drop $t_s - t_{dev}$, and the heat transfer time τ .

Where, t_{dev} – wall surface temperature; t_s – Temperature of the medium washing the wall surface.

The coefficient of proportionality α , which takes into account the specific conditions of heat exchange between a liquid and a solid, is called the heat transfer coefficient.

Assuming that $F = 1 \text{ m}^2$ and $\tau = 1 \text{ sec}$ in formula (2.2.1), we obtain the heat flux density in Watts passing through a surface of one square meter:

$$q = \alpha(t_c - t_{\text{des}}) \quad 2.2.2$$

or

$$q = \frac{t_c - t_{\text{des}}}{1/\alpha} \quad 2.2.3$$

The inverse of the heat transfer coefficient, $1/\alpha$, is called the thermal resistance of heat transfer.

If we solve equation (2.2.3) with respect to α , we get:

$$\alpha = \frac{q}{t_c - t_{\text{des}}} \quad (2.2.4)$$

According to equation (2.2.4), the heat transfer coefficient α is the ratio of the heat flux density q to the difference between the temperature of the surface of the body and the temperature of the surrounding environment.

When the temperature difference is equal to 1°C , the heat transfer coefficient α is numerically equal to the heat flux density.

Convective heat transfer is a rather complex process. The main issue in calculating this process is to determine the heat transfer coefficient α . The heat transfer coefficient α depends on many factors, the main ones being:

Causes of fluid flow

Depending on the causes of its occurrence, the movement of liquids is divided into free and forced movement. Free movement or natural convection occurs in an unevenly heated liquid (gas).

The temperatures that arise in this case lead to a difference in density and the emergence of macroparticles with a lower density in the liquid to the surface of the liquid, which causes movement. The intensity of free movement depends on the type of liquid, the difference in the temperatures of its macroparticles, and the volume in which the process takes place. Forced fluid movement or forced convection is associated with the action of external actuators: fans, pumps, etc.

With their help, it is possible to change the speed of fluid movement over a wide range and thereby control the rate of heat transfer.

Fluid flow pattern

In 1884, O. Reynolds, based on his experiments, showed that the motion of a fluid can be laminar or turbulent.

In laminar flow, the particles of the fluid move without mixing. In this case, heat transfer normal to the direction of flow occurs mainly by thermal conductivity. Since

the thermal conductivity of the fluid is quite small (for water $\chi=0.60$ W/(m.K)), the rate of heat exchange in laminar flow is not high.

As the flow velocity increases above a certain value, the flow characteristics change dramatically. In this case, the straight thread-like shape of the flow changes, becoming wavy and completely mixed.

The movement of the fluid becomes irregular, and the flow is constantly mixed. Such a flow is called turbulent flow.

In turbulent flow, heat is distributed in the flow by thermal conductivity, as well as by mixing almost the entire mass of the fluid. Therefore, heat transfer in turbulent flow is much greater than in laminar flow.

Reynolds showed that the flow behavior of a fluid in a pipe is determined by the value of the dimensionless complex $w d / \nu$. This complex is called the Reynolds number:

$$Re = w d / \nu, \quad (2.2.5)$$

where, w – average velocity of the fluid, m/sec; d – pipe diameter, m; ν – coefficient of kinematic viscosity, m^2/sec .

Using formula (9.5), the Reynolds number can be calculated for the flow in any cross section.

It has been experimentally determined that the critical value of the Reynolds number is $Re_{kr}=2300$. When $Re \leq 2300$, the flow is laminar, and when $Re \geq 10000$, it is turbulent.

The movement of a fluid in pipes has its own characteristics. Let us consider the movement of a fluid with a velocity $w=\text{const}$ along a pipe (Fig. 2.2.1).

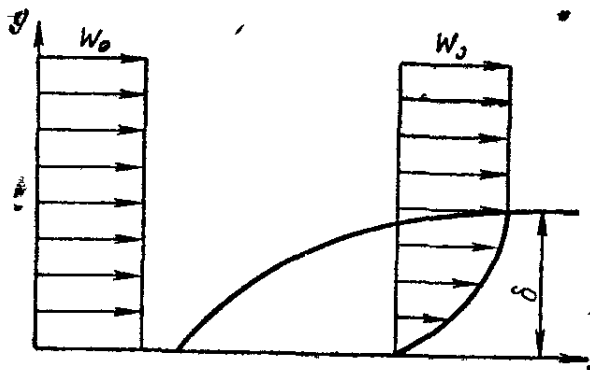


Figure-2.2.1

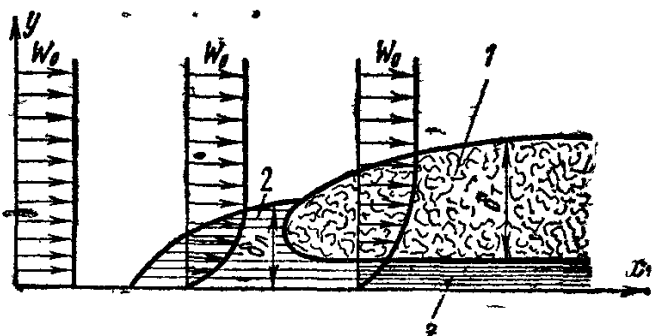


Figure-2.2.2

As the fluid begins to flow through the pipe, friction causes the fluid particles near the walls to stick to the walls, resulting in a velocity near the walls that drops to zero. Since the fluid flow rate remains unchanged, the velocity increases correspondingly in the middle of the pipe section. In this case, a hydrodynamic boundary layer is formed on the pipe walls - a layer in which the fluid velocity w decreases to zero. The thickness of this layer increases along the flow (Figure 2.2.1).

As the flow velocity increases, the thickness of the boundary layer decreases, while as the fluid viscosity increases, the layer thickness increases. In a hydrodynamic boundary layer, the flow can be laminar 1 and turbulent 2. (Figure 2.2.2). The type of flow is determined by the Reynolds number. If the wall and fluid temperatures are not the same, then a thermal boundary layer is formed near the wall, and all temperature changes in the fluid occur in this layer (Figure 2.2.3). Outside this boundary layer, the temperature of the fluid remains constant. In general, the thickness of the thermal and hydrodynamic layers may not coincide (Figure 2.2.4).

The ratio of the thicknesses of these layers is determined by the dimensionless number $Rr=w/a$. For viscous fluids with low thermal conductivity (for example, oils), Rr is 1 and the thickness of the hydrodynamic layer is greater than the thickness of the thermal boundary layer. For gases, $Rr \approx 1$, in which these layers have almost the same thickness.

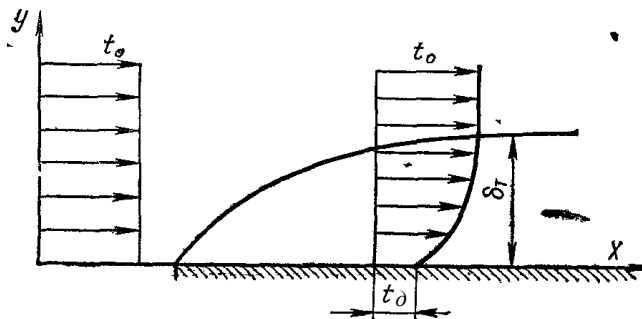


Figure-2.2.3

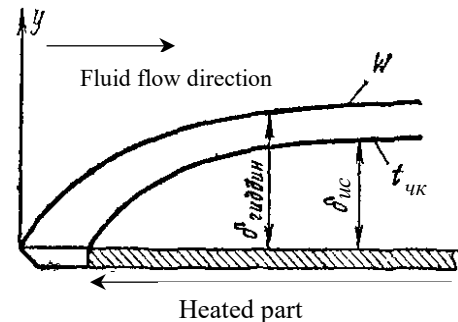


Figure-2.2.4

The mechanism and rate of heat transfer depend on the characteristics of the fluid motion in the boundary layer. If the heat motion within the boundary layer is laminar, then in the direction perpendicular to the wall, heat is transferred by thermal conduction. However, at the outer boundary of the layer, heat is transferred mainly by convection. If the flow in the boundary layer is turbulent, heat is transferred mainly towards the wall due to turbulent mixing of the fluid. This heat transfer is much more rapid than heat transfer by thermal conduction. However, in the laminar layer immediately in front of the wall, heat is transferred to the wall by thermal conduction.

Physical properties of liquid

The following physical properties of the fluid affect the heat transfer process: thermal conductivity coefficient χ , specific heat capacity s , density ρ , thermal conductivity coefficient α , and viscosity coefficient μ . These parameters have specific values for each substance, and are usually functions of temperature, and some are also functions of pressure.

The shape, size, and condition of the surface over which the liquid is flowing

The shape and size of the heat transfer surface have a significant impact on heat transfer. Various heat transfer surfaces can be created from any simple shape of the

body (pipes, plates, etc.). These surfaces create specific conditions for the movement of the heat carrier and heat transfer. Thus, we have seen that the heat transfer coefficient α depends on many factors. Therefore, the value of α varies over a wide range even under the same conditions (Table-2.2.4).

Table-2.2.4

Recommended values of the heat transfer coefficient

Convective heat exchange process	α , W/(m ² ·K)
Natural convection in gases	6 – 100
When gases are forced to move in a pipe or between pipes	12 – 120
Movement of water vapor in a pipe	110 – 2200
Natural convection of water	110 – 1100
Water movement in a pipe	500 – 11000
Boiling water	2200 – 11000
Condensing water vapor	4500 – 22000

Differential equations of convective heat transfer

The main task of the theory of convective heat transfer is to determine the amount of heat transferred through the surface of a solid body through which the fluid flows. In general, the process of stationary convective heat transfer can be described by the following system of differential equations.

Heat transfer equation

The heat flux transferred through the laminar boundary layer of the fluid through the elementary surface area ds of the heat exchange surface is transferred by thermal conduction and is expressed by the following equation based on Fourier's law:

$$dQ = -\chi \frac{dt}{dn} ds$$

According to Newton's law, $dQ = \alpha \Delta t ds$. Equating the right-hand sides of these equations, we obtain the following:

$$-\chi \frac{dt}{dn} = -\alpha \Delta t \quad \text{or} \quad \alpha = -\frac{\chi}{\Delta t} \frac{dt}{dn} \quad (2.2.6)$$

Equation (2.2.6) is called the differential equation of convective heat transfer.

This equation describes the process of heat transfer at the boundary between a solid and a liquid and shows that in order to find the heat transfer coefficient, it is necessary to know the temperature gradient and, therefore, the temperature distribution in the liquid. The temperature distribution is determined from the differential equation of energy.

Energy transfer equation

This equation is derived based on the differential equation of heat conduction (Fourier's law) and the law of conservation of energy.

$$\partial t / \partial \tau + w_x \partial t / \partial x + w_y \partial t / \partial y + w_z \partial t / \partial z = a (\partial^2 t / \partial x^2 + \partial^2 t / \partial y^2 + \nu^2 t / \partial z^2) \quad (2.2.7)$$

Equation (2.2.7) is called the differential equation of energy transfer or the Fourier–Kirchhoff equation. This equation defines the relationship between the time and spatial variations of the temperature at any point in a moving fluid.

The left-hand side of this equation is the complete derivative of the temperature with respect to time $t=f(\tau, x, u, z)$. Since such a derivative is related to the moving matter or substance, it is called the substantial derivative and is denoted by the letter D:

$$Dt / \partial \tau = \partial t / \partial \tau + w_x \partial t / \partial x + w_y \partial t / \partial y + w_z \partial t / \partial z$$

$$\text{and } \nabla^2 t = \partial^2 t / \partial x^2 + \partial^2 t / \partial y^2 + \partial^2 t / \partial z^2$$

Then equation (2.2.7) is written in the following form:

$$Dt / \partial \tau = a \nabla^2 t \quad (2.2.8)$$

The presence of new variables w_x , w_y , and w_z in equation (2.2.7) means that the temperature field in a moving fluid depends on the velocity distribution. This relationship is determined from the differential equation of fluid motion.

Differential equation of fluid motion

The differential equation of fluid motion (Navier-Stokes equation) for an incompressible viscous fluid has the following form::

$$\rho Dw / d\tau = \rho g - \nabla r + \mu \nabla^2 w \quad (2.2.9)$$

Where, g – acceleration; r – pressure; μ – dynamic viscosity coefficient;

For compressible liquids (gases), taking into account the temperature dependence of density $\rho=f(t)$, the above equation can be written as follows:

$$Dw / d\tau = g \beta_p \Delta t - \frac{1}{\rho} \nabla p + \nu \nabla^2 w \quad (2.2.10)$$

where, β_r – temperature coefficient of volumetric expansion; ν – kinematic viscosity coefficient.

Since this equation includes another variable $\rho=1/\nu$ in addition to w_x , w_y , and w_z , it is necessary to introduce another equation into the system of equations. The differential equation of flow continuity (integrity) is such an equation.

Differential equation of flow continuity

This equation has the following form for compressible liquids (gases):

$$\partial \rho / \partial \tau + \partial (\rho w_x) / \partial x + \partial (\rho w_y) / \partial y + \partial (\rho w_z) / \partial z = 0 \quad (2.2.11)$$

For incompressible fluids (stationary state):

$$\partial w_x / \partial x + \partial w_y / \partial y + \partial w_z / \partial z = 0, \text{ or } \operatorname{div} w = 0 \quad (2.2.12)$$

Thus, we have considered four differential equations that describe convective heat transfer.

These differential equations describe the heat transfer process in general.

When solving specific examples of convective heat transfer, it is necessary to add boundary conditions (homogeneity conditions) to the system of differential equations. These conditions are:

1. Geometric conditions that determine the shape and dimensions of the body;
2. Physical conditions that describe the physical properties of the body;
3. Boundary conditions that describe the specific aspects of the process at the boundaries of the system and the external environment;
4. Time conditions that indicate the specific course of the process under consideration in time. Time conditions are not needed for stationary processes.

The above conditions, numerical values, functional dependences are given in the form of tables, etc.

Today, many problems of convective heat transfer have not yet found a solution. Therefore, it is necessary to verify the correctness of the mathematical expression of the process through experiment.

2.3. Laws of conservation of mass, momentum, and energy in laminar flow. Reynolds analogy

Causes of fluid flow

Depending on the causes of occurrence, the movement of liquids is divided into free and forced movement. Free movement or natural convection occurs in an unevenly heated liquid (gas).

The temperatures that arise in this case lead to a difference in density and the emergence of less dense macroparticles in the liquid to the surface of the liquid, which causes movement. The intensity of free movement depends on the type of liquid, the difference in the temperatures of its macroparticles, and the volume in which the process takes place.

Forced movement of liquids or forced convection is associated with the influence of external actuators: fans, pumps, and the like. With their help, it is possible to change the speed of movement of the liquid over a wide range and thereby control the rate of heat exchange.

Fluid flow pattern

In 1884, O. Reynolds showed, based on his experiments, that the motion of a fluid can be laminar or turbulent.

In laminar flow, the particles of the fluid move without mixing. In this case, heat transfer normal to the direction of flow occurs mainly by thermal conductivity. Since the thermal conductivity of the fluid is quite small (for water $\chi=0.60$ W/(m.K)), the heat exchange rate in laminar flow is not high. As the flow velocity increases beyond a certain value, the characteristics of the flow change dramatically. In this case, the straight thread-like shape of the flow changes, becomes wavy, and finally completely mixed.

The motion of the fluid becomes irregular, and the flow is constantly mixed. Such a flow is called turbulent flow.

In turbulent flow, heat is transferred within the flow by conduction and by mixing of almost the entire mass of the fluid. Therefore, heat transfer in turbulent flow is much greater than in laminar flow.

Reynolds showed that the flow of a fluid in a pipe is determined by the value of a dimensionless complex $wd\nu$. This complex is called the Reynolds number:

$$Re=wd/\nu, \quad (2.3.1)$$

where, w – average velocity of the fluid, m/sec; d – pipe diameter, m; ν – coefficient of kinematic viscosity, m²/sec.

Using formula (2.3.1), the Reynolds number can be calculated for any flow in any cross section.

It has been experimentally determined that the critical value of the Reynolds number is $Re_{kr}=2300$. When $Re \leq 2300$, the flow is laminar, and when $Re \geq 10000$, it is turbulent.

The behavior of real fluids in many cases differs sharply from the behavior of laminar flow. They have a special property called turbulence. In real fluid flows in pipes, channels, and boundary layers, the transition from laminar to turbulent flow is clearly observed with increasing Reynolds number. This transition from laminar to turbulent flow is sometimes called the emergence of turbulence, which is of fundamental importance in the entire field of hydrodynamics. Initially, such a transition was observed in flows in straight pipes and channels. In a straight pipe with smooth walls of constant cross section, at small values of the Reynolds number, each particle of the fluid moves along a straight linear trajectory. Due to the presence of viscosity, the particles of the fluid located near the wall begin to move more slowly than the particles located far from the wall. The flow moves as a laminar flow (layers) sliding relative to each other in an ordered manner. However, observations show that at high Reynolds numbers, the flow becomes disordered, i.e., turbulent. The flow undergoes

intense mixing, which can be visually represented by introducing a dye stream into the liquid in the pipe.

In 1883, Osborn Reynolds, observing the movement of water in a pipe, found that with an increase in the speed of the water flow, the steady laminar nature of the moving flow is disturbed. Excitations appear, which are manifested in the fact that the previously laminar motion of the liquid particles, moving in a straight line, in some parts passes into a chaotic state, while the general direction of the motion is preserved. A further change in the flow velocity leads to the appearance of chaotic motion throughout the flow. This state is called unsteady, when the laminar flow turns into an agitated-turbulent flow. The presence of viscosity in liquids provides resistance to the movement of liquid layers relative to each other. In other words, in laminar (layered) flows, internal friction appears instead of viscosity, which is expressed by the amount of applied stresses at the boundaries of the layers, that is, the amount of applied force per unit area. Several concentric layers of fluid move relative to each other in such a way that the fluid velocity is directed along the principal axis. This type of fluid motion is called laminar flow.

The motion of incompressible viscous fluids is described by the Reynolds number $Re = (\rho UL)/\mu = UL/\nu$, ρ -density (kg/m^3 , lb/ft^3), U -maximum velocity of the main flow (m/s , ft/s), L -characteristic length scale (m , ft), μ -dynamic viscosity ($\text{N}\cdot\text{s/m}^2$, $\text{lb/s}\cdot\text{ft}$), $\nu = \mu/\rho$ - kinematic viscosity (m^2/s , ft^2/s), starting from exactly one value of the Reynolds number, the laminar flow becomes turbulent. This value of the Reynolds number is called the critical Reynolds number.

It can be seen from Figure 2.3.1 that when $Re < Re_{\text{crt}}$, the flow is laminar, and when $Re_{\text{crt}} < Re$, the flow becomes turbulent. Consider a circular pipe with a constant diameter along its entire length. Due to viscosity, the velocity at the pipe walls is zero, and in the middle of the pipe the velocity reaches its maximum value. At points on the cylindrical surface whose major axis coincides with the pipe axis, the flow velocity is constant. Separate concentric layers move relative to each other in such a way that the velocity is directed along the axis everywhere. This type of movement is called laminar flow. At a sufficiently large distance from the initial point of the pipe, the velocity distribution in the pipe does not depend on the coordinate directed along the radius, along the length.

The example statement

The following is a description of the forces acting on a fluid for flow in a cylindrical pipe.

Consider the motion of a fluid flowing through a pipe of constant diameter along its entire length and through n pipes of length L and radius r placed inside the pipe.

In real fluids, the fluid adheres to the walls of the pipes and exerts a shear stress on the surface through which it flows. This force is called internal friction, and in liquids this force is viscosity.

Viscosity is a property of gases and liquids that resists the action of external forces that cause the fluid to move.

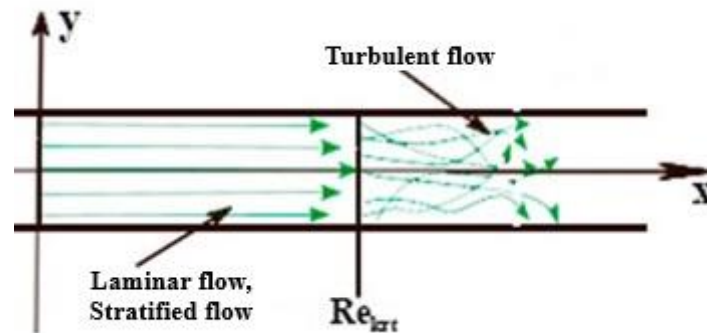


Figure-2.3.1. Transition from laminar flow to turbulent flow.

The presence of shear stresses and the adhesion of the fluid to solid walls cause real fluids in motion to differ qualitatively from ideal fluids.

Now we calculate the forces acting on the fluid motion in the pipe, taking into account n pipes. Due to viscosity, the velocity at the walls of the pipes is zero, and in the middle of the pipes the velocity reaches its maximum value. Separate concentric layers move relative to each other in such a way that the velocity is directed along the axis everywhere and the flow is in a laminar mode.

At a sufficiently large distance from the starting point of the pipes, the velocity distribution of the flow in the pipe does not depend on the coordinate along the radius, but on the longitudinal direction.

The movement of fluid in a pipe occurs at the expense of a decrease in pressure along the pipe axis, but at the expense of the fact that the pressure in each cross-section perpendicular to the pipe axis can be considered constant. The movement of each element of the fluid is accelerated by the decrease in pressure and slowed down by the shear stress caused by friction.

The pressure p is assumed to be constant, that is, $p_0, p_l = \text{const}$ over the entire pipe cross section.

The cylinder is acted upon by the pressure forces $p_0 n \pi y^2$ and $p_l n \pi y^2$ along the major axis, which correspond to the inlet and outlet bases of the cylinder, respectively, and there is a shear force $2\pi n L \tau$ acting on the side surface of the cylinder.

It is necessary to determine the maximum flow velocity in this cylinder, the volume of fluid flowing through the cross-section of the pipe, the coefficient of friction of the pipe along the flow length, and the maximum value of the shear stress.

Solution to the example.

By equating the forces acting on the fluid in the pipes (Figure-2.3.2), we obtain this equation as the equilibrium condition in the direction of motion:

$$p_0 n \pi y^2 = p_1 n \pi y^2 + 2 \pi y n L \tau \quad (2.3.1)$$

The projection of the internal friction force is taken with a positive sign, since the velocity gradient is negative (the flow velocity of the layer decreases with increasing radius y). From this we obtain the following equation: From formula (2.3.1) we determine τ ,

$$\tau = -y(p_0 - p_1) / (2L) \quad (2.3.2)$$

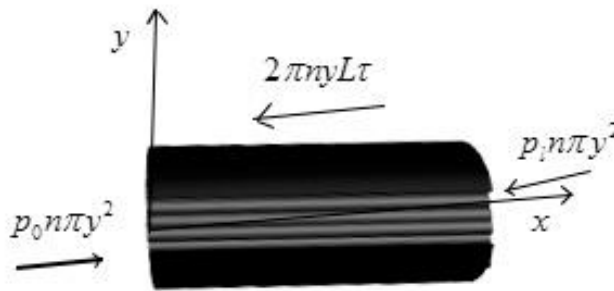


Figure-2.3.2. n tubes are placed inside a tube.

In the case of the velocity U , the coordinate decreases with increasing y and becomes zero when $y=r$. Therefore, according to Hooke's law of friction, we must assume that $\tau = -\mu du/dy$. Substituting this expression into (2.3.2), we obtain:

$$-\mu du/dy = y(p_0 - p_1) / (2L)$$

from this

$$du/dy = -y(p_0 - p_1) / (2\mu L) \quad (2.3.3)$$

Now, when $y=r$, we obtain this by integrating equation (2.3.3) with the initial condition $u(y)=0$ in exchange for viscosity.

$$u(y) = -y^2(p_0 - p_1) / (4\mu L) + C \quad (2.3.4)$$

To find the constant C in equation (4), we use the condition that the velocity $u(y)=0$ when $y=r$, i.e.

$$u(r) = -r^2(p_0 - p_1) / (4\mu L) + C$$

from this

$$(2.3.5)$$

$$C = r^2(p_0 - p_1) / (4\mu L)$$

We find that is. Substituting this value of the constant C into (2.3.4),

$$u(y) = -y^2(p_0 - p_1) / (4\mu L) + r^2(p_0 - p_1) / (4\mu L)$$

equation and in turn

$$u(y) = ((r^2 - y^2)(p_0 - p_1)) / (4\mu L) \quad (2.3.6)$$

Thus, we have a parabolic distribution of velocity along the radius of the tubes, Figure-2.3.3. This velocity reaches its maximum value in the middle of the tube ($y=0$), and it has the following maximum value:

$$u_{\max} = (p_0 - p_1)r^2 / (4\mu L) \quad (2.3.7)$$

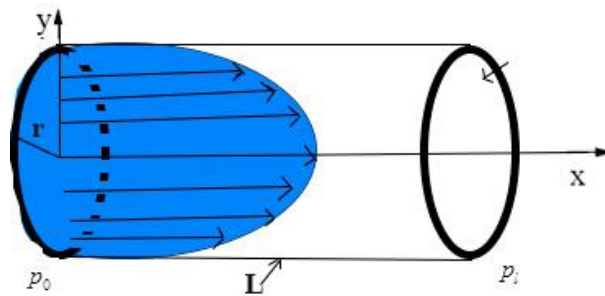


Figure-2.3.3. Flow behavior for a single pipe.

The total amount of fluid flowing through the pipe cross section Q (fluid flow) is defined as the volume of a paraboloid of revolution (Fig. 2.3.3) and is, in turn, determined as follows:

Equation (2.3.6) gives us the following formula:

$$u(y) = -y^2(p_0 - p_1) / (4\mu L) + r^2(p_0 - p_1) / (4\mu L)$$

from this,

$$u(y) = u_{\max}(1 - y^2/r^2) \quad (2.3.8)$$

Using the Hagen-Poiseuille formula for the total flow of fluid through a pipe with a circular cross-section, we determine:

$$Q = \int_0^r u(y) 2\pi y dy = 2\pi u_{\max} \int_0^r \left(y - \frac{y^3}{r^2} \right) dy = 2\pi u_{\max} \left[\frac{y^2}{2} - \frac{y^4}{4r^2} \right]_0^r$$

from this

$$(2.3.9)$$

$$Q=\pi(p_0-p_1)r^4/(8\mu L)$$

We introduce the average flow velocity, the value of which is determined by the cross-section of the pipe as follows:

$$u=Q/(\pi r^2) \quad (2.3.10)$$

Taking into account formula (2.3.9), we write (2.3.10) as follows:

$$u=(p_0-p_1)r^2/(8\mu L)$$

Comparing the function $u(y)$ with $u_{\max}$, we can see that $u(y)=(1/2)u_{\max}$, that is, in laminar flow in a pipe, the average velocity is half the maximum velocity (Figure-2.3.3).

We determine the pressure difference (p_0-p_1)

$$(p_0-p_1)=(8\mu L u)/r^2$$

from this

$$(p_0-p_1)=(32\mu L u)/(2r \cdot 2r)=((32\mu u)/(D)) \cdot (L/D) \quad (2.3.11)$$

where, $D=2r$ pipe diameter.

The pressure loss along the length of the flow is found by the Darcy-Weisbach equation. Now we derive the following formula for N pipes:

$$p_0-p_1=n(\lambda_n/2)\rho u(L/D) \quad (2.3.12)$$

from this

Substituting the value of p_0-p_1 from formula (2.3.11) into (2.3.12), we obtain the following formula

$$\lambda_n=64/nRe \quad (2.3.13)$$

It can be seen that, where n is the number of tubes, the resistance coefficient decreases as the number of tubes increases.

From formula (2.3.12), we introduce the dimensionless resistance coefficient λ_n , which is determined by the following relation.

$$(p_0-p_1)/L=n(\lambda_n/2)\rho u(1/D) \quad (2.3.14)$$

According to formula (2.3.2), the tensile stress reaches its maximum value at the tube wall, where this stress is equal to:

$$\tau_0=(p_0-p_1/L)(r/2) \quad (2.3.15)$$

and this formula is valid regardless of the flow regime (laminar or turbulent). Thus, the shear stress on the pipe wall can be determined experimentally by measuring the pressure drop.

Substituting the value of $p_0 - p_l / L$ from formula (2.3.14) into (2.3.15), we obtain the following formula

$$\tau_0 = (n \lambda_n \rho u^2) / 8$$

This is a formula for calculating the maximum value of the tensile stress in pipes.

Analysis of results.

Above (2.3.13), we present the results obtained based on the formula (Figure 2.3.3).

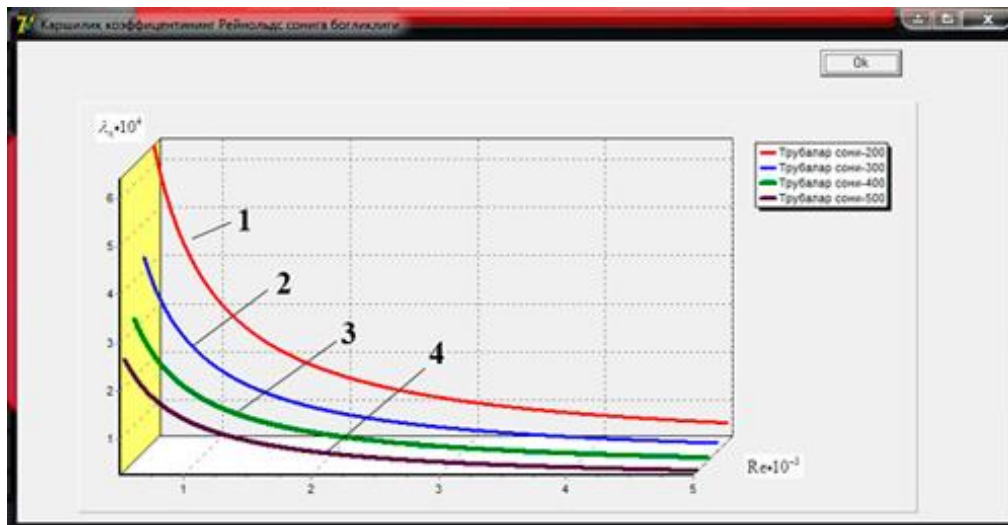


Figure-2.3.4. The dependence of the resistance coefficient in a smooth pipe on the number of tubes n and Re : 1) $n=200$, 2) $n=300$, 3) $n=400$, 4) $n=500$.

Figure 2.3.3 shows the results of calculations showing the dependence of the pipe resistance coefficient λ_n on the Reynolds number Re for smooth n pipes. A comparison of the obtained results shows that the theoretical formula (2.3.14) is valid for all values of the number Re .

At higher values of Re , the drag decreases due to the active activation of turbulence mechanisms.

In the computational experiment, the characteristic parameters Reynolds number Re and drag coefficient λ_n were studied in the following ranges: $Re=500$ to 5000 , $\lambda_n=0.0001$ to 0.0007 .

It can be seen from the figure that the drag coefficient decreases as the number of tubes increases. It can be seen that the above range of critical Reynolds number is proportional to the critical Reynolds number $Re=5770$ obtained for Poiseuille flow in a straight channel.

2.4. Convective heat transfer during flow in cylindrical, spherical, and tubular.

Heat transfer through single-layer and multilayer cylindrical walls

Let heat be transferred through a homogeneous cylindrical wall from a hot heat carrier with temperature t_1 and heat transfer coefficient α_1 to a cold heat carrier with temperature t_2 and heat transfer coefficient α_2 (Figure 2.4.1).

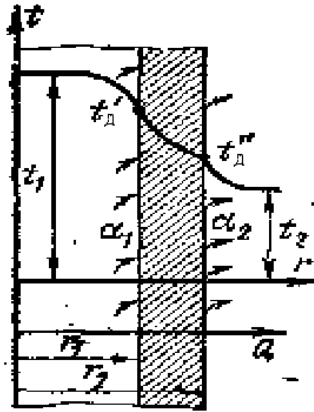


Figure-2.4.1. Heat transfer through a homogeneous cylindrical wall.

Then the following three equations can be written for the heat flow:

$$Q = \alpha_1 \pi d_{uq} l (t_1 - t_o^I)$$

$$Q = \frac{\pi l}{\frac{1}{2\chi} \ln \frac{d_{mau}}{d_{uq}}} (t_o^I - t_o^{II})$$

$$Q = \alpha_2 \pi d_{mau} l (t_o^{II} - t_2)$$

We solve these three equations for the temperature difference and then add them term by term to obtain the following::

$$Q = \frac{\pi l (t_1 - t_2)}{\alpha_1 d_{uq} + \frac{1}{2\chi} \ln \frac{d_{mau}}{d_{uq}} + \frac{1}{\alpha_2 d_{mau}}} \quad (2.4.1)$$

here,

$$k_y = \frac{1}{\frac{1}{\alpha_1 d_{uq}} + \frac{1}{2\chi} \ln \frac{d_{mau}}{d_{uq}} + \frac{1}{\alpha_2 d_{mau}}} \quad (2.4.2)$$

is called the linear heat transfer coefficient, its unit is W/(m·grad).

The heat flux density passing through the cylindrical wall is equal to:

$$q_u = \frac{Q}{l} = k_y \pi (t_1 - t_2)$$

The linear heat transfer coefficient is equal to the amount of heat transferred per unit time from a hot coolant to a cold coolant through a pipe 1 m long, when their temperature difference is 1°. Therefore, equation can be written as:

$$Q = K_{\Pi} \pi l (t_1 - t_2) \quad (2.4.3)$$

The heat flux through a multilayer cylindrical wall is equal to:

$$Q = \frac{\pi l(t_1 - t_2)}{\frac{1}{d_1 d_{uy}} + \sum_{i=1}^{i=n} \frac{1}{2\chi_i} \ln \frac{d_{i+1}}{d_i} + \frac{1}{\alpha_2 d_{mau}}} \quad (2.4.4)$$

The heat flux density obtained with respect to the internal or external surfaces is determined from the following equations:

$$q_{y1} = \frac{Q}{\pi d_1 l} = \frac{k_y}{d_1} (t_1 - t_2)$$

$$q_{y2} = \frac{Q}{\pi d_2 l} = \frac{k_y}{d_2} (t_1 - t_2)$$

The inverse of the linear coefficient of heat transfer is called the linear thermal resistance to heat transfer:

$$R_y = \frac{1}{Q_y} = \frac{1}{\alpha_1 d_{uy}} + \sum_{i=1}^{i=n} \frac{1}{2\chi_i} \ln \frac{d_{i+1}}{d_i} + \frac{1}{\alpha_2 d_{mau}} \quad (2.4.5)$$

where, $\frac{1}{\alpha_1 d_{uy}}$ ба $\frac{1}{\alpha_2 d_{mau}}$ - external thermal resistances; $\sum_{i=1}^{i=n} \frac{1}{2\chi_i} \ln \frac{d_{i+1}}{d_i}$ - thermal resistance of a multilayer cylindrical wall; Unit of measurement of R_s m·grad/W.

We determine the temperature of the inner surface using the following formula:

$$t_o^I = t_1 - \frac{Q}{\alpha_1 d_{uy} \pi l} \quad (2.4.6)$$

and the outer surface:

$$t_o^{II} = t_2 + \frac{Q}{\alpha_2 d_{mau} \pi l} \quad (2.4.7)$$

Critical diameter of the cylindrical wall

It is known that materials with a thermal conductivity coefficient of less than 0.2 W/(m·deg.) are called thermal insulation materials. Such materials include asbestos, cork, felt, wood chips, fiberglass, peat, and similar materials.

Let us consider the conditions for reducing heat loss of materials used for pipe insulation. Let us consider how the thermal resistance changes with the change in insulation thickness, while α_1 , α_2 , d_1 , d_2 , χ_1 , χ_2 , t_1 and t_2 remain unchanged.

From the equation of the thermal resistance of a double-layer cylindrical wall

$$R_y = \frac{1}{k_y} = \frac{1}{\alpha_1 d_1} + \frac{1}{2\chi_1} \ln \frac{d_2}{d_1} + \frac{1}{2\chi_2} \ln \frac{d_3}{d_2} + \frac{1}{\alpha_2 d_3}$$

As the insulation diameter d_3 increases, the resistance of the insulation layer $(\frac{1}{2\chi_1} \ln \frac{d_3}{d_2})$ also increases, but at the same time the thermal resistance of heat transfer from the wall to the external environment $(\frac{1}{\alpha_2 d_3})$ decreases. We take the first derivative with respect to d_3 from the right-hand side of the above equation and set it to zero:

$$\frac{d(R_u)}{d(d_3)} = \frac{1}{2\chi_2 d_3} - \frac{1}{\alpha_2 d_3^2} = 0$$

In this case, the critical diameter corresponding to the extreme point p of the curve $R=f(d_3)$ is determined by the following formula:

$$d_{kp} = d_{u3} = \frac{2\chi_{2u3}}{\alpha_2} \quad (2.4.8)$$

It can be seen from this equation that the critical diameter of the insulation does not depend on the dimensions of the pipe. It decreases as the thermal conductivity of the insulation decreases and increases with decreasing α_2 . The second derivative of the above equation with respect to R_s is greater than zero, which means that the critical diameter corresponds to the minimum of thermal resistance and the maximum of heat flow (Fig. 2.4.2).

Analysis of equation (2.4.8) shows that if d_{iz} increases but remains less than d_{kr} , then the heat loss increases and becomes higher than the heat loss of the pipe without insulation (AK curve).

When $d_{iz}=d_{kr}$ (point K), the losses are maximum. With further increase in the insulation diameter $d_{iz}>d_{kr}$, the heat losses decrease compared to the case of $d_{iz}=d_{kr}$ (curve VK).

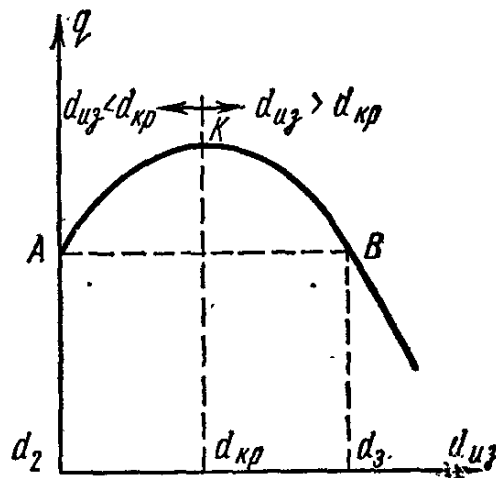


Figure-2.4.2.

Only when $d_{iz} = d_3$, the heat losses are again the same as the heat losses of the insulated pipe. Therefore, for the insulation to work effectively, the critical diameter must be smaller than the diameter of the uninsulated pipe, $d_{kr} \leq d_2$ (Figure-2.4.2).

So, to reduce the heat loss of the wall, the insulation should be as follows:

$$\chi_{u3} \leq \frac{\alpha_2 d_2}{2}$$

$$Q = \frac{\chi}{\delta} F(t_{\text{oes}_1} - t_{\text{oes}_2})$$

It can be seen from the equation that if for a steel pipe $\chi=46$

$\text{Vt}/(\text{m}\cdot\text{K})$ and $\alpha_2 = 14 \text{ W}/(\text{m}^2\cdot\text{K})$ if, then it will be $d_{\text{kr}} = 2 \cdot 46 : 14 = 6,6 \text{ m}$.

Thus, for metal pipes, the critical diameter limit is extremely large, measured in meters. Further increase in diameter leads to a decrease in the heat transferred through the pipe. Within this diameter limit, as the thickness of the metal pipe increases, more heat is transferred from the hot to the cold medium.

If the pipe is made of thermal insulation material, for example asbestos ($\chi=0,167 \text{ W}/(\text{m}\cdot\text{K})$), then at the previous value of α_2

$$(\alpha_2 = 14 \text{ Vt}/(\text{m}^2\cdot\text{K}))$$

$$d_{\text{kr}} = 2 \cdot 0,167 : 14 = 0,024 \text{ m} = 24 \text{ mm}$$

That is, for such pipes with an outer diameter of more than 24 mm, heat transfer decreases as the insulation thickness increases.

For concrete pipes $\chi = 1,17 \text{ W}/(\text{m}\cdot\text{k})$ and $\alpha_2 = 14 \text{ W}/(\text{m}^2\cdot\text{k})$ will result in $d_{\text{kr}} \approx 160 \text{ mm}$.

Heat transfer through a spherical wall

In the stationary regime and with the third type of boundary conditions, let the following be known: the inner diameter of the sphere d_1 , the outer diameter d_2 , the temperature of the hot source inside the sphere t_1 and the temperature of the cold source t_2 , the coefficient of heat transfer from the hot environment to the inner surface of the sphere α_1 , and the coefficient of heat transfer from the outer surface of the sphere to the environment α_2 .

In the stationary case, the heat flow is constant for all isothermal surfaces:

$$Q = \alpha_1 \pi d_1^2 (t_1 - t_o^I)$$

$$Q = \frac{2\pi\chi}{\frac{1}{d_1} - \frac{1}{d_2}} (t_o^I - t_o^{II})$$

$$Q = \alpha_2 \pi d_2^2 (t_o^{II} - t_2)$$

By solving these equations for the temperature difference and adding terms, we find the value of the heat flow:

$$Q = \frac{\pi(t_1 - t_2)}{\frac{1}{\alpha_1 d_1^2} + \frac{1}{2\chi} \left(\frac{1}{d_1} - \frac{1}{d_2} \right) + \frac{1}{\alpha_2 d_2^2}}$$

or

$$Q = k_{\text{sh}} \cdot \pi (t_1 - t_2) \quad (2.4.9)$$

Here k_{sh} is the heat transfer coefficient of the spherical wall, its unit is W/deg .

$$k_{\text{sh}} = \frac{1}{\frac{1}{\alpha_1 d_1^2} + \frac{1}{2\chi} \left(\frac{1}{d_1} - \frac{1}{d_2} \right) + \frac{1}{\alpha_2 d_2^2}} \quad (2.4.10)$$

The quantity that is the inverse of k_{sh} is called the thermal resistance of the spherical wall:

$$R_{uu} = \frac{1}{k_{uu}} = \frac{1}{\alpha_1 d_1^2} + \frac{1}{2\chi} \left(\frac{1}{d_1} - \frac{1}{d_2} \right) + \frac{1}{\alpha_2 d_2^2} \quad (2.4.11)$$

Heat transfer through a ribbed wall

If one side of the wall is flushed with a liquid with a high heat transfer coefficient and the other side is flushed with a gas with a low heat transfer coefficient, then ribbed surfaces are used to equalize the thermal resistances to heat transfer. Ribbed walls increase the surface area of the wall, which results in a decrease in the thermal resistance to heat transfer and an increase in heat flux.

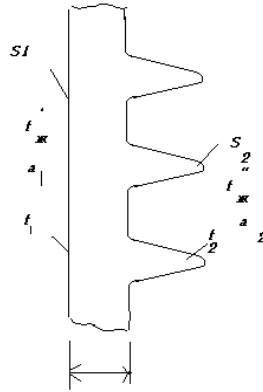


Figure-2.4.3. The diagram shows heat transfer through a ribbed wall.

Let the smooth side of this wall be the surface S_1 , the ribbed side be the surface S_2 , the temperatures of the fluids washing the wall are t'_c and t''_c ($t'_c > t''_c$), the heat transfer coefficients α_1 and α_2 ($\alpha_2 \ll \alpha_1$), the wall surface temperatures t_2 and t_1 , the thermal conductivity coefficient of the wall is λ , and the thickness δ .

The heat transfer through this wall can be expressed by the following equations.

$$Q = \alpha_1 S_1 (t'_c - t_1); \quad Q = \frac{\lambda}{\delta} S_1 (t_1 - t_2); \quad Q = \alpha_2 S_2 (t_2 - t''_c)$$

Solving the above equations for temperature changes and adding terms, we get:

$$Q = \frac{t'_c - t''_c}{\frac{1}{\alpha_1 S_1} + \frac{\delta}{\lambda} \frac{1}{S_1} + \frac{1}{\alpha_2 S_2}} = k_p \Delta t \quad (2.4.12)$$

where, k_p – heat transfer coefficient of a ribbed wall

$$k_p = \frac{1}{\frac{1}{\alpha_1 S_1} + \frac{\delta}{\lambda} \frac{1}{S_1} + \frac{1}{\alpha_2 S_2}} \quad (2.4.13)$$

The heat flux density is determined per unit of smooth wall surface q_1 and per unit of ribbed wall surface q_2 .

$$q_1 = \frac{Q}{S_1} = \frac{\Delta t}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2 S_2}} = k_1 \Delta t \quad (2.4.14)$$

where, k_1 - The heat transfer coefficient obtained relative to the smooth surface of the ribbed wall is:

$$k_1 = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2} \frac{S_1}{S_2}}$$

and

$$q_2 = \frac{Q}{S_2} = \frac{\Delta t}{(1/\alpha_1)(s_2/s_1) + (\delta/\lambda)(s_2/s_1) + 1/\alpha_2} = k_2 \Delta t \quad (2.4.15)$$

where, k_2 - The heat transfer coefficient obtained with respect to the ribbed surface of the wall is:

$$k_2 = \frac{1}{(1/\alpha_1)(s_2/s_1) + (\delta/\lambda)(s_2/s_1) + 1/\alpha_2}$$

The ratio S_2/S_1 is called the coefficient of roasting. We derived the above formulas approximately, because in deriving them we took t_2 and α_2 as constant along the ribbed surface. In reality, the top of the rib is colder than the base, and therefore α_2 varies along the ribbed surface.

For a pipe with a ribbed outer surface, formula can be written as follows:

$$q_t = k_t(t_1 - t_2) \quad (2.4.16)$$

and

$$k_T = \frac{1}{\left[\frac{1}{\alpha_1 d_2} + \frac{1}{2\lambda} \ln \frac{d_2}{d_1} + \frac{1}{\alpha_2 d_2 S_2 / S_1} \right]} \quad (2.4.17)$$

where, d_1 - inner diameter of the pipe; d_2 – outer diameter of the pipe.

Ribbed surfaces are widely used in technology to accelerate heat transfer. For example, mainly the outer surfaces of heating devices are ribbed, since the coefficient of heat transfer from the wall to the air ($\alpha_2=12-60\text{W}/(\text{m}^2\cdot\text{K})$) is much smaller than the coefficient of heat transfer from hot water to the wall ($\alpha_2=2500-6000\text{W}/(\text{m}^2\cdot\text{K})$).

2.5. Free and forced convection

Free convection

The calculation of the heat transfer coefficient for free movement of a cooling fluid over a large volume is usually carried out using a criterion equation of the form.

$$Nu = C(Gr_{\text{жс}} Pr_{\text{жс}})^m (Pr_{\text{жс}} / Pr_c)^{0,25} \quad (2.5.1)$$

where, C and m – coefficients depending on heat exchange conditions.

For gases $Pr=const$, and $(Pr_{\text{жс}}/Pr_c)=1$ we can assume that and therefore all the given formulas are simplified.

When calculating the similarity numbers, the physical parameters γ , λ , α are selected based on the average temperature of the coolant in the volume and at the wall:

$$t_{cp} = 0,5(t_{\text{жс}} + t_c).$$

The volumetric expansion coefficient of the gas β is determined by the formula $\beta = \frac{1}{T_{onp}}$ or is selected depending on the application for a particular coolant.

The average value of the heat transfer coefficient for natural convection of a vertical surface can be obtained from formula (2.5.2).

$$\alpha_{\kappa} = 1,66\sqrt[3]{\Delta t} , \quad (2.5.2)$$

where, $\Delta t = t_{\text{air}} - t_c$ – temperature difference between the air temperature in the room and the wall surface temperature, °C.

To determine the average value of α_{κ} with a horizontal surface, if the heated surface is located below or the cold surface is above, the following formula can be used.

$$\alpha_{\kappa} = 2,16\sqrt[3]{\Delta t} . \quad (2.5.3)$$

Table-2.5.1

The values of the coefficients C and m

Heat transfer conditions	C	m	Size designation
Vertical surfaces (pipes, plates) $10^3 < Gr \cdot Pr < 10^9$ (laminar mode)	0,76	0,25	pipe length, plate height
$Gr \cdot Pr > 10^9$ (turbulent mode)	0,15	0,33	pipe length, plate height
Horizontal pipe $10^3 < Gr \cdot Pr < 10^8$	0,50	0,25	pipe diameter
For objects of any shape $O = (Gr \cdot Pr)_{cp} < 10^{-3}$	0,50	0	Diameter for pipes and balls and height for plates
$10^{-3} < (Gr \cdot Pr)_{cp} < 5 \cdot 10^2$	1,18	0,125	
$5 \cdot 10^2 < (Gr \cdot Pr)_{cp} < 2 \cdot 10^7$	0,54	0,25	
$2 \cdot 10^7 < (Gr \cdot Pr)_{cp} < 5 \cdot 10^{13}$	0,135	0,33	
Horizontal plate with laminar flow			short side of the plate
cooling from above	0,54	0,25	
cooling from below	0,27	0,25	

If the heating surface is on top or the cold surface is on the bottom

$$\alpha_{\kappa} = 1,16\sqrt[3]{\Delta t} . \quad (2.5.4)$$

The calculation of thermal conductivity in a confined volume is carried out according to the heat conduction equations using equivalent thermal conductivity.

$$\lambda_{\text{эКв}} = \lambda \cdot \varepsilon_{\text{к}}, \quad (2.5.5)$$

where, $\varepsilon_{\text{к}}$ – Convection coefficient determined by the result of $Gr \cdot Pr$

$$\begin{aligned} Gr \cdot Pr < 10^3 & \quad \varepsilon_r = 1; \\ 10^3 < Gr \cdot Pr < 10^6 & \quad \varepsilon_r = 0,105 (Gr \cdot Pr)^{0,3}; \end{aligned} \quad (2.5.6)$$

$$10^6 < Gr \cdot Pr < 10^{10} \quad \varepsilon_r = 0,40 (Gr \cdot Pr)^{0,2}. \quad (2.5.7)$$

In the attached calculations, the dependence can be used instead of (2.5.6) and (2.5.7) for the entire range of values of the arguments $Gr \cdot Pr > 10^3$.

$$\varepsilon_{\text{к}} = 0,18 (Gr \cdot Pr)^{0,25}, \quad (2.5.8)$$

Examples for working on assignments

Example-1. The temperature of the vertical wall surfaces at a height of 3 m is 10 °C. The air temperature in the room is 20 °C. Determine the heat transfer coefficient from air to wall.

Solution. Determine the value of the criteria Gr and Pr

$$Gr = \beta g l^3 \Delta t / \gamma^2; \quad Pr = \gamma / \alpha.$$

Based on Table A1 of Appendix A, we determine the air parameters γ , λ and Pr , based on the average temperature $t_{cp} = 0,5(t_c + t_g) = 15^\circ\text{C}$.

$$\gamma_{\text{жк}} = 14,61 \cdot 10^{-6} \text{ м}^2/\text{с}; \quad Pr_{\text{жк}} = 0,705; \quad \lambda_{\text{жк}} = 2,56 \cdot 10^{-2} \text{ W}/(\text{м} \cdot \text{K}).$$

$$Gr = \frac{9,18 \cdot 3^3 \cdot 10}{288(14,61 \cdot 10^{-6})^2} = 4,31 \cdot 10^{10};$$

$$Gr \cdot Pr = 4,31 \cdot 10^{10} \cdot 0,705 = 3,03 \cdot 10^{10}$$

The air traffic regime is turbulent because $Gr \cdot Pr > 10^9$.

Using formula (2.5.1), we find the Nusselt number for a vertical surface by substituting their values for C and m .

$$Nu = 0,15 (Gr \cdot Pr)^{0,33} (Pr_{\text{жк}} / Pr_c)^{0,25} = 0,15 (3,03 \cdot 10^{10})^{0,33} \cdot 1 = 431,48;$$

$$\alpha_{\text{к}} = \frac{Nu \cdot \lambda_{\text{жк}}}{h} = \frac{431,48 \cdot 2,56 \cdot 10^{-2}}{3} = 3,68 \text{ W}/(\text{м}^2 \cdot \text{K}).$$

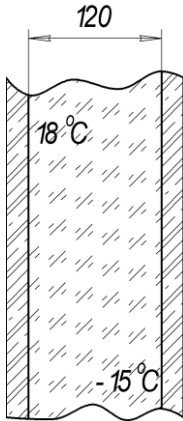
According to the approximate formula (2.5.2) for a vertical surface

$$\alpha_{\text{к}} = 1,66 \sqrt[3]{\Delta t} = 1,66 \sqrt[3]{10} = 3,57 \text{ W}/(\text{м}^2 \cdot \text{K}).$$

Example-2. Determine the heat loss through a double-glazed window measuring $1.2 \times 1.4 \text{ m}^2$, if the distance between the windows is 120 mm, and the glass surface temperatures are 18°C and -15°C .

Solution. We calculate the equivalent thermal conductivity coefficient using the following formula

$$\lambda_{\text{экв}} = \lambda \cdot \varepsilon_K,$$



where, λ - the actual thermal conductivity of air is determined by the average air temperature in accordance with Table A1 of Appendix A;

ε_K - The convection coefficient, which is a function of $Gr \cdot Pr$, can be approximately calculated using the formula $\varepsilon_K = 0,18(Gr \cdot Pr)^{0,25}$

We take the distance between the glasses δ as the defining dimension, and the calculated temperature difference as the defining temperature $\Delta t = t_{c_1} - t_{c_2}$; $t_{\text{онп}} = 0,5(18 - 15) = 1,5^\circ\text{C}$. Using Table A1 in Appendix A, we determine the air parameters

$$\gamma_{\text{жс}} = 13,41 \cdot 10^{-6} \text{ m}^2/\text{s}; \lambda_{\text{жс}} = 24,6 \cdot 10^{-3} \text{ W}/(\text{m} \cdot \text{K});$$

$$Pr = 0,71;$$

$$\beta = \frac{1}{T_{\text{онп}}} = \frac{1}{274,5} \text{ K}^{-1}.$$

Let's count.

$$(Gr \cdot Pr) = \frac{9,81 \cdot 33 \cdot 0,12^3}{274,5(13,41 \cdot 10^{-6})^2} 0,71 = 8,046 \cdot 10^6.$$

Convection coefficient

$$\varepsilon_K = 0,18(8,046 \cdot 10^6)^{0,25} = 9,58$$

Then

$$\lambda_{\text{экв}} = 9,58 \cdot 24,6 \cdot 10^{-3} = 2,36 \cdot 10^{-1} \text{ W}/(\text{m} \cdot \text{K}).$$

The heat loss through a double-glazed window is equal to:

$$\Phi = \frac{\lambda_{\text{экв}}}{\delta} (t_{c_1} - t_{c_2}) A = \frac{2,36 \cdot 10^{-1}}{0,12} (18 + 15) 1,68 = 109 \text{ W}$$

Example-3. Determine the current strength for a nichrome wire with a diameter of 1.0 mm, provided that its temperature does not exceed 300°C . At a temperature of

300 °C, the resistance of 1 linear meter of wire is $R=6.0$ Ohm, and the ambient temperature is $t_m=20$ °C.

We perform calculations for two cases.

- when the wire is in still air;
- when the wire is in still water under pressure at a saturation temperature above 300 °C.

Solution. Using the Newton-Richmann law, we determine the strength of the heat flow from the wire to the environment

$$q_l = \pi d_h \alpha (t_c - t_{\text{жс}}).$$

To determine α , we find the Grashof number and the Nusselt number:

a) The environment is air.

Using Table A1 in Appendix A, we find the physical parameters of air at $t_{\text{оп}}=0,5(300+20)=160$ °C.

$$\lambda_{\text{жс}}=3,64 \cdot 10^{-2} \text{ W/(m}\cdot\text{K)}; \gamma_{\text{жс}}=30,09 \cdot 10^{-6} \text{ m}^2/\text{s}; Pr_{\text{жс}}=0,682.$$

Grashof number

$$Gr = \beta \frac{g \cdot l^3 \cdot \Delta t}{\gamma^2} = \frac{9,81 \cdot (0,001)^3 (300 - 20)}{(160 + 273) \cdot (30,9 \cdot 10^{-6})^2} = 6,64$$

$$Gr \cdot Pr = 6,64 \cdot 0,682 = 4,53.$$

According to Table 2.5.1, with the result $Gr \cdot Pr=4.778$

$C=7,18$; $m=0,125$, where, according to formula (2.2),

$$Nu = 1,18 (Gr \cdot Pr)^{0,125} = 1,18 \cdot 4,53^{0,125} = 1,435.$$

$$\alpha = \frac{Nu \cdot \lambda}{d} = \frac{1,435 \cdot 3,64 \cdot 10^{-2}}{0,1 \cdot 10^{-2}} = 52,2 \text{ W/(m}^2\cdot\text{K)}.$$

$$q_l = \pi d_y \cdot \alpha (t_c - t_{\text{жс}}) = 3,14 \cdot 0,001 \cdot 52,2 \cdot (300 - 20) = 45,89 \text{ W/s}$$

$$q_e = J^2 R \quad J = \sqrt{\frac{q_e}{R}} = \sqrt{\frac{45,89}{6,0}} = 2,25 \text{ A}$$

Exampel-4. A tank-accumulator with a calcium chloride solution has a horizontal cooling coil made of pipes with an outer diameter of 57 mm. The density of the calcium chloride solution at a temperature of 15 °C is $\rho_{\text{жс}}=1220$ /m³. The freezing temperature of the solution is $t_3 = -25.7$ °C, the average temperature of the outer surface of the pipe is $t_{cm}=-20$ °C, and the temperature of the solution far from the pipe surface is $t_{\text{жс}}=-10$ °C.

Determine the heat transfer coefficient α from the solution to the pipe surface with free movement of the liquid.

Solution. The heat transfer during free movement around a horizontal pipe (snake) is determined by the equation

$$Nu=0,5(Gr \cdot Pr)^{0,25}.$$

The defining dimension is the outer diameter of the pipe, and we take the average temperature as the defining temperature

$$t_{cp} = \frac{t_c + t_{\infty}}{2} = \frac{-20 - 10}{2} = -15^{\circ}\text{C}.$$

This formula is valid when $1 \cdot 10^3 \leq (Gr \cdot Pr) \leq 10^3$.

We determine the thermophysical properties of calcium chloride at $\rho=1220 \text{ kg/m}^3$ and $t_3=-25,7^{\circ}\text{C}$.

$$\lambda_{-20}=0,523 \text{ W/(m}\cdot\text{K)}$$

$$\lambda_{-15}=0,518 \text{ W/(m}\cdot\text{K)}$$

$$\lambda_{-20}=0,511 \text{ W/(m}\cdot\text{K)}$$

$$\gamma_{-10}=4,87 \cdot 10^{-6} \text{ m}^2/\text{s}; \quad \gamma_{-15}=6,20 \cdot 10^{-6} \text{ m}^2/\text{s}; \quad \gamma_{-20}=7,77 \cdot 10^{-6} \text{ m}^2/\text{s};$$

$$Pr_{-10}=33; \quad Pr_{-15}=42,5; \quad Pr_{-20}=53,80;$$

$$\beta_{-10}=3,5 \cdot 10^{-4}; \quad \beta_{-20}=3,3 \cdot 10^{-4} \text{ K}^{-1}; \quad \beta_{-15}=3,4 \cdot 10^{-4} \text{ K}^{-1}$$

We define similarity criteria that characterize the heat transfer process:

$$Gr_{\infty, d} = \frac{q \cdot \alpha^3 \Delta t \cdot \beta_{\infty}}{\gamma_{\infty}^2} = \frac{9,81 \cdot 0,057^3 \cdot (-10) \cdot 3,4 \cdot 10^4}{(6,20 \cdot 10^{-6})^2} = 1,607 \cdot 10^5$$

$$Ra = (Gr_d \cdot Pr_{\infty}) = (1,607 \cdot 10^5 \cdot 42,5) = 6829750 = 6,83 \cdot 10^6.$$

$$Nu_{\infty, d} = 0,5(6,83 \cdot 10^6)^{0,25} = 25,56$$

$$\text{Heat transfer coefficient } \alpha = \frac{Nu_{\infty, d} \cdot \lambda_{\infty}}{d_u} = \frac{25,56 \cdot 0,518}{0,057} = 232,29 \text{ W/(m}^2 \cdot \text{K)}.$$

Forced convection

In forced convection, there is a relationship between similarity numbers

$$Nu = c Re^n \cdot Pr^m (Pr_m/Pr_c)^{0,25}. \quad (2.5.9)$$

where, the coefficients c and n are taken according to Table 2.5.2, depending on the type of coolant movement.

The Nusselt number in laminar flow around a $Re < 10^5$ wedge-shaped body with an opening angle $\pi\beta$ is determined by the formula.

$$Nu = 1,12 \frac{(\beta + 0,2)^{0,1}}{(2 - \beta)^{0,5}} Pr^m Re^{0,5} \quad m = 1/3 + 0,067\beta - 0,026\beta^2. \quad (2.5.10)$$

With turbulent flow - $Re < 10^5$

$$Nu = 0,061 \frac{(\epsilon + 0,2)^{0,1}}{(2 - \epsilon)^{0,5}} Pr^m Re^{0,8} \quad m = 1/3 + 0,067\epsilon - 0,026\epsilon^2. \quad (2.5.11)$$

Table-2.5.2

The values of the coefficients c , n , and m in equation 2.5.9

Type of heat carrier movement	c	n	m	Size designation
Laminar along a horizontal plate, $Re < 5 \cdot 10^5$	0,664	0,5	0,33	plate width along the flow
Turbulent along a horizontal plate, $Re \gg 5 \cdot 10^5$	0,037	0,8	0,43	
Laminar in the pipe, $Re < 2300$	0,15	0,33	0,43	inner diameter of the pipe, d_{inner}
Turbulent in the pipeline, $Re = 10^4 \dots 5 \cdot 10^6$	0,021	0,8	0,43	
Pipeline crossing, $Re = 2300 \dots 10^4$	0,008	0,9	0,43	
Laminar along the cylinder $10^{-3} < Re < 10^3$ $Re = 5 \dots 80$ $Re = 80 \dots 10^3$	0,5 0,92 0,73	0,5 0,4 0,46	0,38 0,40 0,40	outer diameter of the cylinder, d_{outer}
Turbulent along the cylinder $10^3 < Re < 10^9$ $Re = 5 \cdot 10^3 \dots 5 \cdot 10^4$ $Re > 5 \cdot 10^4$	0,25 0,22 0,026	0,6 0,6 0,80	0,38 0,4 0,4	
Cross-flow around the corridor pipe bundles $Re < 10^3$ $Re = 10^3 \dots 10^5$	0,56 0,23	0,65 0,65	0,33 0,33	outer diameter of the cylinder, d_{outer}
Cross flow around rigid tube bundles $Re < 10^3$ $Re = 10^3 \dots 10^5$	0,56 0,41	0,65 0,60	0,33 0,33	outer diameter of the cylinder, d_{outer}

The Nusselt number is determined during the passage of the ball for any regime of coolant movement

$$Nu = 2 + 0,03 Pr^{0,33} \cdot Re^{0,54} + 0,35 Pr^{0,35} \cdot Re^{0,58} \quad (2.5.12)$$

In this equation, the diameter of the sphere is taken as a characteristic dimension.

The value of the Nusselt number can also be determined using the Krimer-Loos equation:

$$Nu = 2 + Pr^{0,33} (0,4 \cdot Re^{0,5} + 0,06 Re^{0,67}); \quad (2.5.13)$$

or according to the Rantze-Marshall formula:

$$Nu=2+0,6Pr^{0,33}Re^{0,5} \quad (2.5.14)$$

The Nusselt number calculated by (2.5.9) should be corrected for the average heat transfer coefficient ε_e along the length of the pipe, for the bend ε_{u32} , for the change in the thermal conductivity of the first rows of pipes. (Cylinder) During the transverse flow ε_i , the correction coefficient ε_ψ , taking into account the angle of attack (i.e. the angle between the cylinder axis and the velocity vector of the medium), the correction coefficient ε_s , taking into account the effect of the relative pitch ε_s .

The flow rate included in the Re criterion is defined as the ratio of the volumetric flow rate of the fluid V (m³/s) in the narrowest part of the beam to the area S (m²) of the narrowest section.

$$\alpha=\alpha_{pac}\cdot\varepsilon_e\cdot\varepsilon_{u32}\cdot\varepsilon_i\cdot\varepsilon_{\psi}\cdot\varepsilon_s. \quad (2.5.15)$$

The correction for pipe bends (bends, spirals) is calculated using the formula

$$\varepsilon_{u32}=1+1,77d/R, \quad (2.5.16)$$

where, d – pipe diameter, m; R – spiral radius, m.

The correction for the change in the average heat transfer coefficient along the length of the pipe or channel depending on the driving mode at $l/d<50$ is taken according to Table 2.3.

Table-2.5.3

The correction coefficient ε_l

Movement mode	l/d relationship							
	1	2	5	10	15	20	30	40
Laminar	1,90	1,70	1,44	1,28	1,18	1,13	1,05	1,02
Turbulent Re=10 ⁴	1,65	1,50	1,34	1,23	1,17	1,13	1,07	1,03
2·10 ⁴	1,51	1,40	1,27	1,18	1,13	1,10	1,05	1,02
5·10 ⁴	1,34	1,27	1,18	1,13	1,10	1,08	1,04	1,02
10·10 ⁴	1,28	1,22	1,15	1,10	1,08	1,06	1,03	1,02
100·10 ⁴	1,14	1,11	1,08	1,05	1,04	1,03	1,02	1,01

A correction is taken for the change in heat transfer in the first rows of the pipe bundle:

first row $\varepsilon_i=0,6$;

second row of the corridor set $\varepsilon_i=0,9$;

second row of the chessboard $\varepsilon_i=0,7$;

third and subsequent rows $\varepsilon_i=1,0$.

The correction coefficient ε_s , taking into account the effect of the relative steps S_1/d and S_2/d

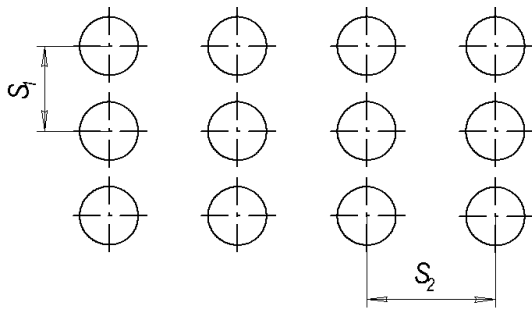


Figure-2.5.1

Corridor layout scheme

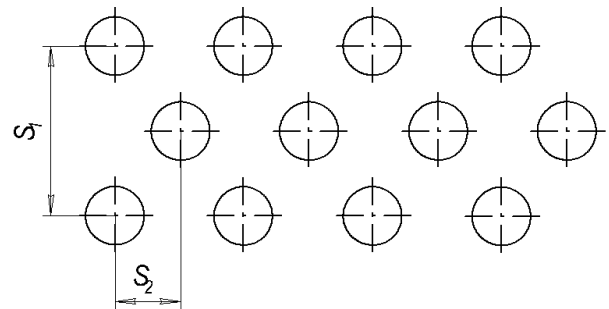


Figure-2.5.2

Chess placement scheme

Table-2.5.4

Correction factor ε_ψ for pipe bundles located at an angle ψ to the fluid flow.

ψ , deg.	90	80	70	60	50	40	30	20	10	0
ε_ψ	1	1	0,98	0,94	0,88	0,78	0,67	0,52	0,42	0,40

For corridor placement:

$$\varepsilon_s = (S_2/d)^{-0,15}$$

For chess placement

When $S_1/S_2 < 2$ $\varepsilon_s = (S_1/S_2)^{1/6}$, when $S_1/S_2 \geq 2$ $\varepsilon_s = 1,12$.

The average value of the heat transfer coefficient for the entire pipe bundle is determined by the formula

$$\alpha_{cp} = \frac{\sum_{i=1}^n \alpha_i A_i}{\sum_{i=1}^n A_i}, \quad (2.5.17)$$

where, α_i - average heat transfer coefficient of the i -row;

A_i - total heat exchange surface of the pipes in the i -row;

n - number of rows in the set.

The average value of the heat transfer coefficient in annular channels can be determined by the following equation

$$Nu = 0,017 Re^{0,8} \cdot Pr^{0,4} (Pr_{\text{жс}}/Pr_c)^{0,25} (d_2/d_1)^{0,18}, \quad (2.5.18)$$

where, d_1 - inner diameter of the annular channel;

d_2 - outer diameter.

Formula (2.19) is valid for $d_2/d_1 = 1, 2 \dots 14$.

$$l/d=50\dots460 \text{ and } Pr=0,7\dots100.$$

Heat transfer calculations in circular and square finned tube bundles with transverse gas flow can be performed using the following equations within the range of values.

$$3 \cdot 10^3 < Re < 25 \cdot 10^3 \text{ and } 3 \leq d/\vartheta \leq 4,8, \quad Nu = c Re^m (d/\vartheta)^{-0,54} (h/\vartheta)^{-0,14}, \quad (2.5.19)$$

where, ϑ – rib step, $h=0,5(D-d)$, where, D – diameter or side of a rib, d – outer diameter of the pipe. $c=0,104$ for round ribbed corridor pipe sets, for square ribs $c=0.096$ and $m=0.72$ for both cases. For chess sets with rounded edges, $c=0.223$; with square edges, $c=0.205$; in both cases, $m=0.65$.

When continuous air flows around the plate, the calculation formula (2.5.9) is simplified and takes the following form:

- in laminar mode

$$Nu = 0,57 Re^{0,5}; \quad (2.5.20)$$

- in turbulent mode

$$Nu = 0,32 Re^{0,8}; \quad (2.5.21)$$

$$\alpha_{pac} = 5,9 \omega^{0,8} l^{-0,2}. \quad (2.5.22)$$

For air with forced cross-flow around a pipe, we can use the following formula

$$\alpha_{pac} = 11,6 \sqrt{\omega}. \quad (2.5.23)$$

For hot water surfaces

$$\alpha_{pac} = 5,7 + 4,1 \omega, \quad (2.5.24)$$

where, ω - coolant speed, m/s;

l – measurement of detection, m.

The heat transfer coefficient from a horizontal pipe to air with proper convection can be determined by the formula

$$\alpha_{pac} = 1,16 \sqrt{(t_{h.n.} - t_o) / d_2}. \quad (2.5.25)$$

For wind-induced forced convection conditions

$$\alpha_{pac} = 4,65 \omega^{0,7} / d^{0,3}. \quad (2.5.26)$$

where, $t_{h.n.}$ – external surface temperature, °C;

t_o – ambient temperature, °C;

d_2 – outer diameter of the pipe, insulation, m.

Example-5. Hot water with a temperature of 50 °C flows through a pipe with a diameter of 50 mm and a length of 3 m, bent in the form of a coil with a diameter of 600 mm, at a speed of $\omega = 0.8$ m/s. Determine the heat transfer coefficient α from the wall to the water, if the wall temperature is $t_c = 70$ °C.

Solution. We determine the water parameters at $t_{жс} = 50$ °C and $t_c = 70$ °C according to Table A5 of Appendix A.

When $t_{жс} = 50$ °C $\lambda_{жс} = 0,648$ W/(m·K); $\gamma_{жс} = 5,56 \cdot 10^{-7}$ m²/s; $Pr_c = 3,54$.

When $t_c = 70$ °C $\lambda_{жс} = 0,668$ W/(m·K); $\gamma_{жс} = 4,15 \cdot 10^{-7}$ m²/s; $Pr_c = 2,55$.

We calculate the Reynolds number and Nusselt number:

$$Re = \omega d / \gamma_{жс} = 0,8 \cdot 0,05 / 5,56 \cdot 10^{-7} = 7,2 \cdot 10^4;$$

$$Nu = 0,021 Re^{0,8} Pr_{жс}^{0,43} (Pr_{жс} / Pr_c)^{0,25} = 0,021 (7,2 \cdot 10^4)^{0,8} \cdot 3,54^{0,43} \cdot (3,54 / 2,55)^{0,25} = 303.$$

We introduce the changes ε_e and ε_{u32}

$$l/d = 3/0,05 = 60 > 50 \quad \varepsilon_e = 1;$$

$$\varepsilon_{u32} = 1 + 1,77 d/R = 1 + 1,77 \cdot 50/300 = 1,295.$$

Then

$$\alpha_{pac} = \frac{Nu \cdot \lambda_{жс}}{d} \cdot \varepsilon_e \cdot \varepsilon_{u32} = \frac{303 \cdot 0,648}{0,05} \cdot 1,295 = 5085 \text{ W/(m}^2 \cdot \text{K)}.$$

Example-6. Determine the average heat transfer coefficient in an air heater with an eight-row cross-section of pipes to the air flow. Pipe diameter $d = 16$ mm; $S_1 = 1.5d$; $S_2 = 2.0d$.

The average air velocity in a narrow section is 10 m/s, the inlet temperature is 20 °C, and the outlet temperature is 60 °C.

Solution. Based on the average air temperature $t_{cp} = 0.5 (t_e^I + t_e^{II}) = 0.5(20 + 60) = 40$ °C in Table A1 of Appendix A, we find the physical parameters of the air: $\lambda = 27,6 \cdot 10^{-3}$ W/(m·K); $\nu = 16,96 \cdot 10^{-6}$ m²/s and Prandtl number $Pr = 0.70$.

$$\text{We calculate the Reynolds number } Re = \frac{\omega d}{\gamma} = \frac{10 \cdot 0,016}{16,96 \cdot 10^{-6}} = 9434.$$

For a transverse pipe bundle at $Re = 10^3 \dots 10^5$, we select the following coefficients using formula (2.9) in Table 2.7.2: $c = 0,26$, $n = 0,65$ and $t = 0,33$.

$$Nu = 0,26 \cdot Re^{0,65} \cdot Pr^{0,33} (Pr_{жс} / Pr_c)^{0,25} \cdot \varepsilon_s,$$

here, for air

$$Pr_{жс} / Pr_c = 1; \varepsilon_s = (S_2/d)^{-0,15} = (2d/d)^{-0,15} = 0,901;$$

$$Nu=0,26(9434)^{0,65} \cdot 0,70^{0,33} \cdot 0,901=79,82.$$

$$\alpha = \frac{Nu \cdot \lambda}{d} = \frac{79,82 \cdot 27,6 \cdot 10^{-3}}{16 \cdot 10^{-3}} = 137,2 \text{ W/(m}^2 \cdot \text{K)}.$$

The average heat transfer coefficient, taking into account heat transfer in the first, second and third rows of pipes

$$\alpha_{\text{avg}} = \alpha \frac{\varepsilon_{i1} + \varepsilon_{i2} + \varepsilon_{i3...6}}{n} = 137,2 \frac{(0,6 + 0,9 + 6)}{8} = 128,6 \text{ W/(m}^2 \cdot \text{K)}.$$

Example-7. In a one-piece pipe pasteurizer, milk moves through the pipes at a speed of $w=1.1$ m/s. The diameter of the pipes is 35/32 mm, and the total length is 18 m. The average temperature of the milk is 41 °C, and the surface of the walls is 65 °C. Calculate the heat flux absorbed by the milk.

Solution. Using Appendix A, Table A17, we determine the thermophysical properties of milk at $t_{\text{mc}}=41^\circ\text{C}$. $\lambda_{\text{mc}}=0,565$ W/(m·K); $\gamma_{\text{mc}}=0,922 \cdot 10^{-6}$ m²/s; $Pr_{\text{mc}}=7,57$.

When $t_{\text{cm}}=65^\circ\text{C}$ $Pr_{\text{cm}}=4,36$.

We define the Reynolds criterion that describes the process

$$Re = \frac{\omega \cdot d_{\text{BH}}}{\gamma_{\text{mc}}} = \frac{1,1 \cdot 0,32}{0,922 \cdot 10^{-6}} = 38178$$

For turbulent motion in a pipe, $Re=10^4 \dots 5 \cdot 10^6$ $c=0,21$, $n=0,8$ and $m=0.43$ and equation (2.10) takes the form

$$Nu_{\text{mc}}=0,21 \cdot Re^{0,8} \cdot Pr_{\text{mc}}^{0,43} (Pr_{\text{mc}}/Pr_{\text{cm}})^{0,25} \cdot \varepsilon_e=0,21 \cdot (38178)^{0,8} \cdot 7,57^{0,49} (7,57/4,36)^{0,25}=266,42$$

When $l/d_{\text{BH}}=18/0,032=562>50$ $\varepsilon_e=$,

Heat transfer coefficient

$$\alpha = \frac{Nu_{\text{mc}} \cdot \lambda_{\text{mc}}}{d_{\text{BH}}} = \frac{266,42 \cdot 0,565}{0,032} = 4703,9 \text{ W/(m}^2 \cdot \text{K)}.$$

We determine the Φ (kW) received by the milk

$$A=\alpha \cdot \pi \cdot d_y \cdot l(t_{\text{cm}}-t_{\text{c}})=4703,9 \cdot 3,14 \cdot 0,035 \cdot 18(65-41)=223,3 \text{ kW}$$

Example-8. An aluminum electric wire with a diameter of 6 mm is cooled by a transverse air flow at a pressure of $p=1,01 \cdot 10^5$ Pa. The temperature and speed of air movement away from the wire are $t_{\text{ac}}=20^\circ\text{C}$; $w=1,2$ m/s, respectively. The specific electrical resistance of the wire $\rho=2,86 \cdot 10^{-8}$ Om·m m²/m.

If the surface heat flux density is $q = 2.7$ kW/m², find the surface temperature of the wire, as well as the current strength in the electric wire that will provide the given surface heat flux density.

Solution. According to Appendix A, Table A1, we determine the thermophysical properties of air at a temperature of 20 °C: $\lambda_{жс}=2,59 \cdot 10^{-2}$ W/(m·K); $\gamma_{жс}=15,06 \cdot 10^{-6}$ m²/s, $Pr_{жс}=0,703$.

For forced motion, we find the Reynolds number $w=1.2$ m/s

$$Re = \frac{w \cdot d_n}{\gamma} = \frac{1,2 \cdot 0,006}{15,06 \cdot 10^{-6}} = 478,08$$

When $Re=478,08$, coefficient $c=0.79$; $p=0.46$ and $m=0.40$ and equation (2.5.9) will have the following form.

$$Nu_{жс}=0,79 Re^{0,46} \cdot Pr_{жс}^{0,40}=0,79 \cdot (478,08)^{0,46} \cdot (0,703)^{0,40}=11,72$$

Heat transfer coefficient

$$\alpha = \frac{Nu_{жс} \cdot \lambda_{жс}}{d_{жс}} = \frac{11,72 \cdot 2,59 \cdot 10^{-2}}{0,006} = 50,6 \text{ W/(m}^2 \cdot \text{K)}$$

Temperature in wire sheath

$$t_{см}=t_{жс}+q/\alpha=20+2,7 \cdot 10^3/50,6=73,36^\circ\text{C}$$

From the energy balance equation

$$\alpha \pi d l (t_{см}-t_{жс})=J^2 R,$$

We determine the current strength J ,

$$\text{where, } R = \frac{\rho \cdot l}{\pi d^2 / 4} = \frac{4 \rho \cdot l}{\pi d^2}; \quad J^2 = \frac{\alpha \cdot \pi \cdot d \cdot l (t_{см} - t_{жс})}{R} = \frac{\alpha \cdot \pi^2 \cdot d^3 \cdot \Delta t}{4 \rho};$$

$$J = \pi d \sqrt{\frac{\alpha d \Delta t}{4 \rho}} = \pi \cdot 6 \cdot 10^{-3} \sqrt{\frac{50,6 \cdot 6 \cdot 10^{-3} \cdot 53,36}{4 \cdot 2,86 \cdot 10^{-8}}} = 224,19 \text{ A}$$

2.6. Convective heat transfer under forced convection conditions

The rate of heat transfer in forced motion is determined mainly by the characteristics of the medium (liquid or gas) motion.

In practice, turbulent motion is often encountered, in which the heat transfer coefficient is much larger than in laminar motion. As the Re number increases, that is, the turbulence intensity increases, the heat transfer between the fluid and the wall increases.

Heat transfer also largely depends on the shape of the surface that the medium is washing. If the flow does not separate from the surface being washed anywhere, such a flow is called continuous flow.

If the system has sharply changing curves or increasing channels, sharp turns, and the like, then in some places the flow separates from the surface, that is, discontinuous flow occurs.

Today, the process of continuous flow has been studied almost completely, and there are various calculation formulas for different surfaces. The process of

discontinuous flow has been studied for simpler cases (when washing a cylinder, sphere, or a set of pipes).

Laminar flow in forced convection can be of two types - viscous and viscous - gravitational. In viscous flow, viscous forces prevail over lift forces. In this flow, natural convection does not occur, and heat is transferred to the channel walls only by heat conduction.

In a viscous-gravity flow, buoyancy forces are taken into account, and in this case, the forced movement of the fluid occurs together with natural convection.

As a result, the laminarity of the flow is disturbed. In such a flow, heat is transferred both by thermal conductivity and convection.

When $Re=2 \cdot 10^3 \div 10^4$ intermittent leakage is observed and at $Re > 10^4$ the flow becomes turbulent.

These two actions differ from each other depending on the value of the product $Gr \cdot Rr$.

If $Gr \cdot Rr > 8 \cdot 10^5$, the flow regime is viscous-gravitational, and the following formula is appropriate for this case:

$$Nu = 0,15 Re^{0,33} \cdot Rr^{0,43} (Gr \cdot Rr)^{0,1} \cdot (rr/Rr_g)^{0,25} \quad (2.6.1)$$

In this equation, the equivalent diameter is taken as the determining dimension, and the average temperature of the liquid is taken as the determining temperature.

It has been experimentally established that for liquid droplets, the heat transfer coefficient α_{is} is during their heating from the wall is always greater than the heat transfer coefficient α_{sov} during their cooling.

The difference $\alpha_{is} - \alpha_{sov}$ increases with increasing temperature pressure $t_g - t_s$. This is taken into account by introducing a multiplier of $(Rr/Rr_g)^{0,25}$ into the criterion equation. As mentioned above, Rr_g is calculated at the average wall temperature.

This value is 0.5–2 for liquids and 1 for gases. Formula (2.6.1) can be used for all liquids if the length of the heat-stabilized part of the pipe is $l \geq 50d$. For short pipes with a length of less than $50 \cdot d$, α , determined from formula (2.6.1), is multiplied by the correction coefficient ε_l . The numerical value of the coefficient ε_l depends on l/d as follows.

l/d	1	2	5	10	15	20	30	40	50
ε_l	1,9	1,7	1,44	1,28	1,18	1,13	1,05	1,03	1.

For the intermediate flow regime of the heat carrier in pipes and channels ($2300 < Re < 10000$), there are currently no formulas that summarize various experiments.

Only when $Re \geq 10^4$ does a developed turbulent motion occur. In this case, the mixing process of liquid particles occurs so rapidly that, as a result, the temperature across the cross section of the turbulent flow core remains almost constant.

M. A. Mikheev proposed the following formula for calculating heat transfer in pipes using the turbulent flow method:

$$Nu = 0,021 Re^{0,8} \cdot Rr^{0,43} \cdot (Rr/Rr_g)^{0,25} \quad (2.6.2)$$

In this formula, the equivalent diameter is taken as the determining dimension, and the average temperature of the liquid is taken as the determining temperature. For air, if we assume $Rr=0.7$, equation (2.6.2) becomes:

$$Nu = 0,018 Re^{0,8} \quad (2.6.3)$$

Formula (2.6.3) is used for various heat carriers: liquids, gases, superheated steams, where $l/d \geq 50$, $Re=10^4 \div 5 \cdot 10^6$ and $Rr=0,6 \div 250$.

If the channel length $l < 50d_{ekv}$, the correction coefficient α calculated from formula (2.6.2) is multiplied by ε_l . The value of the coefficient ε_l is taken from Table 2.6.1.

The dependence of the correction coefficient ε_l on l/d .

Table-2.6.1.

Re	The value of ε_l when l/d is as follows								
	1	2	5	10	15	20	30	40	50
10^4	1,65	1,5	1,34	1,23	1,17	1,13	1,07	1,03	1
$2 \cdot 10^4$	1,51	1,4	1,27	1,18	1,13	1,10	1,05	1,02	1
$5 \cdot 10^4$	1,34	1,27	1,18	1,13	1,10	1,08	1,04	1,02	1
10^5	1,28	1,22	1,15	1,1	1,08	1,06	1,03	1,02	1
10^6	1,14	1,11	1,08	1,05	1,04	1,03	1,02	1,01	1

When the heat carrier flows in bent pipes, the heat transfer process is complicated by the influence of centrifugal forces on the flow.

Its effect is taken into account by introducing the correction coefficient ε_R into formula (9.46). For serpentine pipes, ε_R is determined from the following ratio:

$$\varepsilon_R = 1 + 1,77 d/R,$$

where, R – bending radius; d – pipe diameter.

As is known, in most heat exchangers (air heaters, water economizers, etc.), the tubes are arranged transversely to the flow. Figure 2.6.1 shows a view of the liquid covering the tube transversely.

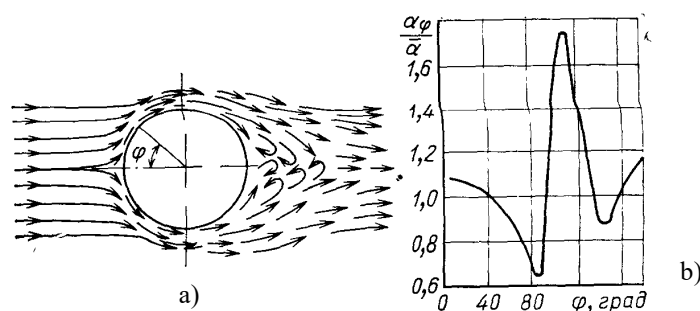


Figure-2.6.1. Description of the flow along the circumference of the cylinder (a) and the change in heat transfer (b).

A boundary layer forms on the surface of the opposite part of the pipe, its thickness increasing along the flow direction. At some points, the boundary layer is observed to break off from the surface, and two symmetrical accumulations appear behind the pipe.

The location of the boundary layer breakpoint depends on the Re number. If the Re number is not very large and the turbulence level of the flow entering the pipe is low, the boundary layer breakpoint is observed at $82\text{--}84^\circ$. As the Re number increases, the motion in the boundary layer becomes turbulent. As a result, due to the increase in kinetic energy, the boundary layer breakpoint shifts downstream ($\varphi \approx 120\div 140^\circ$), which leads to a decrease in the accumulation zone behind the pipe and improved coverage.

This unique coating of the pipe affects the heat exchange between the liquid and the pipe surface.

Figure 2.6.1 (b) shows the ratio of the local heat transfer coefficient α_φ to the average heat transfer coefficient $\bar{\alpha}$ as a function of the angle φ . It can be seen from the figure that the heat transfer occurs most rapidly in the pipe ($\varphi=0^\circ$), is minimal at $\varphi=90^\circ\div 100^\circ$, is most pronounced at $\varphi=120^\circ$, and then decreases again at $\varphi=140^\circ$. The decrease in heat transfer in the section of the pipe with $\varphi=0\div 100^\circ$ is due to the increase in the thickness of the laminar boundary layer.

The first lowest point on the $\alpha_\varphi / \bar{\alpha} = f(\varphi)$ curve corresponds to the transition from laminar flow to turbulent flow ($Re_{kr} = 1 \cdot 10^5 \div 4 \cdot 10^5$) in the boundary layer.

After this, the heat transfer increases sharply. The second lowest point corresponds to the break of the turbulent boundary layer. The average heat transfer for a liquid or gas flowing across a cylinder is calculated using the following formula:

$$\bar{Nu} = (0,43 + C Re^m Rr^{0,38}) \varepsilon \quad (2.6.4)$$

The determining temperature is the temperature of the flow entering the pipe, and the determining dimension is the cylinder diameter. The correction coefficient ε takes into account the degree of turbulence of the incoming flow ($\varepsilon = 1,0\div 1,6$).

The coefficient C and the index m take the following values depending on the number Re :

$$\begin{aligned} Re=1\div 4\cdot 10^3, C=0,35, m=0,5; \\ Re=4\cdot 10^3\div 4\cdot 10^4, C=0,20, m=0,62; \\ Re=4\cdot 10^4\div 4\cdot 10^5, C=0,027, m=0,80. \end{aligned}$$

If the flow covers the cylinder at an angle $\psi < 90^\circ$, then α , calculated by equation (2.6.4), should be multiplied by $\varepsilon_\psi \approx 1 - 0,54 \cos^2 \psi$. If there is not one, but a whole set of pipes in the cross flow, the heat transfer process becomes more complicated. In this case, the heat transfer depends on the location of the pipes in the set and the row number of the pipe. Figure 2.6.2 shows the description of the fluid movement when the pipes are arranged in a strip and in a checkerboard pattern.

The first row of tubes in the stack is washed by the unstirred liquid flow and therefore α is the smallest in this row. In the subsequent rows, heat transfer is much more intense and α is almost the same for the third and subsequent rows.

For a set of pipes, the heat transfer when $10^3 < Re < 10^5$ and $0,7 < Pr < 500$ is determined from the following equation.

$$\overline{Nu} = C Re^m P_r^{1/3} \left(\frac{Pr_c}{Pr_g} \right)^{1/3} \varepsilon_s \cdot \varepsilon_i \quad (2.6.5)$$

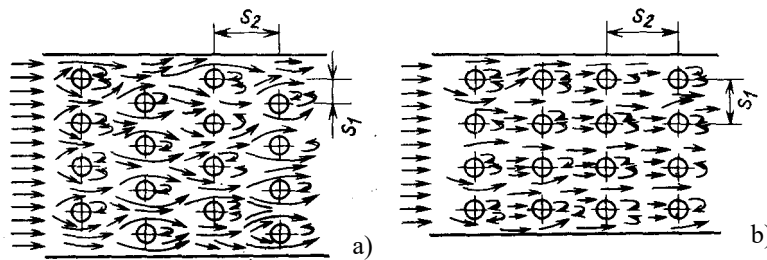


Figure-2.6.2. Description of the movement of fluid in a pipe bundle. a – checkerboard arrangement; b – strip arrangement

When the pipes are arranged in a checkerboard pattern, $s=0.41$, $m=0.65$; when arranged in a strip pattern, $s=0.26$, $m=0.65$. The outer diameter of the pipe is taken as the determining linear dimension. The Re number is calculated from the average velocity of the liquid or gas in the narrowest section of the bundle.

The correction coefficient ε_s takes into account the transverse S_1 and longitudinal pitch of the bundle:

For a chess set

$$\varepsilon_s = (S_1/S_2)^{1/6}, s_1/s_2 < 2 \text{ da.}$$

For a set with a corridor

$$\varepsilon_s = 1,12, s_1/s_2 \geq 2 \text{ da.}$$

The correction coefficient ε_i takes into account the decrease in heat transfer in the first and second rows of pipes. For the first row of pipes $\varepsilon_i=0.7$ (checkerboard pattern) and $\varepsilon_i=0.9$ (striped pattern); for the third and subsequent rows $\varepsilon_i=1$.

The average value of the heat transfer coefficient for the entire pipe set is determined from the following equation:

$$\bar{\alpha} = \frac{\sum_{i=1}^Z \bar{\alpha}_i F_i}{\sum_{i=1}^Z F_i} \quad (2.6.6)$$

where, $\bar{\alpha}_i$ – average heat transfer coefficient of the i-th row; F_i – is the surface of the i-th row; Z - is the number of pipes in the set.

2.7. Radiative heat transfer. Heat transfer radiation between surfaces

Planck's law. Absolute black bodies at temperatures above absolute zero emit light of all wavelengths into space. At the same temperature, the radiation intensity of absolute black bodies is higher than that of other real bodies. The dependence of the radiation intensity $J_{\lambda,0}$ on the wavelength λ and temperature T is determined by Planck's law:

$$J_{\lambda,0} = \frac{C_1}{\lambda^5} (e^{\frac{C_2}{\lambda T}} - 1)^{-1} \quad (2.7.1)$$

where, $C_1=0,374 \cdot 10^{-15} \text{W} \cdot \text{m}^2$; $C_2=1,4388 \cdot 10^{-2} \text{m} \cdot \text{K}$ – Planck constants; e – base of natural logarithm.

The distribution of radiation intensity by wavelength and temperature according to Planck's law is depicted in Figure 2.7.1.

Wien's law. As can be seen from Figure 2.7.1, as the temperature of a body increases, the maximum of its radiation intensity shifts to the shorter wavelength. This law was proposed by W. Wien in 1893 and gave its mathematical expression:

$$\lambda_{\max} = 0,0028989/T$$

or

$$T\lambda_{\max} = \nu = 2,898 \cdot 10^{-3} \text{m} \cdot \text{K}$$

This is called V. Wien's law of displacement. According to the law of displacement, the intensity of electromagnetic waves emitted by objects in the form of light varies at different temperatures. For example, when the temperature of an electric heater is $T=1100 \text{ K}$, it emits light with a wavelength of $\lambda_{\max}=3 \cdot 10^{-6} \text{m}$, its spectrum consists mainly of infrared light. If we take the wave spectrum of sunlight ($T=5500 \text{ K}$),

the wavelength in it corresponds to $\lambda_{\max}=5 \cdot 10^{-7}$. This wavelength corresponds to the visible part of the spectrum.

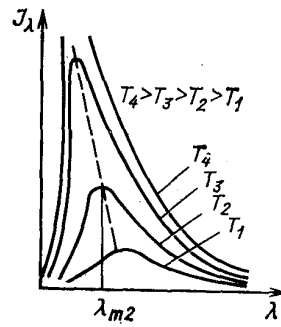


Figure-2.7.1. The dependence of the intensity of radiation from a blackbody on wavelength and temperature.

Stefan-Bolpsmann law. Based on experimental results, I. Stefan (1879) determined the following law: the radiation properties of a black body are directly proportional to the fourth power of its absolute temperature. This law was theoretically proven by A. Bolpsmann in 1884.

$$E_0 = \int_{\lambda=0}^{\lambda=\infty} J_{\lambda_1 0} d\lambda = C_0 \left(\frac{T}{100} \right)^4, \quad (2.7.2)$$

In this, S_0 – blackbody radiation coefficient. [$C_0=5,67 \text{ W}/(\text{m}^2 \cdot \text{K}^4)$]; T – body surface temperature, K.

This law can also be written in the following form:

$$E_0 = G_0 T^4, \quad (2.7.3)$$

For gray bodies:

$$E = \varepsilon E_0 = \varepsilon C_0 \left(\frac{T}{100} \right)^4 = C \left(\frac{T}{100} \right)^4 \quad (2.7.4)$$

where $G_0 = 5,67 \cdot 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K})$ – Stefan's constant; S – gray body radiation; $\varepsilon = E/E_0$ – gray matter blackness.

Kirchhoff's law. G. Kirchhoff studied the relationship between the absorption and scattering properties of heat rays of absolutely black and gray bodies in 1882 and discovered the following law:

$$\frac{E_1}{A_1} = \frac{E_2}{A_2} = \frac{E_3}{A_3} = \dots = \frac{E_0}{A_0} = E_0(T), \quad (2.7.5)$$

where $E_0(T)$ – blackbody radiation property.

The ratio of the emissivity (radiation) of a body to the absorption property does not depend on the nature of the body and is the same for all bodies at the same temperature and is equal to the radiation property of an absolutely black body at this temperature.

It can be seen from equation (2.7.5) that the greater the absorption coefficient of a gray body, the more radiation it emits. Since the absorption coefficient is less than unity ($0 < A < 1$), the radiation property of a gray body at the same temperature, E , is less than the radiation property of an absolutely black body.

From Kirchhoff's law, equations (2.7.5) and (2.7.4) it can be seen that the degree of blackness of a gray body is numerically equal to its absorption property:

$$A = \varepsilon \quad (2.7.6)$$

Since the magnitude ε is constant, the absorption properties of a gray body do not depend on wavelength and temperature.

Table 2.7.1 shows the blackness of some gray bodies. Lambert's law. The radiant energy emitted by a body is distributed in space with different intensities. The law that determines the dependence of the intensity of radiation on direction is called Lambert's law. According to Lambert's law, the amount of energy radiated in different directions from the surface of a completely black body is proportional to the cosine of the angle between the given direction and the normal to the surface of the body.

$$E_{\varphi} = E_n \cos \varphi, \quad (2.7.7)$$

where, E_n – normal radiation energy.

Table-2.7.1

The blackness of different materials

Material	t, °C	ε	Material	t, °C	ε
Aluminum (polished)	50-500	0,04-0,06	Wood	20-70	0,8-0,9
Bronze	50	0,1	Paper	20-100	0,9
Gold (polished)	200-600	0,02-0,03	Glass	20-100	0,9-0,92
Red brick	25-300	0,9	Oil paint	20-100	0,86-0,92
Gypsum	20	0,9	Lacquer	20-100	0,8-0,95
Asbestos	20-300	0,9-0,95	White enamel	20	0,9

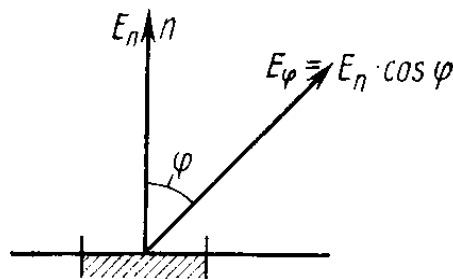


Figure-2.7.2. Radiation from an object in different directions (related to Lambert's law).

Therefore, the energy is radiated most in the direction perpendicular to the radiating surface, that is, at $\varphi=0$. As the angle φ increases, the radiation energy decreases and becomes zero at $\varphi=90^\circ$ (Fig. 2.7.2). Lambert's law is true for an absolutely black body, but for uneven gray bodies this law is true only at $\varphi=0 \div 60^\circ$.

Radiant heat transfer between solids

Parallel surfaces. It is known that all bodies at temperatures above absolute zero emit electromagnetic waves into space. Therefore, when calculating the total radiant energy of a body, the body's own radiant energy (E_1) must also be taken into account.

If radiation with energy E_2 falls on this body from other bodies, then $A_1 E_2$ energy is absorbed, $(1-A_1) \cdot E_2$ energy is returned, then ($D=0$)

$$Y_{e_{ya1}} = E_1 - A_1 E_2 = E_1 + (1-A_1) E_2 - Y_{e1} + R_1 E_2 \quad (2.7.8)$$

The quantity $Y_{e_{ef1}}$ is called the effective radiation of the body. Let us consider the heat exchange between solid bodies using the example of two parallel surfaces (Fig. 2.7.3).

Let the temperatures of these surfaces be T_1 and T_2 ($T_1 > T_2$) and the absorption coefficients A_1 and A_2 , respectively.

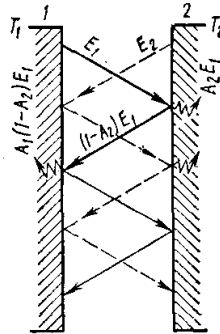


Figure-2.7.3. Scheme of radiant heat exchange between two parallel surfaces.

We choose the distance between the surfaces in such a way that the radiation from each surface reaches the other. The magnitude of the radiant heat exchange between these surfaces is equal to:

$$Y_e = E_{ef1} - Y_{ef2},$$

then

$$Y_{ef1} = E_1 + (1-A_1) E_{ef2},$$

$$Y_{ef2} = E_2 + (1-A_2) E_{ef1}.$$

We solve this system of equations with respect to Y_{ef1} and Y_{ef2} , replacing Y_{e1} and Y_{e2} with their expressions from the Stefan-Boltzmann law, and replacing A_1 and A_2 with ε_1 and ε_2 , respectively (since $A=\varepsilon$). After the substitutions, we obtain the following equation:

$$Q = EF = \frac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} C_o \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] F \quad (2.7.9)$$

where, $\theta = \left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4$ - called a temperature multiplier;

$E_k = \frac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1}$ - The degree of darkness of a system of objects is called;

$$C_k = C_o C_k = \frac{1}{1/C_o \varepsilon_1 + 1/C_o \varepsilon_2 - 1/C_o} = \frac{1}{1/C_1 + 1/C_2 + C_o} = \frac{Q}{\theta F \tau}$$

- is called The radiation coefficient given.

The given radiation coefficient represents the radiant energy transmitted from surface 1 to surface 2 in 1 s, if the temperature multiplier is 1 K and the surface area of each surface is 1 m².

Therefore, formula (2.7.9) can be written as follows:

$$Q = \varepsilon_k C_o \theta F = C_k \theta F [Vt] \quad (2.7.10)$$

Thus, to increase the intensity of radiant heat transfer, it is necessary to increase ε_k and θ , that is, the degree of blackness of the surfaces participating in the heat transfer and the temperature difference.

Radiant heat transfer between a body and its shell. In engineering, problems such as calculating radiant heat transfer when one body is located inside another body are often encountered (Figure 2.7.4).

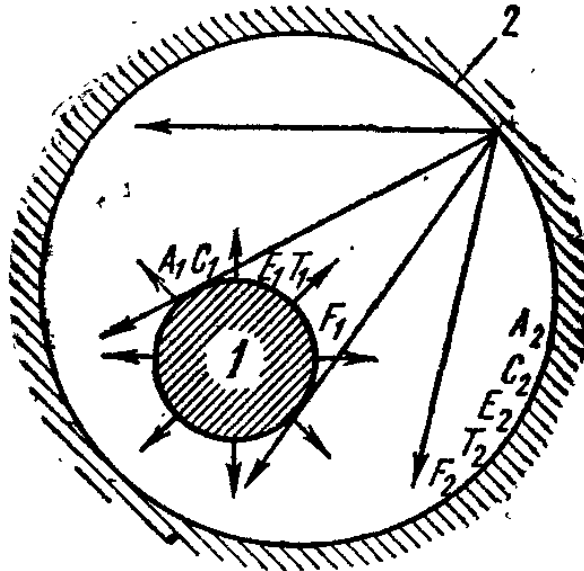


Figure-2.7.4. Scheme of radiant heat exchange between objects in a closed space.

We denote the internal body quantities by $A_1, S_1, \varepsilon_1, T_1, F_1$ and the external body quantities by $A_2, S_2, \varepsilon_2, T_2, F_2$. Unlike heat exchange between parallel surfaces, only a

part ϕ of the external body radiation falls on the internal body, the rest of the energy $(1-\phi)$ falls on the surface of the external body.

The effective radiation of the internal body consists of its own radiation and the radiation incident on the external body (which is reflected by the internal body).

$$Y_{e1ef} = E_1 F_1 + (1-A_1)\phi E_{2ef}, \quad (2.7.11)$$

The effective radiation of an external body consists of its own radiation, the energy reflected from the internal body, and the energy reflected from its own radiation.

$$E_{2\phi} = E_2 F_2 + (1-A_2)E_{1\phi} + (1-A_2)(1-\phi)E_{2\phi} \quad (2.7.12)$$

The magnitude of the heat exchange between bodies is equal to:

$$\phi = E_{1ef} - Y_{e2ef} \quad (2.7.13)$$

Solving equations (2.7.11) and (2.7.12) together and substituting Y_{e1ef} and Y_{e2ef} into the last equation, we obtain:

$$Q = \frac{1}{\frac{1}{C_1} + \frac{F_1}{F_2}(\frac{1}{C_2} - \frac{1}{C_0})} [(\frac{T_1}{100})^4 - (\frac{T_2}{100})^4] F_1 \quad (2.7.14)$$

$$\text{we mark as } C_k = \frac{1}{C_1 + \frac{F_1}{F_2}(\frac{1}{C_2} - \frac{1}{C_0})}$$

In this case, the radiant heat exchange between the body and its shell is determined by the following equation:

$$Q = C_k F_1 [(T_1/100)^4 - (T_2/100)^4] \quad (2.7.15)$$

If we replace C_k with the given blackness level of the system of objects, then equation (2.7.15) can be written as follows:

$$Q = \frac{1}{\frac{1}{\varepsilon_1} + \frac{F_1}{F_2}(\frac{1}{\varepsilon_2} - 1)} C_0 F_1 [(\frac{T_1}{100})^4 - (\frac{T_2}{100})^4] \quad (2.7.16)$$

If $F_1 \ll F_2$, then $F_1/F_2 \rightarrow 0$ and $C_k = C_1$, and the heat transfer equation becomes:

$$Q = C_1 F_1 [(T_1/100)^4 - (T_2/100)^4] \quad (2.7.17)$$

Arbitrarily located bodies. It is very difficult to derive the equation for radiant heat transfer for arbitrarily located bodies, and this problem has found its solution only for some simple cases (Fig. 2.8.5).

The radiant heat transfer for two arbitrarily located bodies can be calculated from the following formula:

$$Q = \frac{C_1 C_2}{C_0} \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \int_{F_1} dF_1 \int_{F_2} \frac{\cos \varphi_1 \cdot \cos \varphi_2}{\pi r^2} dF_2,$$

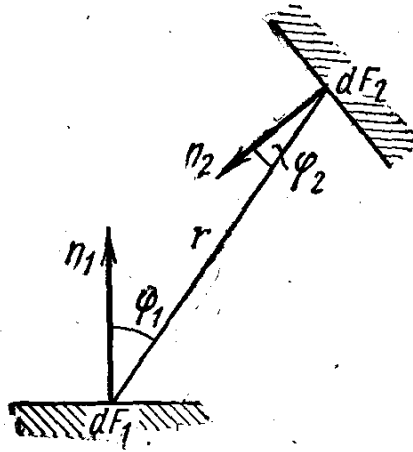


Figure-2.7.5. Radiant heat transfer for arbitrarily located bodies.

in this $\frac{C_1 C_2}{C_0}$ – The radiation coefficient of this system of objects is given;

$$\int_{F_1} dF_1 \int_{F_2} \frac{\cos \varphi_1 \cdot \cos \varphi_2}{\pi r^2} dF_2 - \text{angular coefficient of radiation.}$$

This quantity depends on the size, shape, and relative position of the radiating objects. The mathematical calculation of the angular coefficient is difficult even for simple cases, and it is mainly determined graphically. When solving technical problems, the angular coefficient is usually obtained from a table.

CHAPTER III. SOLAR COLLECTORS AND CONCENTRATORS

3.1. Flat plate solar collectors. The main elements of flat plate solar collectors

Flat plate collector - the most common type of solar collector used in domestic water heating and heating systems. This collector is a thermally insulated glass panel in which an absorber plate is placed. The absorber plate is made of a metal that conducts heat well (most often copper or aluminum). Copper is most often used because it conducts heat well and is less susceptible to corrosion than aluminum. The absorber plate is covered with a special highly selective coating that better retains absorbed sunlight. This coating consists of a very strong thin layer of amorphous semiconductor deposited on a metal base, characterized by high absorption in the visible region of the spectrum and low emissivity in the far-wave infrared region. Due to glazing (flat plate collectors usually use matte, only glass with a low iron content is used), heat loss is reduced. The bottom and side walls of the collector are covered with heat-insulating material, which reduces heat loss.

The principle of operation of a flat plate collector. Sunlight passes through the glazing and hits the absorber plate, which heats up, converting solar radiation into thermal energy. This heat is transferred to the water or antifreeze circulating through the solar collector - the coolant. The coolant is heated and then transfers its thermal energy through a heat exchanger to the water in the capacitive water heater. The hot water remains there until it is used. In addition, if the temperature drops below the set temperature (for example, due to prolonged cloudy weather), an electric heater can be installed in the capacitive water heater to heat the water to the set temperature.

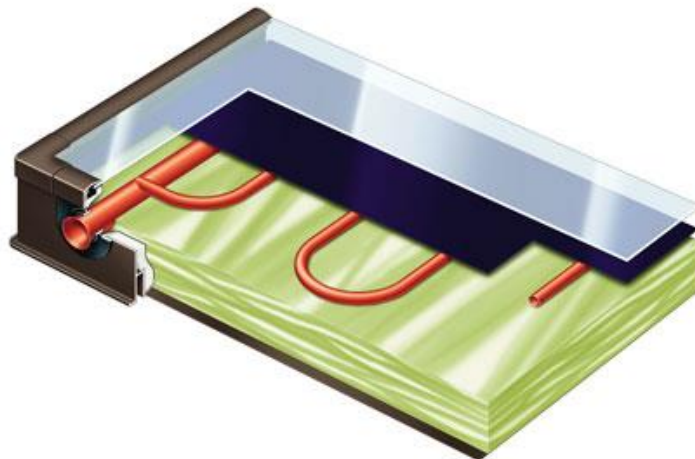


Figure-3.1.1. Schematic diagram of a flat plate solar collector.

Vacuum flat plate collector. In a vacuum water heater-collector, a volume with a dark surface that absorbs solar radiation is separated from the environment by a vacuum space. This allows almost complete elimination of heat loss to the environment due to thermal conductivity and convection. Radiation losses are mainly suppressed by the use of selective coating. Since the total loss factor in a vacuum collector is small, the coolant

in it can be heated to a temperature of 120-160 ° C. The energy absorption coefficient of such collectors, with a vacuum degree of 10^{-5} and 10^{-6} , is 98%. Insulation in the form of a vacuum avoids heat loss.

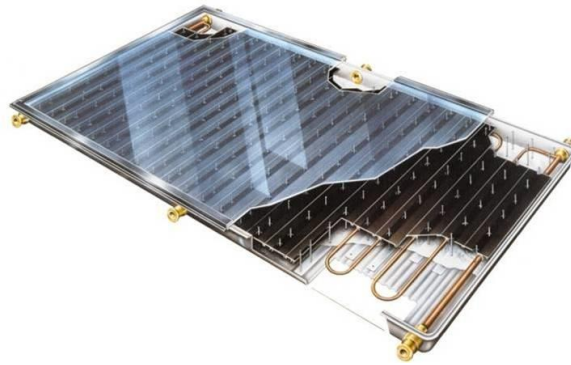


Figure-3.1.2. Vacuum flat plate collector.

The most common liquid-filled flat-plate solar collector (Figure 3.1.3) consists of a special heat-absorbing metal sheet and channels (absorber) attached to it for the circulation of the heat carrier. The absorber, which absorbs solar thermal radiation, has a selective transparent glass coating on the upper part (at several intervals). The entire structure is attached to the body and is provided with thermal insulation material on the bottom and sides.

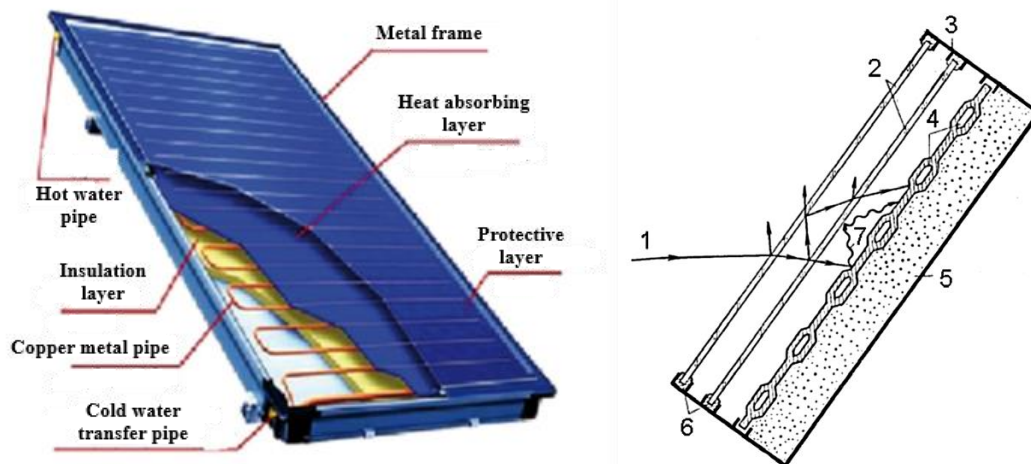


Figure-3.1.3. Structural structure of a flat-plate solar collector: 1 – Sunlight; 2 – glass shell; 3 – housing; 4 – heat-receiving surface (absorber); 5 – heat insulator; 6 – mounting; 7 – specific wavelength.

The improvement of the light transmission properties of the glass is facilitated by texturing the glass surface, reducing the iron content, and selective coatings on the glass additionally help to retain long-wave radiation. Typically, the absorber panel is made of materials with high thermal conductivity (copper, aluminum) or painted black, and is also covered with a black material with a high solar energy absorption coefficient.

If the heat carrier is in contact with the absorber, then it receives this energy. The useful energy at the outlet of the solar collector depends on 3 parameters: the incident solar radiation flux density (solar radiation) on the collector surface plane, the average

temperature of the heat carrier at the collector inlet T_{kir} . and the ambient temperature T_a .

The useful energy extracted from a unit of collector surface area over time is found from the following expression (W · hour).

$$Q_{yuza} = F_R A [I_T \tau \alpha - U_L (T_{kir} - T_a)] \quad (3.1.1)$$

Where: A – collector area (m²); F_R – heat transfer coefficient from the collector, I_T – the incident solar radiation flux density at the collector surface plane (solar radiation) W/m², τ – solar transmittance of a transparent coating, α – absorption coefficient of a transparent coating with respect to solar radiation, U_L – the total heat loss coefficient in the collector, which depends on the wind speed, the number of transparent coatings, and the properties of the insulation materials.



Figure-3.1.4. General view of a flat-plate solar collector.

Simplified version:

$$Q_{yuza} = G C_p (T_{kir} - T_a) \quad (3.1.2)$$

Here G is the specific mass flow rate of the heat carrier (kg/h), C_p is the heat capacity of the heat carrier (J/kgK). Accordingly, the useful efficiency of the solar collector is determined by the following formula:

$$\eta = \frac{Q_{yuza}}{A I_T} \quad (3.1.3)$$

The consumption capacity for hot water supply is determined as follows:

$$Q_{istermol} = G C_p (T_{chiq} - T_{kir}) \quad (3.1.4)$$

3.2. Optical characteristics of light-absorbing transparent coatings of flat-plate solar collectors

Transparent electrically conductive coatings

Transparent heat-reflecting coatings that transmit visible radiation and reflect infrared radiation are widely used in solar radiation converters. In contrast to absorbent

coatings applied to the collector surface, transparent heat-reflecting coatings are created on transparent insulation. Heat-reflecting coatings are transparent to solar radiation and reflective to infrared. Of particular interest are the idealized electrically conductive heat-reflecting coatings described by Drude, whose optical properties depend on the conductivity due to free electrons, since these coatings have high reflectivity at wavelengths smaller than the reference wavelength of plasma oscillations and with respect to long-wave radiation.

The location and sharpness of the reference are determined by the mobility and concentration of free charge carriers. Materials such as tin alloy oxides, indium oxides, mixtures of indium and tin oxides, and cadmium stannate are considered very promising for use as transparent heat-reflecting coatings in solar collectors. These coatings have thermal stability up to 300°C, good adhesion to glass, are resistant to chemically active media, and are easily applied to the transparent insulation of the collector. This coating allows a simple collector to be very efficient. The possibility of coating large areas has been created. Efforts in this area should be focused on further development of promising materials such as indium oxides and cadmium stannate mixtures, as well as on the search for new materials. In the past few years, there has been particular interest in the use of transparent electrically conductive coatings as active components of solar panels, since with the help of such solar panels it is possible to create inexpensive large-scale solar radiation converters. The reduction in the cost of such elements is associated with the transition method and the absence of the need to use special excitation layers. There is literature on elements of the ITO-Si structure with an energy conversion efficiency of 13%.

Since the technology for manufacturing these elements involves low-temperature processes, in the future it is possible to use polycrystalline silicon. Further developments should be aimed at achieving a solar energy conversion efficiency of 10%, and the structure of the oxide semiconductor - insulator - polycrystalline silicon in solar cells. To achieve high energy conversion efficiency, it is necessary to develop methods for creating very thin oxide plates of stoichiometric composition.

Removable coatings by placement method

The most promising absorber coatings for use in low-temperature solar collectors (at temperatures of 150 - 300 ° C) are nickel-zinc sulfide black nickel and black chromium coatings, which are mixtures of chromium and chromium oxides. Currently, the highest values of α and ε of black nickel coatings deposited on nickel substrates are 0.95 and 0.07, respectively. These coatings are stable in vacuum up to 300 ° C. They are also resistant to ultraviolet radiation in vacuum, but are unstable in air. However, these coatings deteriorate at high temperatures and humidity. Black chromium coatings have good optical properties ($\alpha = 0.95$ and $\varepsilon = 0.09$) and are

considered the most durable. Such a coating, applied to a nickel sublayer with a thickness of 12.7 microns, has a service life of more than 20 years. However, the nickel layer significantly increases the cost of the coating (the cost of the nickel layer is half the total cost of the coating). In order for black chrome plating to be fast, it is necessary to find a cheap replacement for the nickel layer in the future. In addition, long-term field tests of these coatings are required. The cheapest electrochemical coating is an iron oxide coating on an iron substrate. The optical properties of this coating are as follows: $\alpha = 0.85$ and $\varepsilon = 0.1$. Its strength increases significantly when organic coatings are applied, so this material is promising for use at low temperatures.

Paint coatings

Recently, special attention has been paid to dyes as inexpensive coatings for large-area low-temperature solar collectors. Selective dyes consist of particles of a semiconducting material dispersed in a thermally stable binder that is transparent in the IR region of the spectrum. The optical properties of selective dyes are determined by the optical properties of the dyes, particle size, and multiple dispersion within the dye-binder composite material. Of the dyes studied, iron-manganese-copper oxide has the best optical properties.

These metal oxide-based selective coatings have $\alpha = 0.92$ and $\varepsilon = 0.13$ with silicone rubber and $\alpha = 0.90$ and $\varepsilon = 0.10$ with polyurethane as a binder. The optical properties of selective dyes in the solar spectrum depend on the absorbance of the dye with respect to solar radiation, the thickness of the coating, and the volume concentration of the dye. Further studies are needed to optimize these parameters. Because binders absorb in the IR region of the spectrum, paint coatings are highly black. The optical properties of the pigments and binders need to be improved.

Due to organic binders, the resistance of selective coatings to environmental influences increases, but these coatings have very low mechanical strength. It is necessary to improve the adhesion of coatings to paints. Currently, very little is known about the resistance of paint coatings to atmospheric and moisture influences, as well as their resistance to thermal cycling. To find out the possibilities of using these coatings, it is necessary to conduct long-term full-scale tests of them. The technology for their production needs to be developed on a large scale. The most promising selective coatings in solar devices are paints with metal powders as a dye, but further strength tests are needed to improve their optical properties and determine their capabilities.

High temperature selective coatings

In the last few years, much attention has been paid to films made of composite materials consisting of very small metal particles embedded in insulating matrices, as

they can be used as high-temperature selective coatings. When studying the development of selective coatings of black chromium by X-ray diffraction, it was found that this material consists of polycrystals of Cr_2O_3 and amorphous silicon. At high temperatures, the optical properties of the coatings deteriorate due to the crystallization of amorphous silicon. The deterioration of optical properties is also caused by the oxidation of chromium in air at 350°C . MgO -Au cermet films are stable up to 400°C . When studying the structure of the films, it was found that both MgO and Au have the form of crystallites with sizes less than $200\text{ \AA}$. Cermets have sufficient transparency to IR radiation and low blackness due to the metal base. Coatings with $\alpha = 0.9$ and $\varepsilon = 0.1$ have the best optical properties.

Composite films $\text{Ni} - \text{Al}_2\text{O}_3$, prepared by the method of simultaneous evaporation of metal and dielectric using an electron lamp evaporator, were stable up to temperatures up to 500°C . The highest optical properties were obtained for $\text{Ni} - \text{Al}_2\text{O}_3$ films ($\alpha = 0.94$, $\varepsilon = 0.1$). Currently, experimental data on cermet films are rarely published, so they need to be further studied under operational conditions.

If the particles that make up the composite films are smaller than the wavelength of the radiation and are located at a sufficient distance from each other to ensure independent scattering, then in this case you can use the Lorentz correction for the local field and optical constants of the bulk material and apply the Maxwell-Garnet theory to explain the experimental data. The optical properties of $\text{Ni-Al}_2\text{O}_3$ and W-MgO cermet films are in good agreement with the results of calculations according to the Maxwell-Garnet theory, but the properties of the Au-MgO film are not. This deviation may be caused by the particle interaction and multiple scattering effects, the size and shape of the particles, and the relative size of the cermet components.

Magnetron sputtering

Magnetron sputtering is a promising method for creating selective surfaces. Its essence is that a magnetic field is applied parallel to the cathode surface. The magnetic field increases the ionization of the gas at a certain pressure, which leads to the appearance of a discharge at a lower pressure and, therefore, increases the speed and efficiency of the coating. Since the coating material enters the vapor phase not due to a chemical or thermal process, but due to the transmission of a mechanical impulse, almost any material can be applied using this method. In addition, the output of the sputtering is quite stable and well controlled. Pipe parts for use in a concentration collector can be coated with three layers: oxide - metal - oxide. Deposition is especially promising for creating such coatings, since it allows you to obtain a coating over very large areas at a moderate deposition rate. However, this method needs to be further developed to reduce the cost of coating large areas.

3.3. Solar concentrators and furnaces

Solar concentrators are optical systems that change the direction of sunlight and focus it on a light absorber. Concave mirrors and lenses can be used to make concentrators. However, from a technological and economic point of view, mirrors are much more preferable than lenses. For this reason, we will mainly focus on mirror concentrators (Figure 3.3.1). Let S_k be the projection of the concentrator surface onto a plane perpendicular to the sunlight, S_p be the light absorber surface, and R_{oyrn} be the specular reflection coefficient of light from the concentrator surface. Concentration occurs when $S_k > S_p$. In this case, the average flux density E_{qr} in the light absorber is:

$$E_{qr} = E_q R_{oyrn} \frac{S_k}{S_p} \quad (3.3.1)$$

is equal to the above. From this, the dimensionless average current density or concentration K is:

$$K = \frac{E_{qr}}{E_q R_{oyrn}} \quad (3.3.2)$$

or considering (3.3.1),

$$K = \frac{E_{qr}}{E_q R_{oyrn}} = \frac{S_k}{S_p} \quad (3.3.3)$$

It is agreed that the concept of concentration is denoted by S when used in relation to the entire sunspot in the light absorber, and by K when used in relation to its parts. Calculation methods for concentrators are based on the concepts of geometric optics and photometry. One of the methods for calculating the concentration level of the concentrator and the temperature generated in the focal spot is the optical-geometric method, which initially refers to the energy balance equation:

$$\Phi_p = \Phi_k \cdot \Phi_{oyrn} \quad (3.3.4)$$

where, Φ_p and Φ_k – are the light flux incident on the light absorber and the concentrator, respectively; We assume that the light absorber has a size that completely absorbs the light flux directed from the concentrator. Then the basic equation is:

$$C_m = \frac{E_{qr}}{E_q R_{oyrn}} \quad (3.3.5)$$

is in the form of. Based on this method, the average flux density E_{qr} is determined and the relationship between the concentrator size and the light absorber is established. Let us consider the relationship (3.3.4) for the paraboloid concentrator–flat light absorber (the light absorber is located in the focal plane) system.

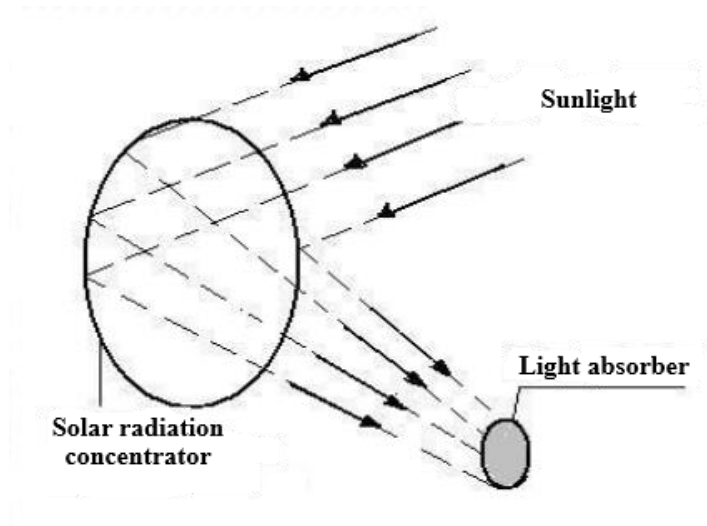


Figure-3.3.1. Operating diagram of a solar radiation concentrator.

It is known that the equation of a parabola in Cartesian coordinates with its center at the vertex of the parabola is

$$R^2 = 2pz \quad (3.3.6)$$

where, r – is the focal parameter of the paraboloid ($p = 2f$). r is the focal parameter of the paraboloid $R^2 = x^2 + y^2$; x, y, z – coordinates of the surface points; in a polar coordinate system centered at the focus F

$$\rho = p(1 + \cos U) \quad (3.3.7)$$

Let the sun's rays fall parallel to the optical axis of the parabola. It is known that the Sun is a light source with an angular radius of $\varphi_0 = 16'$ for the Earth.

Surface of a paraboloid projection $S_k = \pi R^2$

The surface of the light absorber $S_p = \pi r^2$

Also, from Figure 3.3.2 $R = \rho \sin U$

$$r = \frac{\rho \gamma}{\cos U + \gamma}$$

where, $\gamma = \varphi_0 + 2\alpha$. Putting this into (3.3.5), we obtain the following:

$$C_m = \frac{\sin^2 U_0 \cos^2 (U_0 + \varphi_0 + 2\alpha)}{(\varphi_0 + 2\alpha)^2} \quad (3.3.8)$$

Accordingly, for a parabolic cylindrical concentrator (PCC)

$$C_m = \sin U_0 \cos \left(\frac{U_0 + \varphi_0 + 2\alpha}{\varphi_0 + 2\alpha} \right) \quad (3.3.9)$$

we get the equation.

To determine the maximum concentration value C_m

$$\frac{\partial C_m}{\partial U_0} = 0$$

equation is used.

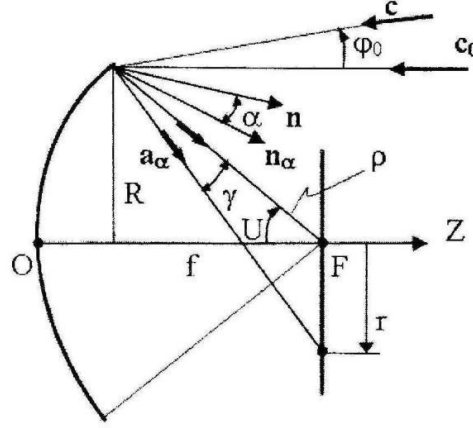


Figure-3.3.2. Scheme of a paraboloid concentrator with a flat beam absorber located in the focal plane.

As an example, let's apply it to a parabolocylindrical concentrator. For clarity, we expand the expression $\cos(U_0 + \varphi_0 + 2\alpha)$:

$$\cos(U_0 + \varphi_0 + 2\alpha) = \cos U_0 - \gamma \sin U_0$$

Differentiating (3.3.9) and noting that $U_0 > 0$,

$$\operatorname{tg} U_0 = (1 + \gamma^2)^{0,5} - \gamma$$

we get. Considering the smallness of γ , $\operatorname{tg} U_0 = 1$ or the optimal angle $U_0 = 45^\circ$. So, for a parabolocylindrical concentrator (PCC)

$$C_m = \frac{1}{2\gamma} \approx 107,5$$

Similarly, for a paraboloid concentrator (PC), $U_0 = 45^\circ$ and

$$C_m = \left(\frac{1}{2\gamma}\right)^2 \approx 11562$$

When the intensity of direct solar radiation directed at the concentrator surface is $Q_0 = 800 \text{ W/m}^2$, the intensity at the focal spot is

$$Q = C_m \cdot Q_0 = 11562 \cdot 800 \text{ Wt/m}^2 = 9,25 \cdot 10^6 \text{ Wt/m}^2$$

According to the Stefan-Boltzmann law, the radiant energy of a blackbody is directly proportional to the fourth power of its absolute temperature:

$$Q = \varepsilon \cdot \sigma_0 \cdot T^4$$

For a black body, $\varepsilon = 1$; Stefan-Bolsmann constant $\sigma_0 = 5,672 \cdot 10^{-8} \text{ W}/(\text{m}^2\text{K}^4)$;; Taking into account the above,

$$C_m \cdot Q_0 = \varepsilon \cdot \sigma_0 \cdot T^4$$

From this, the absolute temperature in the absorber

$$T = \sqrt[4]{\frac{C_m Q_0}{\varepsilon \sigma_0}} = \sqrt[4]{\frac{9,25 \cdot 10^6 \text{ Wt}/\text{m}^2}{5,672 \cdot 10^{-8} \text{ Wt}/\text{m}^2\text{K}^4}} = 3573 \text{ K}$$

It turns out that it is.

It should be noted that even in the ideal case, the temperature of the light absorber in the focal plane of the concentrators cannot be brought to the temperature of the Sun's surface, i.e. 5800 K.

According to the geometric structure of the solar-reflecting surface, concentrators can be paraboloid, parabolocylindrical, ellipsoidal and conical. However, in practice, paraboloid and parabolocylindrical concentrators are more common. The mirrors are made of glass with a highly polished surface and a thin layer of silver or aluminum, pure aluminum of the A95 brand, and even polymeric materials by thermal, mechanical and chemical treatment. The light reflection coefficient of such mirrors is about $\rho = 0.85 \div 0.90$. Solar radiation concentrators can be divided into single-mirror and multi-mirror types depending on the number of mirrors in them. Optical schemes typical for the use of single-mirror concentrators are shown in Figure-3.3.3.

To achieve a much higher operating temperature, it is necessary to use single-mirror schemes that move directly in accordance with the apparent motion of the Sun using advanced technology. Such a scheme was used by French scientists in the 8.4 m diameter “Heliodine” solar furnace built in Algeria. When using the a-scheme, the greatest efficiency is obtained from the reflecting surface of the concentrator, but there are technical limitations in its manufacture on a large scale. In the b-scheme, there is no limit to the size of the concentrator, but in this case the degree of concentration is low and the reflecting surface of the concentrator cannot be used efficiently. The c-scheme can give a good indicator in principle, but it also has the same size limitations as in the a-scheme. In practice, it is made as a system of separate concentrators that direct the concentrated radiation to a common fixed light absorber. The d-scheme is the most suitable design, since it does not require a support - a rotating device that tracks the concentrator in the visible movement of the Sun. However, in this scheme, the concentration level achieved in the operating system is low (about 3-5), and the spot of rays moves along the surface of the light absorber. These systems increase the

efficiency of low-temperature solar devices and, most importantly, increase their operating temperature (up to 100 °C). To eliminate restrictions on external dimensions and achieve high concentrations, multi-mirror optical schemes are used. A typical example of such schemes is the optical scheme of the two-mirror “Heliostat-paraboloid” system shown in Figure 3.3.4.

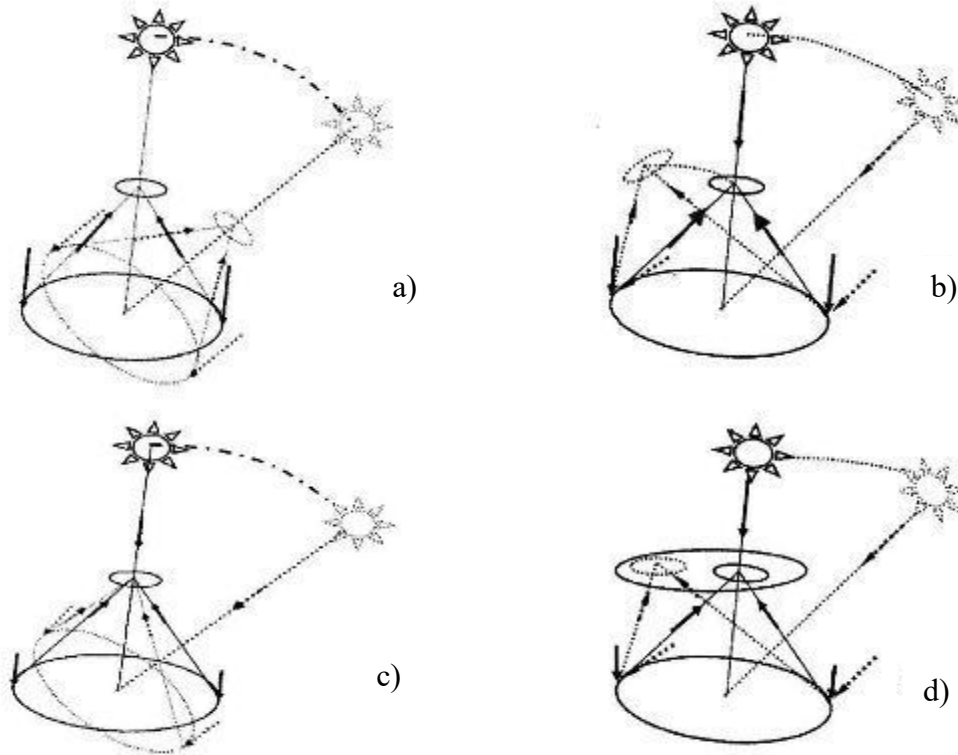


Figure 3.3.3. Optical scheme of a single-mirror solar radiation concentrator: a)–the concentrator and the light absorber are interconnected and move according to the apparent movement of the sun; b)–the concentrator is stationary, the light absorber is movable; c)–the concentrator is movable, the light absorber is stationary; d)–the concentrator and the light absorber are stationary.

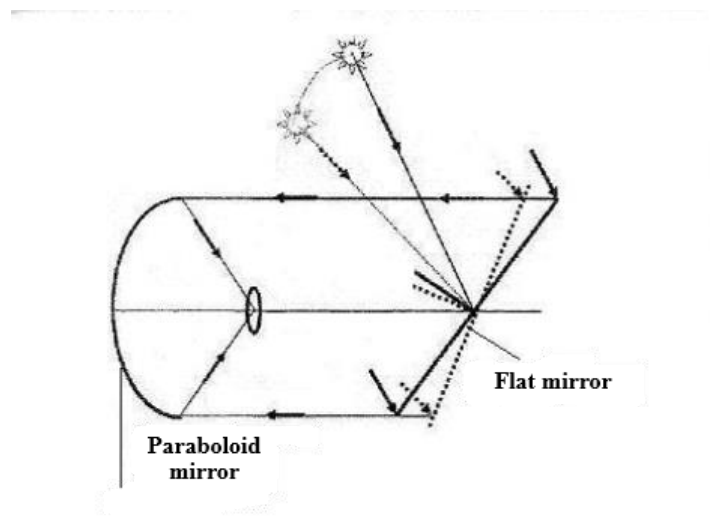


Figure-3.3.4. Optical scheme of a two-mirror solar radiation concentrator.

Concentrators can be divided into solid and faceted types. Solid-type concentrators have a smooth surface that reflects sunlight in a solid state, and the rays reflected from all its points are directed to a specific geometric area of the light absorber. It is known that the amount of energy collected in the focal plane directly depends on the dimensions of the concentrator's reflecting surface. To collect a larger amount of energy, it is necessary to make concentrators with a larger diameter. However, making large-diameter solid mirrors takes a lot of time (the bent mirror is slowly cooled in electric furnaces for several months to become optically and mechanically mature) and money. As the size of the solid concentrator increases, it becomes more difficult to install and control, stresses and deformations occur under the influence of wind, and repair work is difficult and expensive.

In faceted concentrators, the solar-collecting surface consists of a number of small elements (facets) with a single surface. In this case, all the facets return the sunlight to one geometric area of the light absorber, or individual facets or groups of facets to different geometric areas of the light absorber. The main difficulty in making faceted concentrators is to correctly position the facets and direct each facet towards the Sun, thereby focusing the reflected light.

In 1946, based on the project of F. Molaro, a researcher at the G.M. Krzhizhanovsky Moscow Energy Institute, a paraboloid concentrator with a diameter of 10 meters was put into operation on the roof of a canning factory building in Tashkent.

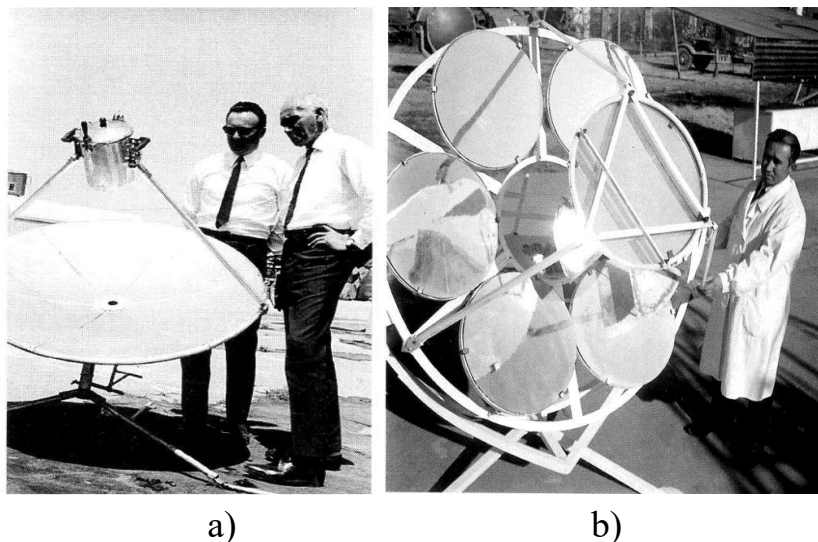


Figure-3.3.5. a) Testing of the solar kitchen (Moscow solar engineers Yu.N. Malevsky and B.A. Garf. Tashkent. 1970); b) Device for irradiating plant seeds with collected sunlight (UzASc PhTI).

The base of this solar radiation concentrator, which had no equal in the world in terms of its size and high accuracy of collecting rays, was reinforced concrete, on the surface of which a large number of glass mirror pieces were glued. The temperature at the focus of the device exceeded 1000 °C. However, due to the different coefficients of

linear thermal expansion of glass mirrors and glues, the mirrors shifted and broke from the reinforced concrete base. Taking into account such shortcomings, several faceted solar radiation concentrators with diameters of 1.50, 2 and 5 meters were built and tested at the Institute of Physics and Technology of the Academy of Sciences of Uzbekistan under the leadership of G.Yo. Umarov in 1960–1967 (Fig. 3.3.6). In these concentrators, circular, hexagonal, trapezoidal and strip-shaped facets made of glass were installed in special electric furnaces, and comparative experiments were conducted. Experiments conducted under the same conditions showed that, with the same solar radiation surface (19.63 m^2), a 5-meter diameter concentrator with 60 circular facets installed had a thermal power of 7.7 kW and a useful coefficient of performance of 0.69, while another concentrator with the same, but 6-sided facets had a power of 9 kW and a useful coefficient of performance of 0.71. Because in the first case, the useful surface was 14.25 m^2 , and in the second - 16.2 m^2 . Therefore, concentrators with 6-sided facets are technologically and economically preferable to concentrators with circular facets.

Parabolic concentrators allow you to concentrate sunlight several thousand times in the focal plane. They can be used as a separate device, as well as combined into groups to supply a single consumer. When a parabolic concentrator works separately, a light absorber is installed at its focus, which transfers energy to a heat engine (Stirling engine). In California, under conditions of solar radiation at noon, the efficiency of such a device has been achieved at more than 40 percent.

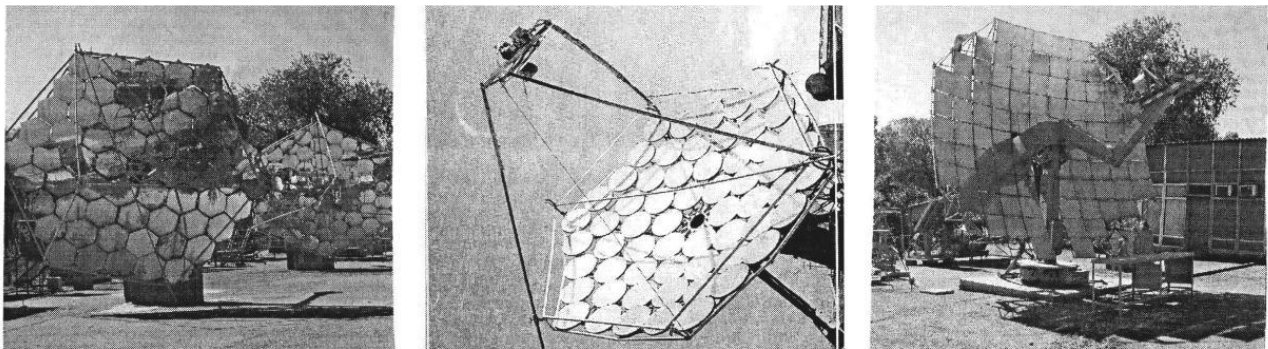


Figure-3.3.6. Faceted solar concentrators (Institute of Physics and Technology of the Academy of Sciences of the Republic of Uzbekistan).

The unit power of such devices depends on the diameter of the concentrator aperture. Currently, concentrators with an aperture diameter of up to 15 meters are used in experimental devices. Under conditions of solar radiation of 800 W/m^2 , the power of these devices reaches 50 kW.

The main drawback of these devices is that they must be constantly moved in two coordinates in accordance with the apparent motion of the Sun. At the same time, the Stirling engine is a rather complex machine, and its service personnel must have sufficient qualifications.

In the USA and Spain, there are grouped devices with parabolic concentrators. A helioradiator is installed at the focal spot of each parabolic concentrator, through which a heat carrier flows. Separate paraboloids are connected in series and parallel along the heat carrier. The heated and accumulated heat carrier transfers heat to the working body of the steam-power cycle. The efficiency of the group of devices is quite high, but at the same time there are certain difficulties associated with ensuring that the tubes through which the heat-carrying substance flows also move during the focusing of the paraboloids.

Small-diameter solar concentrators can be used for boiling water and milk, cooking, making coffee, obtaining fresh water by evaporating salt water, and pulsed irradiation of plant seeds (Figure 3.3.5), while large-diameter concentrators act as high-temperature furnaces for welding metals, melting difficult-to-dissolve substances, and other technological processes. A large amount of metal is required to manufacture concentrators, which, of course, affects their weight and price. For this reason, attention is being paid to creating concentrators using metallized lavsan film as a light-reflecting surface and polymeric materials such as foam plastic, polystyrene foam, and polyurethane foam as a base. The mass of a 2-meter diameter concentrator made of metallized polyethylene terephthalate film at the Institute of Physics and Technology of the Academy of Sciences of the Republic of Uzbekistan is only 3 kg, and in the summer the energy density in its focal plane reaches 4.3 kW/m^2 , and the temperature reaches 1700 K. Paraboloid concentrators with a diameter of 1.5–2 m can be used for brewing tea and cooking. These include “solar boilers” such as MSK-1, MSK-3, SFK-1, KS-8 and QQ-7F, created at the Institute of Physics and Technology of the Academy of Sciences of the Republic of Uzbekistan. Figure-3.3.7 shows a diagram of a “solar boiler” of the QQ-7F type.

The device is attached to a metal base and has 7 aluminum facets that move along the horizontal and vertical planes using a turning mechanism (not all facets are shown in the drawing for a clearer view of the mechanisms). The surface of the facets is 1.72 m^2 (0.246 m^2 each), all of which reflect sunlight onto the bottom of a lidded container installed in the focal plane. The device is turned towards the Sun every 10–15 minutes by means of a turning mechanism. In summer, the device allows you to boil 3–3.5 liters of water in half an hour and cook food in about 1–1.5 hours. When using one solar boiler, the amount of fuel equivalent to 350–400 kg of conventional fuel is saved annually.

The degree of solar radiation concentration of parabolic concentrators is up to 70, and the temperature of the heat carrier passing through the pipes located in its longitudinal focal plane can be raised to 400°C .

One of the largest structures built using parabolic concentrators is the water discharge device of Schuman-Boyz in the Midi settlement (Egypt) in 1913. The total

surface area of the mirror collector in the device is 1277 m², and it consists of 5 large parabolic concentrators with a length of 62.5 meters and a width of 4.1 meters.

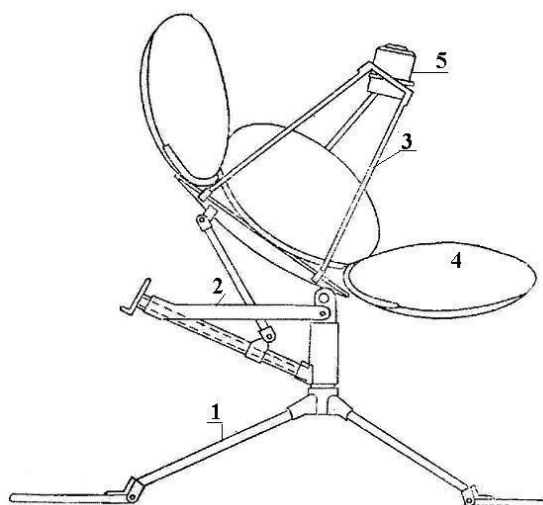


Figure 3.3.7. Scheme of a QQ-7F type “solar boiler”: 1–three-legged base; 2–sun-orientation mechanism; 3–container handle; 4–aluminum facets; 5–container (boiler, metal kettle, pan, etc.).

The production and use of high-density flows using solar furnaces for high-temperature research is a new scientific direction founded in the 20th century. When carrying out technological processes in solar furnaces, the operating temperature is about 3000 K.



Figure 3.3.8. Parabolosylindrical concentrator (Institute of Physics and Technology of the Academy of Sciences of the Republic of Uzbekistan).

Solar furnaces are usually built on the basis of a two-mirror scheme consisting of a “paraboloid mirror + heliostat” (Fig. 3.3.4). In this case, the working area at the focal point has a fixed location, which makes it convenient to work or conduct experiments. The solar furnace is of great importance as a highly integrated radiation device for scientific research and technology.

The French scientist Felix Tromb and his students have made great contributions to the creation of solar furnaces. They built a large solar furnace with a thermal power of 1000 kW in Mont-Odeillo (Eastern Pyrenees).

In 1981–1987, a large solar furnace was built in Uzbekistan (Parkent district of Tashkent region) based on a similar thermal power, but technologically and optically much improved project.

This unique facility, built 50 km from Tashkent, at a latitude of $41^{\circ}20'$, at an altitude of 1050 m above sea level in the Tien Shan mountain range, is a complex high-precision optical-energetic device with an automatic control system, the main part of which is a field of heliostats and a paraboloid concentrator that form a highly concentrated light flux in the focal area of the technological tower (Fig. 3.3.9).

The field of heliostats is located in a checkerboard pattern on a hill with a slope of 13° from the horizon and consists of 62 heliostats that continuously move in accordance with the apparent movement of the Sun and illuminate the entire mirror surface of the concentrator throughout the day. All heliostats have the same design and dimensions. The reflecting surface of each heliostat is flat, measuring 7.5 m x 6.5 m (48.75 m^2) and consists of 195 flat mirrors-facets measuring 0.5 m x 0.5 m with a thickness of 6 mm. The total number of facets is 12,090. The reflecting layer of the facets is formed by vacuum sputtering of aluminum. The facets are protected on the back side with acrylic paint of the EM AK-5164 brand.

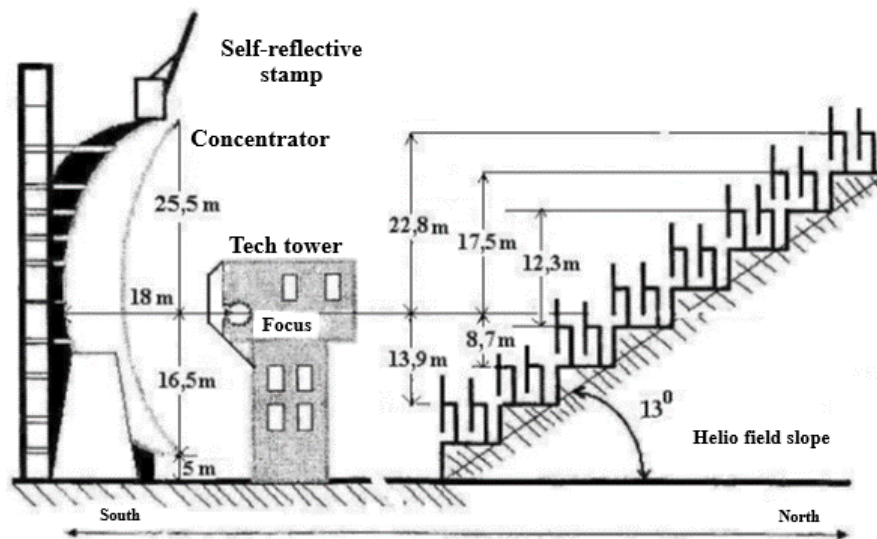


Figure-3.3.9. Optical scheme of the Large Solar Furnace.

The sun's rays falling on the reflecting surfaces of the heliostats with an area of 3022.5 m^2 are directed towards a paraboloid concentrator with an area of 1840 m^2 , built facing north. The paraboloid concentrator ensures that these rays are collected in a highly concentrated flow in a focal area with a diameter of 1 meter (the focal area is located 18 meters above the level of the building foundation).

The technological tower where the focal area is located is equipped with special equipment for studying physicochemical processes occurring in substances under the influence of high temperatures. The area where the Large Solar Furnace is located is distinguished by its clear atmosphere and a large number of sunny days (250–270 days). Since 1977, regular measurements of the direct solar radiation density (W/m^2) have been carried out here using the actinometric method.

The automatic control system of the heliostats of the Large Solar Furnace (ACSH) ensures that the heliostats are automatically oriented towards the Sun. The concentrators are aligned using an Obscure camera. To align the heliostats, a reflection mark and 195 red projectors are installed on the top of the concentrator. This method allows the heliostat facets to be quickly inspected and aligned.

Table 3.3.1 presents the heating temperature of a black body located directly in the center of the concentrator at different values of the radiation flux density.

The Large Solar Furnace has the following important advantages:

- It is an extremely clean heating source, with no foreign substances mixed into the heated material;
- The polychromatic spectrum in the focal spot ensures that the radiation is absorbed by the material 10 times more than monochromatic laser radiation;
- The material heating rate is high ($100\text{--}800\text{ }^\circ\text{C/min.}$);
- The openness of the thermally treated area of the material allows the use of additional technological methods (processing in an ultrasonic or electromagnetic field, tempering, etc.);
- No special requirements are imposed on the electromagnetic properties of materials;
- The fact that the radiation flux density in the center of the focal spot reaches 1300 W/cm^2 makes it possible to work with almost all high-temperature materials in an air environment;
- The formation of quasi-uniform radiation in the focal spot and the ability to control the flux density serve to implement fast, flexible technological processes.

This scientific and technological center was the first to discover the phenomenon of threshold condensation of vapors under free cooling conditions after melting high-temperature materials, and to study the dependence of the condensation temperature on the heating regime.

A new, world-renowned direction in high-temperature spectrophotometry - the “probe flash method” - was developed, which allows measurements to be made to determine the spectral optical properties of modern materials at high temperatures. The Large Solar Furnace has become a modern scientific center where scientific research is conducted in international scientific cooperation with scientists from different countries.



Figure-3.3.10. General view of the Large Solar Furnace and its heliostat field (Parkent district, Tashkent region).

For many years, extensive scientific research and practical work has been carried out here to study the physical and chemical properties of materials, especially refractory and refractory materials, by means of high-temperature thermal treatment under the influence of sunlight. In the center of the focal spot, the solar radiation flux density reaches 1300 W/cm^2 , and the temperature reaches 3500°C . During the day, 80–90 kg of aluminum oxide (AlO , melting point 2050°C), 40–50 kg of spinel alloy ($\text{AlO}+\text{MgO}$, melting point 2350°C), 5–7 kg of stable zirconium oxide (ZrO , melting point 2500°C) can be melted in the focal area. The device ensures extremely high purity of the melt, since there are no fuel or carbon electrodes, which are the main reason for the poor purity of the product, as in conventional furnaces. In recent years,

more than 160 oxide materials have been synthesized here, which are notable for their low thermal expansion and electrical conductivity, as well as fire resistance. Currently, such materials are used in space and in extremely sensitive technological processes that require high purity. The device also serves as a giant telescope for astrophysical research at night.

Table-3.3.1

The radiant flux density at the focal plane of the concentrator and the temperature reached during heating

S/n	Average value of current density at the Earth's surface, W/m^2	Luminous flux density in the focal plane of the concentrator, kW/m^2	The heating temperature of the substance located in the focal plane of the concentrator, °C;	
			Absolute solid	Aluminum oxide
1	600	6560	2081	1539
2	650	5690	2254	1667
3	700	6120	2430	1798
4	750	6560	2601	1924
5	800	7000	2775	2053
6	850	7440	2948	2181
7	900	7880	3122	2310
8	950	8310	3295	2438
9	1000	8780	3469	2567

The “Physics-Sun” Scientific and Production Association of the Academy of Sciences of the Republic of Uzbekistan, which has a large facility like the Large Solar Furnace, has accumulated extensive experience in creating and operating mirror systems that concentrate sunlight, non-contact temperature measurement, and studying the thermoradiative properties of modern materials at high and ultra-low temperatures.

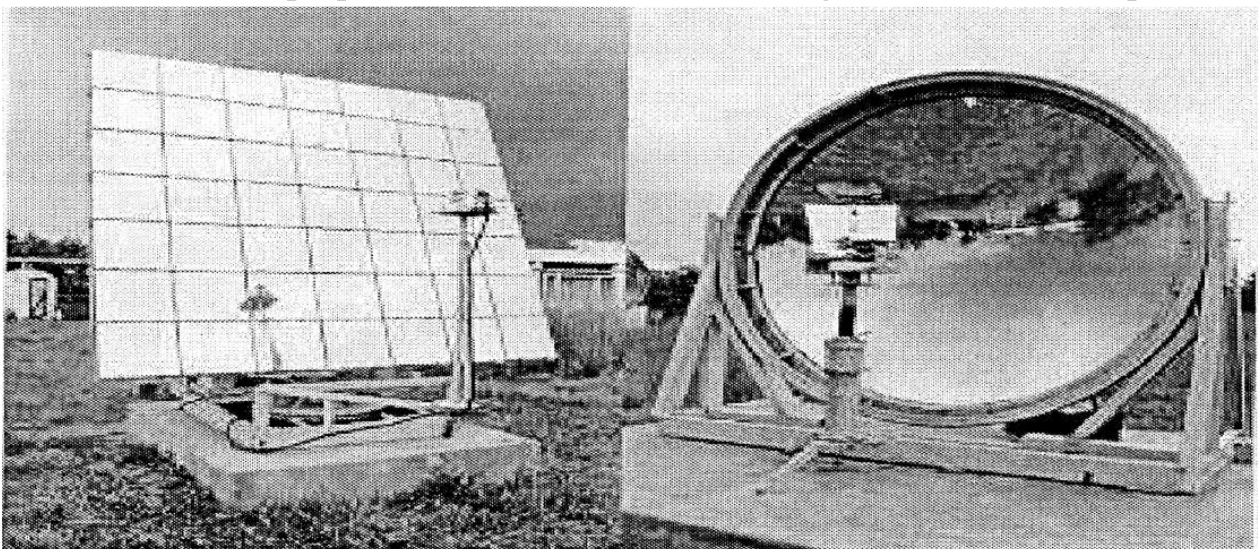


Figure-3.3.11. Small Solar Oven with 1500 W thermal power (India, Hyderabad).

Based on many years of research conducted here, a Small Solar Furnace with a thermal power of 1500 W was created for conducting scientific research in heliomaterials.

In accordance with agreements between the countries in the field of trade, economic and scientific and technical cooperation, such furnaces were installed in 1999 at the International Center for Powder Metallurgy in Hyderabad, India, and in 2000 at the Tabbin Institute of Metallurgy in Egypt (Figures-3.3.11 and 3.3.12). These devices consist of a single flat heliostat measuring 2.8 m x 2.8 m, a solid parabolic concentrator with a diameter of 2 m, a solar sensor and additional measuring instruments, as well as the necessary equipment for its operation.

The heliostat moves during the day in accordance with the invisible movement of the Sun and directs the sunlight falling on its surface onto a parabolic mirror. The temperature in the working area of the focal plane of the concentrator is above 2500 °C.

Such conditions are sufficient for processing high-temperature materials. For example, it is possible to synthesize zirconium oxide with a melting temperature of 2500 °C. Also, the institute's specialists performed dimensional and energy calculations for the

Small Solar Furnace built at the Cologne Institute of Aerodynamics in Germany, and collaborated on the creation and preparation of a secondary concentrator.



Figure 3.3.12. Small Solar Oven (Egypt, Cairo).

3.4. Heat losses to the environment from flat-plate solar collectors

Thermal collectors available on the market today are almost exclusively manufactured with a transparent cover or even without a cover. The heat transfer coefficient (U_f) between the absorber and the ambient air through the collector front

cannot be set as constant, since the absorber temperature covers a range significantly larger than the typical temperatures in buildings. Therefore, for each given absorber temperature, it is necessary to iteratively calculate the convective and radiative heat transfer coefficients and determine the U_f - value of the front. The heat transfer coefficients through the back U_b and side panels U_c of an insulated collector can be assumed to be constant and can be calculated from the layer thickness s [m], the thermal conductivity $1/h_a$ [W/mK] and from the outside, the thermal resistance between the insulating material and the environment $1/h_a$ [m²K/W].

$$U_b = U_s = \left(\frac{s}{\lambda} + \frac{1}{h_a} \right)^{-1} \quad (3.4.1)$$

The external thermal resistance consists of a ratio of radiation-dependent and convective, wind-speed-dependent ($1/h_a = 1/(h_c + h_r)$). It is quite sufficient to calculate using the standard value of external thermal resistance $1/h_a = 0.04$ m²K/W, which is common in construction. If all heat transfer coefficients are calculated from the absorber relative to the ambient temperature, then the parallel heat transfer coefficients can be added to the total loss coefficient U_t . The losses through the collector sides with a small side panel surface A_s are here related to the aperture surface of the collector A .

$$U_t = U_f + U_b + U_s \frac{A_s}{A} \quad (3.4.2)$$

Heat transfer coefficient of transparent cover U_f

The heat transfer coefficient of a transparent cover is the sum of the thermal resistances between the absorber and the environment. The thermal resistances between the absorber and the glazing R_{a-g} and the glazing and the environment R_{g-o} are calculated based on the parallel heat transfers for convection h_c and radiation h_r .

$$R_{a-g} = \frac{1}{h_{c,a-g} + h_{r,a-g}} \quad R_{g-o} = \frac{1}{h_{c,g-o} + h_{r,g-o}} \quad (3.4.3)$$

$$U_f = \frac{1}{R_{a-g} + R_{g-o}} \quad (3.4.4)$$

The temperature-dependent heat transfer coefficients are calculated first by taking into account the cover plate temperature T_g . Q_f passes through the collector front as a heat flux between the absorber and the environment,

$$\frac{Q_f}{A} = U_f (T_a - T_o) \quad (3.4.5)$$

equal to the heat flow between the absorber and the cover,

$$\frac{Q_{a \rightarrow g}}{A} = (h_{c,a-g} + h_{r,a-g})(T_a - T_g) \quad (3.4.6)$$

By equating the heat flows, the temperature of the new cover glass can be calculated.

$$\frac{Q_f}{A} = U_f (T_a - T_o) = (h_{c,a-g} + h_{r,a-g})(T_a - T_g) = \frac{Q_{a \rightarrow g}}{A}$$

$$T_g = T_a - \frac{U_f(T_a - T_o)}{(h_{c,a-g} + h_{r,a-g})} \quad (3.4.7)$$

The equations for the required heat transfer coefficients are discussed below.

Heat transfer coefficient for radiation h_r

Using the Stefan-Boltzmann law, the radiant power from a diffusely radiating black surface A_1 [m²] with emissivity $\varepsilon = 1$ is

$$Q_1 = A_1 \sigma T_1^4 \quad (3.4.8)$$

is emitted into the room, the Stefan-Boltzmann constant $\sigma = 5.67 \times 10^{-8}$ W/m²K⁴. Part Φ_{12} of this radiant power is absorbed by a second black surface A_2 , randomly placed in the room. A_2 also emits radiation for its part depending on the temperature T_2 , while the first surface absorbs part Φ_{21} with $Q_2 = A_2 \sigma T_2^4$. The net radiation exchange Q between surfaces A_1 and A_2 is the difference between the radiation emitted to the other surface and the radiation returned to it.

$$Q = Q_{1 \rightarrow 2} - Q_{2 \rightarrow 1} = \underbrace{\frac{\Phi_{12} A_1 \sigma T_1^4}{A_1 \text{ dan } A_2 \text{ ga radiatsion oqim}}}_{\text{A}_1 \text{ dan } A_2 \text{ ga radiatsion oqim}} - \underbrace{\frac{\Phi_{21} A_2 \sigma T_2^4}{A_2 \text{ dan } A_1 \text{ ga radiatsion oqim}}}_{\text{A}_2 \text{ dan } A_1 \text{ ga radiatsion oqim}} \quad (3.4.9)$$

Since both surfaces are defined as black emitters with absorption and emission coefficients of 1, no reflection of radiation occurs at the surfaces.

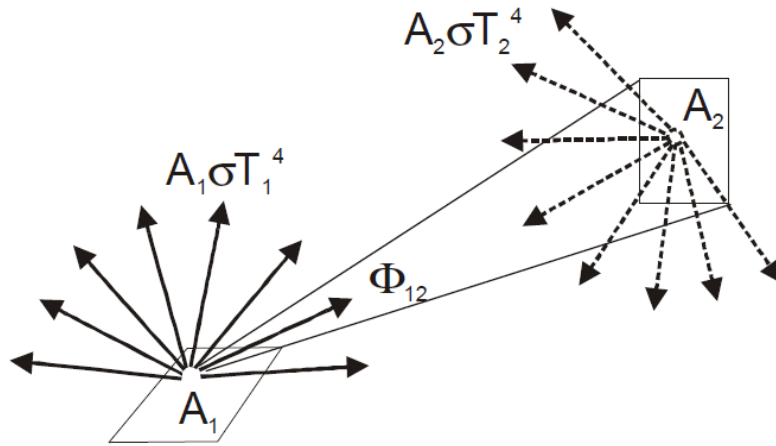


Figure-3.4.1. Radiation flux from a small (differential) two-dimensional element A_1 to a surface element A_2 at temperature T_1 , which absorbs a proportion Φ_{12} and in turn emits radiation.

If the two surfaces are at the same temperature, the net radiative heat transfer is zero, and the reciprocal conditions for the form factors Φ_{12} and Φ_{21} are obtained:

$$\Phi_{12} A_1 = \Phi_{21} A_2 \quad (3.4.10)$$

The heat flux resulting from radiation exchange between two black emitters is expressed as:

$$Q = \Phi_{12} A_1 \sigma (T_1^4 - T_2^4) = \Phi_{21} A_2 \sigma (T_1^4 - T_2^4) \quad (3.4.11)$$

The form factors Φ_{ij} , as the geometric ratio of the radiation-receiving surface A_j to the entire hemisphere on which the power is radiated, are usually complex functions of the surface geometry. In cases where only the radiation emitted by the second surface can be absorbed, it is easy to determine the radiation numbers. Two cases are of particular importance for calculating the heat transfer coefficients in solar thermal collectors; the radiation exchange between two plane-parallel plates (absorber and transparent cover) and the radiation exchange between a flat cover and the celestial hemisphere. If the parallel plates are assumed to be infinitely extended, then the entire radiant flux of surface A_1 is absorbed by surface A_2 and vice versa, i.e., $\Phi_{12} = \Phi_{21} = 1$.

In the radiation exchange of a flat surface with the celestial hemisphere, the radiation emitted by the surface A_1 is completely absorbed by the celestial hemisphere, i.e. $\Phi_{12} = 1$. On the contrary, the ratio of the small surface A_1 to the total radiation of the celestial hemisphere is very small. Based on the reciprocal condition in the above equation, here for $A_1 \ll A_2$ is $\Phi_{21} = A_1/A_2 \approx 0$.

Typically, surfaces have an emission or absorption coefficient less than one (gray emitters), so the radiation received from the second surface with $(1 - \varepsilon_2) = (1 - \alpha_2)$ is partially reflected back and can in turn be absorbed by the surface, A_1 with ε_1 and reflected back with $(1 - \varepsilon_1)$, etc. To create a radiation energy balance for surface A_1 , it is necessary to subtract from the amount of radiant energy Q_1 the radiation $Q_{1 \rightarrow 2 \rightarrow 1}$ reflected by surface A_2 and absorbed by A_2 . In addition, the radiation $Q_{2 \rightarrow 1}$ emitted by surface A_2 and absorbed by A_1 should be considered as an energy gain. This balance should be considered for the simple case of plane-parallel plates with form factor 1. For radiation of intensity Q_1 emitted by surface A_1 , the following amount of heat is returned:

$$\begin{aligned}
 \frac{Q_{1 \rightarrow 2 \rightarrow 1}}{Q_1} &= \underbrace{(1 - \varepsilon_2)}_{\text{First reflection from } A_2} \times \underbrace{\varepsilon_1}_{\text{absorption } A_1} \\
 &+ \underbrace{(1 - \varepsilon_2)(1 - \varepsilon_1)}_{\text{First back reflection from } A_1 \text{ to } A_1} \times \underbrace{(1 - \varepsilon_2)\varepsilon_1}_{\text{Second reflection from } A_2 \text{ absorbed by } A_1} \\
 \frac{\dot{Q}_{1 \rightarrow 2 \rightarrow 1}}{\dot{Q}_1} &= \underbrace{(1 - \varepsilon_2)}_{\text{first reflection from } A_2} \times \underbrace{\varepsilon_1}_{\text{absorption at } A_1} + \underbrace{(1 - \varepsilon_2)(1 - \varepsilon_1)}_{\text{first back reflection from } A_1} \times \underbrace{(1 - \varepsilon_2)\varepsilon_1}_{\text{second reflection from } A_2 \text{ absorbed by } A_1} \\
 &+ (1 - \varepsilon_2)(1 - \varepsilon_1)(1 - \varepsilon_2) \underbrace{(1 - \varepsilon_1)}_{\text{second reflection from } A_1} \times \underbrace{(1 - \varepsilon_2)\varepsilon_1}_{\text{third reflection from } A_2 \text{ absorbed by } A_1} + \dots \\
 &= (1 - \varepsilon_2)\varepsilon_1 \left(1 + (1 - \varepsilon_2)(1 - \varepsilon_1) + ((1 - \varepsilon_2)(1 - \varepsilon_1))^2 + \dots \right) = \frac{(1 - \varepsilon_2)\varepsilon_1}{1 - (1 - \varepsilon_2)(1 - \varepsilon_1)}
 \end{aligned}$$

Furthermore, surface A_2 emits an intensity of Q_2 , its surface A_1 absorbs a ratio of ε_1 and reflects $(1 - \varepsilon_1)$. After several reflections, an infinite sequence of radiation absorbed by A_1 leads to:

$$\frac{Q_{2 \rightarrow 1}}{Q_2} = \varepsilon_1 + (1 - \varepsilon_1)(1 - \varepsilon_2)\varepsilon_1 + \dots = \frac{\varepsilon_1}{1 - (1 - \varepsilon_1)(1 - \varepsilon_2)} \quad (3.4.12)$$

Thus, the energy balance of the surface A_1 results in the radiation exchange between the plane-parallel plates with $A_1 = A_2$ and $\Phi_{12} = \Phi_{21} = 1$:

$$\begin{aligned} Q_{1,net} &= Q_1 - Q_{1 \rightarrow 2 \rightarrow 1} - Q_{2 \rightarrow 1} = Q_1 - Q_1 \frac{(1 - \varepsilon_2)\varepsilon_1}{1 - (1 - \varepsilon_2)(1 - \varepsilon_1)} - Q_2 \frac{\varepsilon_1}{1 - (1 - \varepsilon_2)(1 - \varepsilon_1)} = \\ &= \frac{\frac{Q_1 \varepsilon_2 - Q_2 \varepsilon_1}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1 \varepsilon_2}}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1 \varepsilon_2} = \frac{1}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_1} - 1} \sigma A (T_1^4 - T_2^4) \end{aligned} \quad (3.4.13)$$

This heat flow equation is linearized by taking the temperature difference $T_1 - T_2$, thereby determining the heat transfer coefficient for radiation h_r .

$$Q = h_r A (T_1 - T_2) \quad (3.4.15)$$

$$h_r = \frac{\sigma}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} (T_1^4 + T_2^4)(T_1 + T_2) \quad (3.4.16)$$

To calculate the radiation exchange between the absorber at temperature T_a and the glazing at temperature T_g , $T_1 = T_a$ and $T_2 = T_g$ are set.

Selective coating

The long-wave emission coefficients of black absorbing paint are typically 95%, while those of glass are 88%. If a solar-absorbing coating is applied to a substrate with a low emissivity, such as metal, the emission can be reduced. Galvanizing metals with black chromium is increasingly being replaced by environmentally friendly cathodic sputtering methods, which consume only one tenth of the energy required for galvanizing. The emission level can reach 5%, while the absorption of short-wave radiation remains at 95%.

If one or more of the form factors Φ_{ij} is not equal to unity, the energy balances must be modified accordingly. This can be analyzed using the example of the radiation exchange of the surface with respect to the celestial hemisphere.

The radiation $Q_1 = \varepsilon_1 \sigma A_1 T_1^4$ emitted by surface A_1 is incident on the celestial hemisphere with a form factor of $\Phi_{12} = 1$ and is reflected by $(1 - \varepsilon_2)$. Now, only a part of the radiation reflected from the sky reaches surface A_1 , i.e. $\Phi_{21} = A_1/A_2$. Thus, the infinite sequence of radiation reabsorbed by A_1 is transformed as follows:

$$\begin{aligned} \frac{Q_{1 \rightarrow 2 \rightarrow 1}}{Q_1} &= \underbrace{(1 - \varepsilon_2)}_{1. \text{ refl. form } A_2} \underbrace{\frac{A_1}{A_2} \varepsilon_2}_{1. \text{ abs. from } A_1} + \\ & \underbrace{(1 - \varepsilon_2)}_{1. \text{ refl. form } A_2} \underbrace{\left(1 - \frac{A_1}{A_2} \varepsilon_1\right)}_{1. \text{ refl. from } A_1} \underbrace{(1 - \varepsilon_2)}_{2. \text{ refl. from } A_2} \frac{A_1}{A_2} \varepsilon_1 + \dots = \frac{(1 - \varepsilon_2)\varepsilon_1 \frac{A_1}{A_2}}{1 - (1 - \varepsilon_2)\left(1 - \frac{A_1}{A_2} \varepsilon_1\right)} \end{aligned} \quad (3.4.17)$$

Similarly, the radiation emitted from A_2 at A_1 is modified by the factor A_1/A_2 .

$$\frac{Q_{2 \rightarrow 1}}{Q_2} = \frac{\varepsilon_1 \frac{A_1}{A_2}}{1 - \left(1 - \frac{A_1}{A_2} \varepsilon_1\right)(1 - \varepsilon_2)} \quad (3.4.18)$$

Thus, the net radiant flux of surface A_1 is expressed as:

$$Q_{1,net} = \frac{1}{\frac{A_1}{A_2 \varepsilon_2} + \frac{1}{\varepsilon_1} - \frac{A_1}{A_2}} \sigma A_1 (T_1^4 - T_2^4) \quad (3.4.19)$$

from which equation (3.4.13) follows as a special case for equal surfaces $A_1 = A_2$. If surface A_1 is much smaller than surface A_2 , equation (3.4.19) simplifies to:

$$Q_{1,net} \approx \varepsilon_1 \sigma A_1 (T_1^4 - T_2^4) \text{ for } A_1 \ll A_2 \quad (3.4.20)$$

If the sky temperature is determined by T_{sky} ($T_2 = T_{sky}$), the result for the heat transfer coefficient between the coating ($T_1 = T_g$) and the sky is:

$$h_r = \sigma \varepsilon_1 (T_g^2 + T_{sky}^2) (T_g + T_{sky}) \quad (3.4.21)$$

The sky temperature takes into account the transparency of the atmosphere, i.e. the back radiation lost in the wavelength range of approximately 8 to 14 μm . The sky temperature can be calculated from the dew point temperature T_{dp} [K]:

$$T_{sky} = T_o \left(0.8 + \frac{T_{dp} - 273}{250} \right)^{0.25} \quad (3.4.22)$$

As a simpler equation, without calculating the partial pressure of water vapor and the dew point temperature, the modified Swinbank equation [K] can be used (Fuentes, 1987):

$$T_{osmon} = 0.037536 T_o^{1.5} + 0.32 T_o \quad (3.4.23)$$

In order to be able to calculate heat flows as usual with the temperature difference between the cover T_g and the ambient temperature T_o , the heat transfer coefficient for radiation is normalized to this temperature difference.

$$h_r = \sigma \varepsilon_1 (T_g^2 + T_{osmon}^2) (T_g + T_{osmon}) \frac{(T_g - T_{osmon})}{(T_g - T_o)} \quad (3.4.24)$$

Convective heat transfer coefficients h_c

Convective heat transfer occurs as natural convection between the absorber and the transparent cover, and as forced convection between the cover and the environment by wind forces. The characteristic nominal value for convective heat transfer is always the Nüsselt number Nu , from which the convective heat transfer coefficient can be calculated as a function of the characteristic length L and the thermal conductivity λ of the air.

$$Nu = \frac{h_c}{\lambda} \quad (3.4.25)$$

Free convection in a stable air layer

In a stationary air layer between flat parallel plates, the plate spacing d is used as the characteristic length L to calculate h_c . To calculate the Nüsselt number, a set of empirical correlations is available for flat film collector geometries with a temperature gradient ΔT between the absorber and the cover and different collector angles β . Based

on Duffy and Beckmann (1980), the following equation can be used to sum inclination angles up to 75°; for inclination angles greater than 75°, the 75° function value is retained.

$$Nu = 1 + 1.44 \left[1 - \frac{1708(\sin 1.8\beta)^{1.6}}{Ra \cos \beta} \right]^+ \left[1 - \frac{1708}{Ra \cos \beta} \right]^+ + \left[\left(\frac{Ra \cos \beta}{5830} \right)^{1/3} - 1 \right]^+ \quad (3.4.26)$$

The plus sign of the parenthetical term means that only positive results should be used; with negative parenthetical terms, the term is zero. The Rayleigh number characterizes the rise due to thermally induced density changes and is given by the product of the Grashof (Gr) and Prandtl (Pr) numbers:

$$Ra = GrPr = \frac{g\beta'\Delta TL^3}{v^2} \times \frac{vpc_p}{\lambda} = \frac{g\beta'\Delta TL^3}{va} \quad (3.4.27)$$

where, g : Gravitational constant [m/s^2];

$\beta' = 1/T$: Coefficient of volumetric expansion of ideal gases [K^{-1}];

ΔT : Temperature difference between panels [K];

L : Characteristic length, where panel separation d [m];

v : Kinematic viscosity [m^2/s]

$a = \lambda/\rho c_p$: Thermal conductivity [m^2/s].

Forced convection by wind

The convective heat flow in the collector glass cover is mainly caused by wind, i.e. forced convection. The ratio between the panel and the ambient temperature is smaller than that of free convection. A good approximation is given by the following superposition of the two heat transfer coefficients:

$$h_c = \sqrt[3]{h_{c,w}^3 + h_{c,free}^3} \quad (3.4.28)$$

The forced convection transfer coefficient $h_{c,w}$ is calculated from the Nüsselt correlation for a plate with flow parallel to a plate with a turbulent boundary layer (VDI, 1994).

$$Nu_{turb} = \frac{0.037Re^{0.8}Pr}{1+2.443Re^{-0.1}(Pr^{\frac{2}{3}}-1)} \Rightarrow h_{c,w} = \frac{Nu_{turb}\lambda}{L} \quad (3.4.29)$$

The Reynolds number Re is derived from the wind speed v_w and the plate length L $Re = v_w L / v$ under parallel flow conditions. The ambient air temperature v , ρ , λ , c_p is used as the material values. The heat transfer coefficient $h_{c,w}$ is also calculated with the panel length L as the characteristic length. To calculate the free convection ratio, the glass temperature T_g and the ambient temperature T_o must be known.

$$h_{c,free} = 1.78(T_g - T_o)^{1/3} [Wm^{-2}K^{-1}] \quad (3.4.30)$$

The simplified approach only considers wind speed, so,

$$h_{c,w} = 4.214 + 3.575v_w \quad (3.4.31)$$

Example-3.4.1.

Calculation of the U -value U_f of collectors with absorber ($\varepsilon = 0.1$) and without selective coating ($\varepsilon = 0.9$) under the following boundary conditions:

Absorber temperature: $T_a = 70^\circ\text{C}$

Ambient temperature: $T_o = 10^\circ\text{C}$

Wind speed: $v_w = 3 \text{ m/s}$

Emissivity of transparent cover: $\varepsilon_g = 0.88$

Absorber-glass at plate distance: $d = 2.5 \text{ cm}$

Collector slope angle: $b = 45^\circ$

The panel temperature adopted for the first iteration: $T_g = 40^\circ\text{C}$

first iteration:

The radiation heat transfer coefficient $h_{r,a-g}$ between the absorber and the glass is $0.79 \text{ W/m}^2\text{K}$ for the absorber with the selected coating and $6.44 \text{ W/m}^2\text{K}$ for the black absorber. The convective heat transfer coefficient of the standing air layer is determined by the material values of the average air temperature $(70 + 40)/2 = 55^\circ\text{C}$ ($\lambda = 0.0286 \text{ W/m K}$, $\nu = 1.85 \times 10^{-5} \text{ m}^2/\text{s}$, $\rho = 1.045 \text{ kg/m}^3$, $c_p = 1.009 \text{ kJ/kg K}$). The Prandtl number is 0.71 and the Grashof number is 43787 , the average temperature difference between the absorber and the received panel temperature is 30K , so the Rayleigh number is 31089 . From this, the Nusselt number is 2.78 and the convective heat transfer coefficient $h_{c,a-g} = 2.9 \text{ W/m}^2\text{K}$.

Between the glass cover and the sky, according to Swinbank, the radiant heat transfer coefficient $h_{r,g-o} = 7.23 \text{ W/m}^2\text{K}$ at a sky temperature of 269 K is obtained, and the simplified Equation (3.83) for the convective heat transfer, which depends on the wind speed, is $h_{c,w} = 14.9 \text{ W/m}^2\text{K}$. From this, after the first iteration, the heat transfer coefficient of the transparent cover is obtained

$$U_f = \frac{1}{\frac{1}{2.9 + 0.79} + \frac{1}{14.9 + 7.23}} = 3.17 \text{ Wm}^{-2}\text{K}^{-1}$$

for the selectively coated absorber, and $6.57 \text{ W/m}^2\text{K}$ for the black absorber. With these values, the new panel temperature is calculated:

$$T_{g,new} = 70^\circ\text{C} - \frac{3.17 \text{ Wm}^{-2}\text{K}^{-1} \times 60^\circ\text{C}}{2.9 \text{ Wm}^{-2}\text{K}^{-1} + 0.79 \text{ Wm}^{-2}\text{K}^{-1}} = 18.6^\circ\text{C}$$

or 27.8°C for a non-selective absorber. The temperature is clearly lower than the initially set temperature of 40°C . With this temperature, the heat transfer coefficients are calculated again. After the iteration is complete, the following values are obtained:

Table-3.4.1

Results obtained

Parameters	Selective coating	Non-selective
$h_{r,a-g}$	$0.72 \text{ Wm}^{-2}\text{K}^{-1}$	$6.08 \text{ Wm}^{-2}\text{K}^{-1}$
$h_{c,a-g}$	$3.4 \text{ Wm}^{-2}\text{K}^{-1}$	$3.2 \text{ Wm}^{-2}\text{K}^{-1}$
$h_{r,g-o}$	$11.88 \text{ Wm}^{-2}\text{K}^{-1}$	$8.3 \text{ Wm}^{-2}\text{K}^{-1}$

$h_{c,w}$	$14.9 \text{ Vt} \cdot \text{m}^{-2} \text{K}^{-1}$	$14.9 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$
T_g	18.0°C	27.1°C
U_f	$3.56 \text{ Vt} \cdot \text{m}^{-2} \text{K}^{-1}$	$6.64 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$

3.5. Absorption of light in different materials

When light passes through a medium, it is always partially absorbed, this absorption is associated with the conversion of radiant energy into heat and other types of energy. Substances that absorb light weakly are called transparent substances, and substances that absorb it strongly are called opaque substances. However, this is a relative characteristic of substances, since transparency depends not only on the nature of the substance, but also on the thickness of the layer of this substance. In fact, for example, aluminum is not transparent, while water is considered transparent. However, a very thin layer of aluminum transmits light well, while a thick layer of water is practically not transparent (in the deep layers of the ocean there is practically complete darkness).

We know that if we say that the wavelength is λ , then the speed of light in a plane wave is $c = \lambda/T$, and assuming that $2\pi/T = \omega$, then $c = \lambda\omega/2\pi$. This is the wave equation of light.

$$x = A \sin 2\pi(t/\tau - y/\lambda) = A \sin 2\pi(yt - y/\lambda) = A \sin(\omega t - 2\pi y/x) \quad (3.5.1)$$

is expressed in the form of. Since the passage of a light wave occurs along with the vibrations of the particles of the medium, the energy of the vibrations of the medium also increases along with the wave. The amount of energy carried by the wave through a unit of surface perpendicular to the light per unit of time is the wave intensity, and the concept of light intensity J is introduced to quantitatively assess the process.

The intensity of light (or the energy density of the luminous flux) is the amount of energy carried through a surface of 1 m^2 perpendicular to the light beam for 1 sec. Let a parallel beam of monochromatic rays fall perpendicularly on the surface of an absorbing substance, that is, a clean glass or film plate (Fig. 3.5.1).

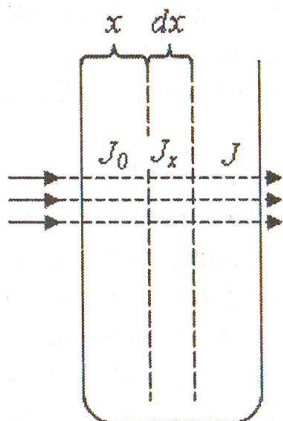


Figure-3.5.1

The change in light intensity dJ_x in an infinitely thin layer inside a glass or film plate at a distance dx from its surface x is proportional to the thickness of the layer and the intensity of the light incident on the layer:

$$dJ_x = -kJ_x dx \quad (3.5.2)$$

where k is the absorption coefficient characterizing the absorption capacity of the material and does not depend on the thickness of the layer. The minus sign dx indicates the

attenuation of light in the layer. We transfer $(dJ_x < 0)J_x$ to the left side of equation 3.5.2 and integrate the resulting expression over the entire thickness of the plate a (from $x = 0$ to $x = a$ and from $J_x = J_0$ to $J_x = J_1$):

$$\int_{J_0}^{J_1} \frac{dJ_x}{J_x} = -k \int_0^a dx \quad (3.5.3)$$

Then the equality $\ln J/J_0 = -ka$ follows or $J = J_0 e^{-ka}$. Here, J_0 is the intensity of light incident on the plate, J is the intensity of light transmitted through the plate (found by the French physicist Bouguer in 1799). From the formula (3.5.3): when $A = 1/k$, it follows that $J_0/J = e \approx 2.72$.

Thus, the absorption coefficient of a substance is the inverse of the thickness of a layer of this substance, and when passing through this layer, the light intensity weakens by e times (approximately three times).

The values of the absorption coefficient for different substances are different: for example, the value of k at normal pressure is 10^{-3} m^{-1} for air, 1 m^{-1} for glass, and 10^6 m^{-1} for metals. From this it follows that to attenuate the light intensity by a factor of 3, a metal layer with a thickness of $a = 1/10^6 \text{ m} = 1 \text{ } \mu\text{m}$, a glass layer $a = 1/1 \text{ m} = 1 \text{ m}$, and air layer $a = 1/10^{-3} \text{ m} = 1 \text{ km}$.

In general, any substance has a more or less selective absorption ability, that is, the value of the absorption coefficient depends on the wavelength of light. For example, glass, water vapor, film, condensation on the surface strongly absorb infrared rays. Ordinary glass transmits visible light well, but significantly weakens infrared rays (wavelength $\lambda > 2 \mu\text{m}$) and almost completely absorbs ultraviolet rays ($\lambda < 0.38 \mu\text{m}$). The leaves of living plants strongly absorb the entire range of the visible spectrum, except for the green ($0.52 \mu\text{m} < \lambda < 0.60 \mu\text{m}$) and dark red ($\lambda > 0.70 \mu\text{m}$) regions: such absorption is caused by the pigment chlorophyll (green leaf) contained in the leaves.

Light filters - glass plates and gelatin films mixed with a certain dye - work on selective absorption. A light filter transmits a certain part of the spectrum, absorbing all the rest. The phenomenon called the greenhouse effect is also based on selective absorption. The soil in a greenhouse is usually heated from three heat sources: the first is visible light that freely passes through the windows of the greenhouse frames or the surface of the polyethylene film, the second is biofuel, and the third is special (water or electric) heating systems. The temperature of the heated soil is not very high, it is in the form of infrared radiation. However, as we have already mentioned, the glass absorbs infrared radiation. Due to this, a significant part of the energy radiated by the soil is converted into heat and serves to heat the greenhouse. Thus, the glass protects the greenhouse from being wasted by long-wave radiation. In recent years, transparent polymer - polyamide films have been used instead of glass to cover greenhouses and greenhouse limonaries with transparent surfaces. Unlike glass, this film transmits not

only visible light, but also ultraviolet rays and strongly absorbs infrared rays (90%). At the same time, it is very elastic, lightweight, sufficiently strong and durable. These qualities of the film allow the construction of temporary structures that protect some rare plants.

The greenhouse effect under polyamide film is stronger than under glass. The use of polyamide films (and some other transparent polymers) allows you to significantly increase the yield of agricultural crops and improve the quality of the crop. The effect of light on any substance, including its physiological effect, is associated with the absorption of light.

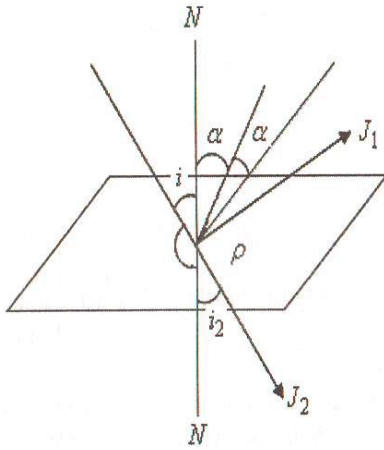
Of the processes that occur on Earth under the influence of light, the most important is undoubtedly photosynthesis. Photosynthesis is the conversion of inorganic substances into organic substances.

Photosynthesis occurs under the influence of sunlight absorbed by chlorophyll, which releases gaseous oxygen. Photosynthesis is a complex oxidation-reduction reaction, which can be written without intermediate links in the form of such a conditional equation:



According to estimates, terrestrial and aquatic plants on Earth produce about 450 billion tons of organic matter annually through photosynthesis. Photosynthesis produces organic matter, purifies the atmosphere from carbon dioxide and enriches it with oxygen. In this way, photosynthesis creates the necessary conditions for life on our planet. Light also affects animal organisms. In particular, it affects their visual organs and the skin layer. Visible infrared rays mainly heat the surface of the skin. Ultraviolet rays cause a photochemical reaction in the outer layer, as a result of which a brown pigment is formed. "Burning" is a pigment called melanin, which strongly absorbs ultraviolet rays and protects the body from their extraneous effects. The weak (normal) effect of ultraviolet rays has a beneficial effect on the body: it increases its resistance to infectious diseases and improves metabolism. Light is a necessary life factor for humans, animals, and plants, because its deficiency or absence disrupts the normal functioning of their bodies, and the lack of light cannot be compensated for by any other influences (heating, nutrition, and the like).

Mechanical and optical properties of glass and polymer films. The performance of solar devices is determined by the absorption of the radiant flux by transparent surfaces. The radiant flux depends on the illumination of the receiver and the transparency of the device. Typically, a transparent surface can consist of one or more layers. The optical characteristics of transparent surfaces (polymer film, glass, etc.) depend on their structural structure, layers.



3.5.2-rasm

When the layer is one layer, part of the light flux incident on it is scattered by this layer (Fig. 3.5.2), part is absorbed, and the rest passes through the transparent surface. The flow of parallel rays incident on a transparent surface can be considered as a second source of light. The luminous flux J characterizes the radiation in a given direction. The light incident on a transparent surface is scattered at an angle to the direction of incidence, and a certain part is transmitted straight through. In such a process, the characteristic of the

radiation is determined by the light intensity. Usually, a transparent surface is wavy, and part of the parallel rays

incident on this surface are scattered, and the rest are absorbed and spread apart. As a result of the incidence of light in a given direction, the coefficient of clarity depends on the scattering base, the polarization of the transparent surface, the spectral composition of the light, the angle of incidence, and the azimuth.

The brightness B of light passing through a transparent surface and its scattering ρ from the surface are expressed as follows.

$$\left. \begin{aligned} \tau &= \frac{B_1}{B_0} \\ \rho &= \frac{B_2}{B_0} \end{aligned} \right\} \quad (3.5.4)$$

B_1 and B_2 are the energy brightness of the light transmitted and scattered by the transparent surface. B_0 is the brightness of the light that is only scattered or transmitted by the transparent surface.

$$B_0 = \frac{E}{\pi} \quad (3.5.5)$$

where, E - indicates the energetic illumination of a clear surface.

T, t and $\bar{B}$ are the total coefficients of direct and scattered light transmission through a transparent surface. A is the light absorption coefficient. R, τ and ρ are the coefficients expressing the total value of the light scattered from transparent surfaces and the coefficients expressing its direct and scattered part. These quantities are related to each other as follows.

$$\left\{ \begin{aligned} T + A + r &= 1 \\ T + t + \bar{B} \\ R &= \tau + \rho \end{aligned} \right. \quad (3.5.6)$$

Based on the Second Law of Thermodynamics, A.A. Gernin relates these phenomena as follows.

$$\begin{cases} t_D = t_{-D} \\ \overline{B}_D = \overline{B}_{-D} \\ T_D = T_{-D} \end{cases} \quad (3.5.7)$$

$$\begin{cases} \overline{B}_{ij} = \overline{B}_{ji} \\ t = t_{-i} \\ \rho^{ij} = \rho_{ji} \end{cases} \quad (3.5.8)$$

i, j and d – are indexed quantities. i - characterizes light incident at an angle. j - characterizes light scattered at an angle. d - characterizes ideal radiation.

If two of these letters are used in the index of a symbol, then in the first case it characterizes the incidence of rays on a clear surface, and in the second case the indices of two are used to characterize the scattered light. One side of a clear surface is called the front side, the other side is called the back side. The minus sign (-) characterizes the incidence of rays not on the surface, but on the back side.

(Formulas (3.5.7) and (3.5.8) show that the light transmission coefficient of a transparent surface does not depend on which side of the transparent surface the scattered part of the solar radiation falls on.

3.6. Thermal calculation of high-temperature solar devices

There are various types of concentrators that allow you to collect sunlight, and in college conditions there is an opportunity to design, manufacture and test parabolic-cylindrical concentrators.

Taking into account that the power of this type of concentrators is capable of boiling water, the technological processes of manufacturing and testing a solar water heater and heater were developed and tested. The main parts of the device are a surface that directs light back to the focus and a boiler placed at the focus. As is known from the mathematics course, the equation of a parabola with a given focus is as follows:

$$y=4Fx \quad (3.6.1)$$

in this, F - focal length.

When designing, the focal length can be selected taking into account the rotation of the device and the convenience of placing the boiler.

To construct a graph of the function using the above formula, a table is prepared showing the values of y and x . Based on the obtained values, a graph is drawn on millimeter paper in centimeters on a scale of 1:1. The resulting graph is transferred to wooden planks and a rail is placed along the curve, forming a base. The planks are sawn and placed parallel to each other. Thin tin or fiberboard is installed along the torus and parabola. An aluminum foil is placed on the resulting parabolic-cylindrical surface. The boiler is made of copper, aluminum or nickel-plated pipe.

To install the boiler along the focus of the parabola, supports are installed on a box with a reflecting surface. Fingers are installed on the supports for installing and removing the boiler. The boiler is filled with water and placed on the fingers and the concentrator is directed towards the sun. The water is boiled. To obtain hot water, the water flow to the boiler is controlled by a special tank and a tap from it. An insulated tank accumulator is used to store hot water.

Thermal calculation of a concentrated solar water heater. Solar energy falling on the earth's surface arrives as direct and diffuse radiation. Considering that concentrators operate mainly on the basis of direct radiation, we consider the thermal calculation of the device. The direct solar radiation that can fall on the territory of Uzbekistan is given in the table in W/m^2 . The figures given in Table 3.6.1 are the average values of many years of observations and may differ from the actual ones. Given that the error does not exceed 5-7%, this information can be used in practical calculations.

Table-3.6.1

The correct solar radiation that can fall on the territory of Uzbekistan

Months	Hours of the day						
	12	11, 13	10, 14	9, 15	8, 16	7, 17	6, 18
December	840	800	745	640	370	-	-
January, November	860	840	780	675	460	-	-
February, October	910	890	840	760	600	180	-
May, September	930	910	880	830	705	550	500
April, August	940	935	910	865	780	630	300
May, June	940	930	910	865	800	620	465
July	930	925	910	865	800	700	520

To calculate the heat output of a concentrator solar system, we use the following flow chart:

- 1) The energy of light incident on a perpendicular surface is:

$$Q_1 = J_1 S \tau \quad (3.6.2)$$

- 2) Light energy reflected from a reflecting surface:

$$Q_k = K_F J_F S \tau \quad (3.6.3)$$

- 3) Light energy absorbed by the receiving surface:

$$Q_{yu} = K_2 Q_k \quad (3.6.4)$$

- 4) The amount of light energy absorbed by the water in the boiler:

$$Q_{ol} = mc(t_2 - t_1) \quad (3.6.5)$$

- 5) The light energy that destroys the boiler:

$$Q_{yoiq} = K_3 S_2 (t_k - t_1) \quad (3.6.6)$$

6) Concentration level:

$$n = \frac{S_1}{S_2} \quad (3.6.7)$$

7) Device Efficiency

$$\eta = \frac{Q_f}{Q} \cdot 100\% = \frac{mc(t_2 - t_1)}{J_f S \tau} \quad (3.6.8)$$

Physical quantities included in the formulas.

J_l - The energy of light incident on a unit surface perpendicular to the light (W/m²);

K_f - surface reflectance (unitless);

S_l - concentrator surface (m²); K_2 - light absorption coefficient of the boiler (unitless);

m - The amount of water in the boiler (kg); C - specific heat capacity of water -4200 $\frac{J}{kggrad}$; t_2 - hot water temperature (degrees);

t_1 - ambient temperature (degrees); K_3 - boiler heat transfer coefficient (W/m²degrees) in practical calculations, it can be assumed to be 5-8 W/m² degrees;

S_{kc} - surface of the pot (m²); S_2 - cross-sectional area of the boiler along the axis (m²);

t_q - boiler surface temperature (degrees).

3.7. Vacuum tube collectors

Solar water heating (collector) devices are divided into 2 types: integral and separate types of construction. The integral type of collector (monoblock) consists of a system of vacuum tubes, a tank (thermos) - a hot water reservoir, as well as a galvanized metal frame attached to a single structure. The monoblock collector is mainly installed on the roof of a house or building and provides the necessary hot water pressure to the source of consumption.

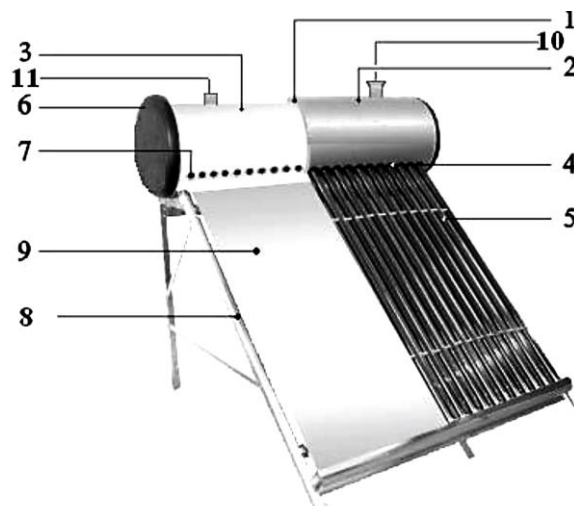


Figure-3.7.1. Vacuum tube collector of the entire type: 1 – Water tank; 2 – outer layer of the tank; 3 – inner layer of the tank; 4 – external fastener; 5 – vacuum tubes; 6 – water tank covers; 7 – rubber fastener; 8 – support frame, material – galvanized steel or stainless steel; 9 - reflective plate – additional option; 10 – emergency air valve; 11 – controller sensor.

Circulation inside the tank is carried out due to natural processes. The set also includes a frame-support system, a smart controller, an electromagnetic valve and an electric heater. 2 types of tanks are offered for selection: simple and with a coil heat exchanger. The efficiency of the collector equipped with a coil heat exchanger is $\sim 30\%$ higher than that of the simple one, since at the moment of heating the flowing water passing through the heat exchanger, the hot water in the tank is also heated by solar energy. Depending on the level of consumption, the volume of water in the tank is 100 l, 150 l, 200 l, 250 l, 300 l.

The tank accumulator consists of 3 layers:

1. The inner part of the tank is made of stainless steel grade M-304, which ensures its high safety in hygienic terms, as well as resistance to corrosion, and durability in long-term practical use.

2. The middle layer of the tank consists of high-quality polyurethane insulation with a thickness of 55 mm, which has a high accumulation function, ensuring long-term storage of hot water temperature. In winter, when the ambient temperature is below 0°C , the total heat loss is $\sim 3\text{--}6^{\circ}\text{C}$. For example, if the water temperature in the collector is $+60^{\circ}\text{C}$ in the evening, in the morning this temperature indicator decreases by 5°C , that is, it is $+55^{\circ}\text{C}$.

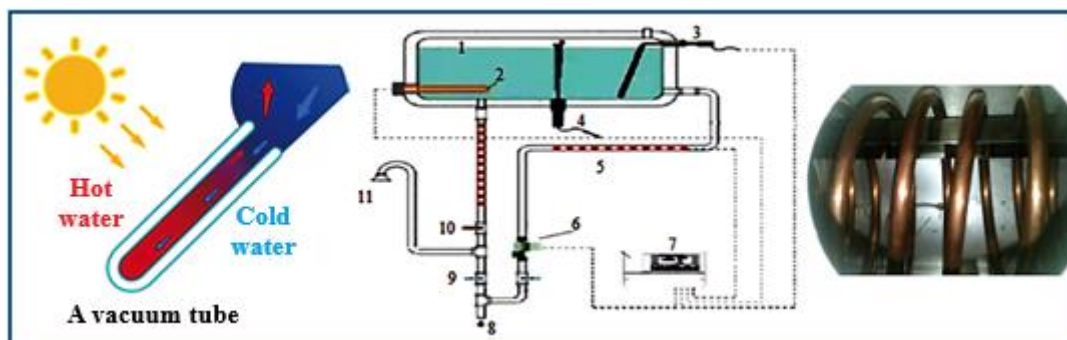


Figure-3.7.2. Important components of a vacuum tube collector: 1 – water tank, 2 – electric heater, 3 – high level sensor, 4 – low level sensor, 5 – heating cable, 6 – solenoid valve, 7 – smart controller, 8 – cold water inlet, 9 – cold water tap, 10 – hot water tap, 11 – consumer.

3. The outer metal casing of the tank is specially coated to protect it from external influences (sunlight, precipitation, i.e. snow, rain, hail). The remaining parts are made of rubber and plastic, taking into account external influences. Vacuum flasks are made of durable borosilicate tempered glass with a light-absorbing layer, which converts solar radiation into thermal energy and heats the water.

Due to natural circulation, the water heated in the tube rises and accumulates in the tank. The smart controller controls all the working processes of the collector (water temperature in the tank, water level in the tank, the operating mode of the electromagnetic valve for filling the tank with water, adding and removing a 1.5 kW

heater when necessary) (Figure 3.7.3). Using this collector, you can save 100% of the energy used to heat water for 9 months.



Figure 3.7.3. Technical characteristics of a solar evacuated tube collector with a water capacity of 150 l.

Outer shell of the tank: Painted steel 0.4 mm.

Inner shell of the tank: SUS 304–0.5 mm stainless steel.

Vacuum tubes: 58 mm / 1800 mm.

Frame: Galvanized steel – 1.5 mm.

Thermal insulation material – Polyurethane.

Insulation thickness: 50 mm.

Frame inclination angle: 35–45 degrees.

Tank diameter: 375 mm / 475 mm.

Table-3.7.1

Characteristics of vacuum tube 58mm/1800mm

Composition	Concentric all-glass solar tubes
Length	1800 ± 5 mm
Outer diameter of the flask	58 ± 0,7 mm
Thickness of the outer glass of the flask	1,8 ± 0,15 mm
Inner diameter of the flask	47 ± 0,7 mm
Thickness of the inner glass of the flask	1,6 ± 0,15 mm
Glass material Borosilicate glass	3.3
Absorbent coating performance	
Inner lining of the flask	Single layer or three layers
Composition of the three-layer coating of the vacuum flask	Selective solar absorption coating: composite copper - stainless steel - aluminum - CU/SS- ALN(H)SS/ALN(L)/ALN
Spray (launch) method	DS jet spray
Absorption rate	>91%
Loss of solar radiation	<8% (80 °C ± 1,5 °C)
Vacuum level	$P \leq 5 \times 10^{-3}$ Pa
Max. temperature	270-300 °C

Nominal pressure	0,6 MPa
Average heat loss coefficient	$\leq 0,6 \text{ W}/(\text{m}^2 \text{ } ^\circ\text{C})$
Resistance to precipitation	<35 mm
Resistance to overheating	300 °C
Working process at low temperatures	0 °C – 10 °C
Expiration date	~ 15 yıl

Separate type solar evacuated tube collector

In a separate type of solar system, the solar collector is installed on the roof of the building, and the storage tank is installed inside a separate building. Thus, the collector and the tank are separate. The water is heated by transferring the temperature of the heat carrier (antifreeze, propylene glycol) to the water through a heat exchanger located in the tank, which is moved to the collector by a pump.



Figure-3.7.4. A separate type of solar evacuated tube collector.

The tank performs the function of a thermos, maintaining the temperature of boiling water for a long time. Its water volume is determined based on consumer demand. In addition, the components of the device include a smart controller, an electric heater, a pump and heating devices. A special type of design of solar collectors can be used as a hybrid system, in which solar collectors are combined with gas or electric boilers.

The efficiency of solar collectors depends on the solar radiation power falling on the collector surface unit, the ambient temperature and the temperature of the heat carrier passing through the collector.

In our experiments, the efficiency of a vacuum solar collector is equal to the ratio of the useful thermal energy received from the device to the solar radiation power falling on the collector surface unit:

$$\eta_{v.t.k} = \frac{Q_k}{I_T F_k} \quad (3.7.1)$$

The useful energy obtained from a solar collector, taking into account the effects of heat losses and the optical efficiency, can be written as follows:

$$Q_k = I_T \cdot (\tau\alpha)F_k - U_k F_k (T_k - T_a) \quad (3.7.2)$$

Based on relationships (3.7.1) and (3.7.2), the expression for calculating the efficiency of a solar collector can be written as follows:

$$\eta_{v.t.k} = (\tau\alpha) - \frac{U_k(T_k - T_a)}{I_T} = G \cdot C_p \cdot (T_k - T_{ch}) \quad (3.7.3)$$

where, I_T – Solar radiation flux density falling on the collector absorber area m^2 ; $(\tau\alpha)$ – the effective optical efficiency of the collector, i.e. τ – light transmission coefficient of a vacuum tube; α – absorption capacity of the absorber; F_k – collector area; U_k – total heat loss coefficient in the collector; T_k – inlet temperature of the heat carrier; T_a – ambient temperature; G – Heat carrier mass flow rate (kg/h), C_p – heat capacity of the coolant (J/kgK); T_{ch} – temperature of the coolant at the outlet.

In many cases, the efficiency of different types of solar collectors is estimated by the sum of heat loss coefficients. Some literature states that for unglazed solar collectors $U_k \approx 21 \text{ W}/(m^2 \cdot K)$, for glass-covered flat solar collectors $U_k \approx 4 \text{ W}/(m^2 \cdot K)$, and for vacuum-type solar collectors $U_k \approx 1.5 \text{ W}/(m^2 \cdot K)$.

3.8. Air heating collectors

Air heating collectors are solar collectors that use air as a heat carrier. They are currently widespread in the USA and, to a lesser extent, in central Europe. Air collectors are used much less often for water heating. The largest air heating solar collector in Europe is located in Oschatz near Leipzig and has an area of 1175 m^2 . It is designed to heat a warehouse for finished products and building materials.

Air heating collectors are best used to produce hot air (not hot water). Examples of air heating collectors:

- Buildings with air heating systems, such as gyms, warehouses, workshops, buildings with high outdoor air consumption and residential buildings. Partly because of the introduction of low energy consumption standards in Western Europe, the share of heat consumption for air heating in the ventilation system has become more significant than the total heat consumption. A system used in combination with controlled ventilation and an air collector can cover a significant part of the required heat. It is certainly impossible to provide full heat in winter with the help of air heating solar collectors, due to the unfavorable relationship between the amount of incoming solar energy and the amount of heat required for heating.

- Buildings for drying agricultural and industrial products, as well as grain, seeds, medicinal and herbal plants, wood and construction materials. The drying potential of an air-heating solar collector is approximately 0.2 to 0.7 kg of water evaporation per hour per 1 m^2 of collector surface.

- In solar cooling systems.

It should be noted that air-heated collectors are less common than liquid-heated collectors, but they have significant advantages over liquid-heated collectors:

- Air heating collectors do not freeze in winter;
- There is no risk of heat carrier leakage during overheating in summer;
- There are very few problems with corrosion;
- Air heating collectors are less demanding on materials, much cheaper;
- When using directly heated air in the collectors, there is no heat loss during heat exchange;
- Fire safety.

Due to these advantages, they are always suitable for individual constructions, are easy to assemble during installation, and require little effort. Air collectors are also effective for heating industrial buildings, garages, and small outbuildings in front of houses. At the same time, air collectors have the following disadvantages:

- Air vents can reduce the usable area of a building;
- They cannot achieve effective heat accumulation;
- Due to the low density of the heat-carrying air, the heat generation efficiency of the system is low compared to liquid-based collectors;
- Typically, large electrical power is consumed to pump air during the operation of air-heating collectors;
- The efficiency of an air-heating collector is lower than that of a liquid-based collector, which is determined by the physical properties of the air.

Two different designs of air heater collectors

Air heating collectors are mainly made in a flat form. They consist of a heat-insulating material located in the lower and side wall housing, an absorber, a highly transparent coating and a housing. When choosing various components, housing and other materials, taking into account resistance to atmospheric and other influences, the same basic rules are followed as for liquid collectors.

Depending on the design of the absorber, air heating collectors are divided into 2 classes:

- in the form of an air-permeable matrix absorber (Fig. 3.8.1);
- in the form of an air-impermeable flat absorber (from the back side) (Fig. 3.8.2).

An air collector with an air-permeable matrix absorber (hereinafter referred to as a matrix absorber) consists of a porous material with open pores (the term “matrix” is used here). For example, this can be foam plastic plates placed in the air duct as an air filter, or black porous material used in weaving. Solar radiation is absorbed not only on the surface of the absorber layer, but also inside it. A slit of variable width is placed

in the middle of the absorber with a transparent coating, which serves to pass air to the absorber. The air also absorbs heat during filtration through the absorber.

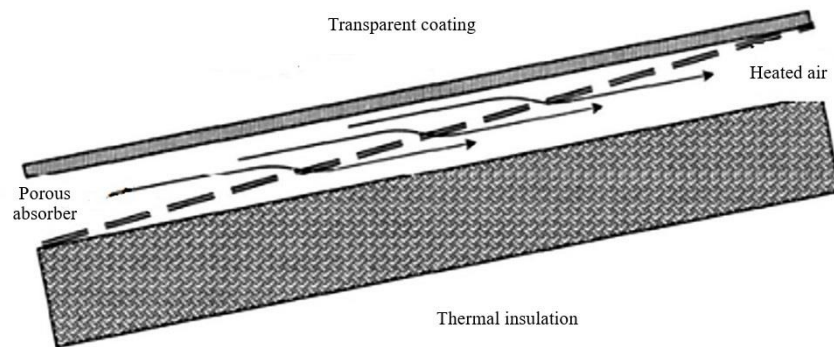


Figure-3.8.1. Solar air heating collector with matrix absorber.

As shown in Figure 3.8.2, the air moves parallel to the absorber, either along the two surfaces of the absorber or along its back. Heat is transferred to the air by convection or conduction along the absorber wall. When a collector is used to heat the outside air, a collector with an air-permeable matrix absorber is usually more efficient. The fact is that when the outside air temperature is low, the highly transparent coating is cold, while when using an airtight absorber structure with air moving along the two absorber surfaces, heat losses through the transparent coating are high. In this case, if it is required to heat the air to a high temperature under normal conditions, a structure with air movement primarily from the back of the airtight absorber structure is preferable.

The heated air does not come into contact with the transparent upper coating at a cold temperature, the heat transfer coefficient is small. It is necessary to consider good heat transfer from the absorber to the heat carrier, and when designing air collectors compared to liquid collectors, it is necessary to consider a sufficiently high efficiency coefficient of the absorber. The thermal conductivity of air is 24 times lower than that of water. Therefore, it is necessary to create a large surface for heat exchange and create an air flow of narrow cross-section. At the same time, it is necessary to consider the electrical energy consumption and hydraulic resistance of the fans that provide the movement of the heat carrier. With an increase in the average air velocity in the channel, the hydraulic resistance also increases rapidly. In general, the heat transfer coefficient from the absorber to the air increases slowly. In the collector, it is necessary to optimize the air channels in specific designs. The goal of optimization is to increase the possibility of creating a large surface area of contact with the absorber to avoid excessive air pressure drop for efficient heat transfer.

Also, a combination of photovoltaic cells and air heating collectors has been developed, called an air photothermal device (PV-T collector). Typically, photovoltaic modules installed on building facades and roofs are cooled by creating natural

convection or forced air circulation on the back side to cool the solar elements (Fig. 3.8.3).

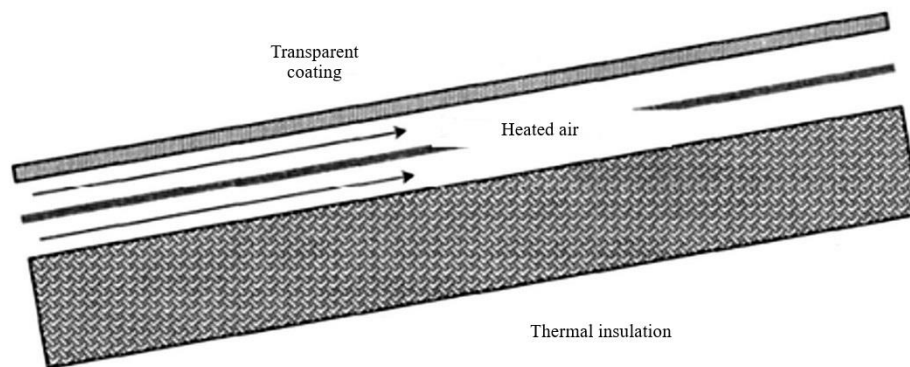


Figure -3.8.2. Solar air collector with airtight absorber.

SolarVenti introduced a similar design (Figure 3.8.3) to the world market in 2001. The design was developed by the company to automatically ventilate country houses by cooling photovoltaic modules. Several developments by SolarWall have also found their way into the market. The heat released from the back of the photovoltaic modules is directed to the building's heating system using ventilation or air conditioning.

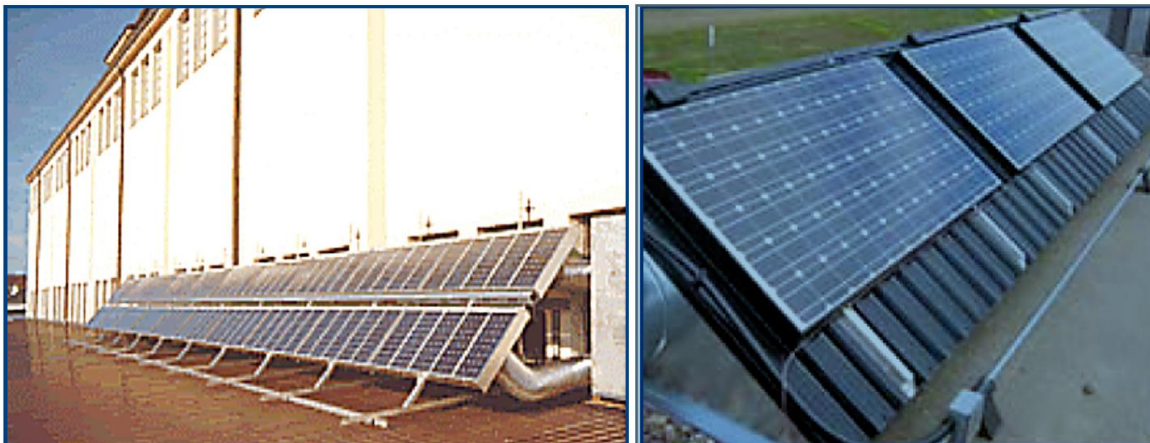


Figure-3.8.3. Air photothermal devices created by the company SolarWall.

3.9. The working process of passive and active circulation solar water heaters

Let's consider a system of collectors, a tank-accumulator and connecting pipelines filled with cold water. Solar radiation passes through a transparent coating (glass) and heats the light-absorbing panel of the collector and the water in its channels. During heating, the density of the water decreases, the heated liquid begins to move to the upper point of the collector, and then flows along the pipe to the tank-accumulator. The heated water in the tank moves to the upper point, and the colder water settles in the lower part of the tank, that is, the separation of water into layers occurs depending on the temperature.

Cold water moves along the pipe from the lower part of the tank to the lower part of the collector. Thus, with sufficient solar radiation, a constant circulation is

established in the collector circuit, the speed and intensity of which depend on the solar radiation flux density.

As a result, a pressure difference (Δp , Pa) is created in the system, which causes natural circulation in the solar collector, i.e.

$$\Delta p = g \times H \times (\rho_1 - \rho_2)$$

in this $g = 9,81 \text{ m/s}^2$; $N = H_1 - H_2 \text{ (m)}$ – the difference between the upper limit of the hot water inlet in the tank accumulator and the lower cold water inlet of the solar collector; ρ_1 and ρ_2 – are the density of the cold water at the bottom of the tank accumulator (T_1) and the water heated in the tank accumulator (T_2), respectively.

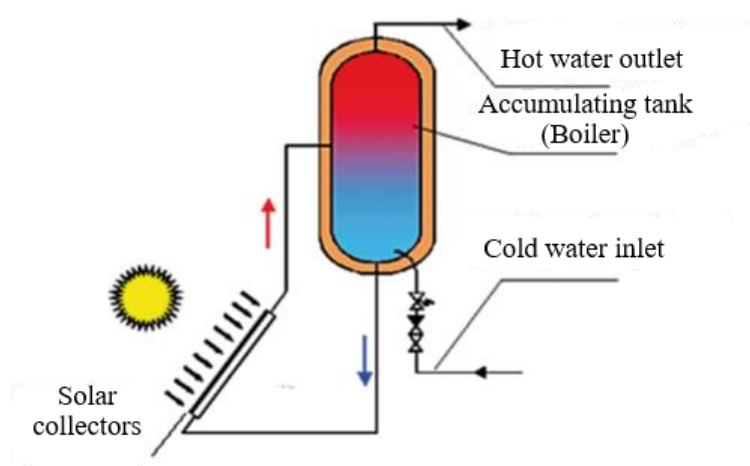


Figure-3.9.1. Single-circuit thermosiphon system for direct water heating.

The higher the difference between (T_1) and (T_2) and the larger the value of N , the higher the intensity of natural water circulation in the device. The peculiarity of such a system is that in the case of a thermosiphon system, the lower point of the tank accumulator should not be higher than 3–4 m from the upper point of the collector, and in the pump circulation, the tank-accumulator can be placed arbitrarily. This condition is important not only to ensure normal water circulation in the presence of solar radiation during the day, but also to prevent reverse water circulation in the device. These devices are widely used all over the world, especially in countries with a hot climate, and are considered very simple in terms of operation and manufacture.

In cold climates, it is advisable to use a dual-circuit scheme of solar water heating collectors, rather than a single-circuit one. In this case, any non-freezing, chemically inactive liquid (for example, a mixture of water with ethylene or propylene, antifreeze, glisantine (a mixture of water with glycerin), etc.) serves as the main heat carrier heated in the collector.

Two-circuit system with passive circulation of heat carrier: the operation of such a system is similar to the operation of a single-circuit system, but the system has a separate closed collector circuit consisting of a heat exchanger, pipelines, collectors in the tank-accumulator. This circuit is supplied with a non-freezing heat carrier according to a special rule.

After heating in the collector, the heat carrier falls to the upper part of the heat exchanger and transfers its heat to the water in the tank, then cools down and moves down, that is, to the inlet of the collector. In this case, it performs constant circulation under the influence of solar radiation.

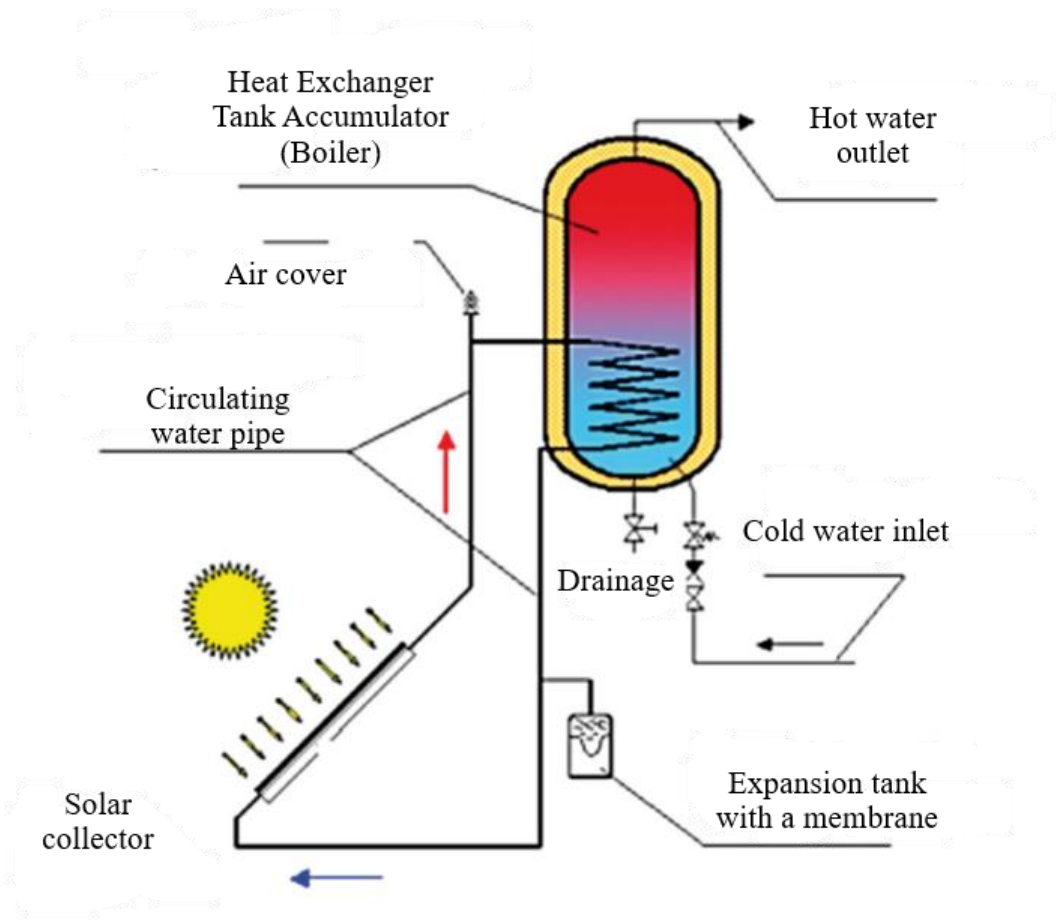


Figure-3.9.2. Schematic diagram of a dual-circuit system with passive circulation of the heat carrier.

In this case, in the thermosiphon case, the difference between the lower limit of the tank-accumulator and the upper point of the collector should not exceed 3-4 meters, in the pump mode, the tank-accumulator can be installed in any place.

In systems with forced circulation, a circulation pump is added to the collector circuit, which allows the tank-accumulator to be placed in any part of the building. The direction of movement of the heat carrier in the collector must coincide with the direction of forced circulation.

The pump mode is switched on and off using an electronic control unit consisting of a differential control relay, that is, a device that compares the readings of temperature sensors installed in the tank and at the outlet of the collectors. If the temperature in the collectors is higher than the water temperature in the tank, the pump is turned on. There are also units that allow you to change the pump operation and the speed of rotation of the heat carrier, taking into account the constant temperature difference between the tank and the collector. Systems with active circulation are usually two-circuit.

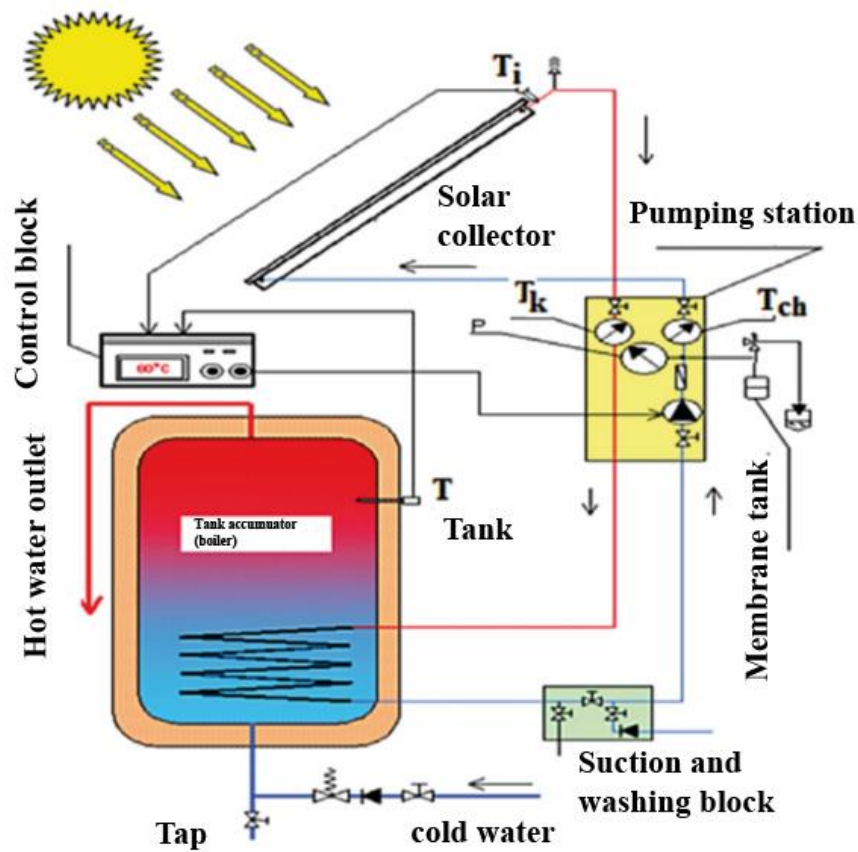


Figure-3.9.3. Principal diagram of a dual-circuit solar hot water supply system with forced circulation.

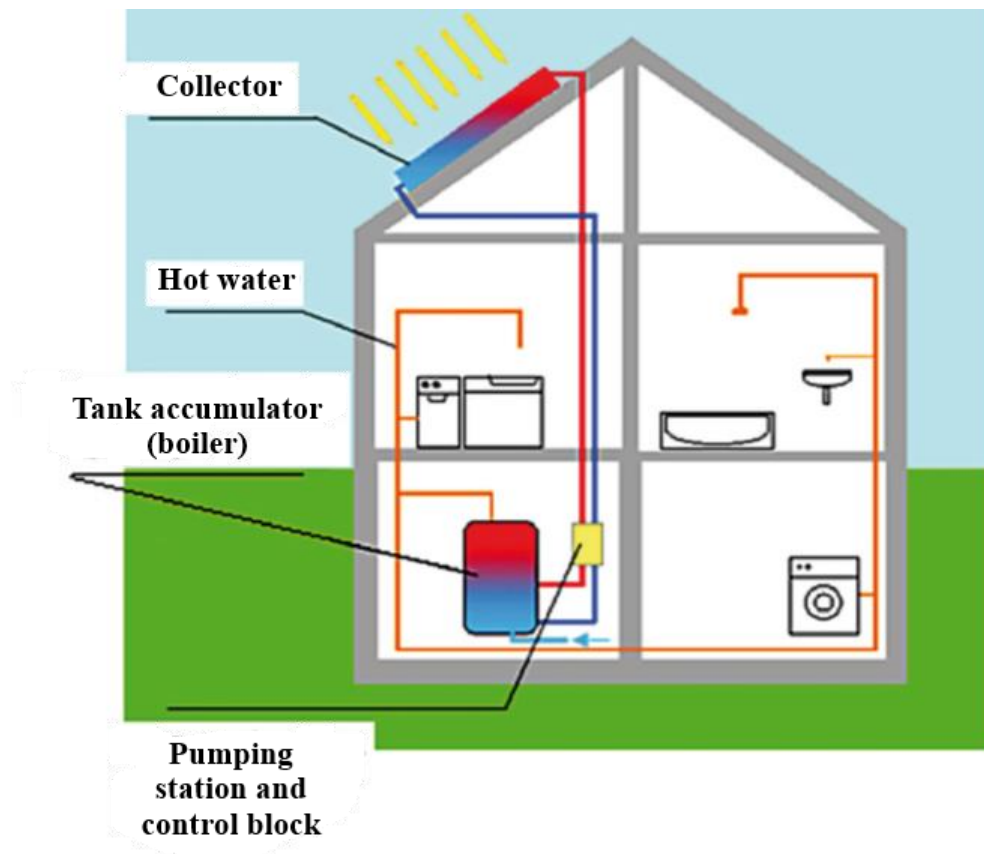


Figure-3.9.4. Location of solar system elements in a building for hot water supply.

3.10. Preparation, installation and operation of solar energy devices

Selection of materials for solar energy installations

Heat supply of solar energy devices requires more careful design, construction, installation and operation of elements compared to simple systems. To ensure the effective operation of solar devices throughout the entire accounting period, it is necessary to make the right choice of solar devices and materials for their manufacture.

Materials for the manufacture of solar collector housings. The main element of active solar systems is a solar energy collector and a heat accumulator. Various materials are used to manufacture these elements of the system - materials, plastics, glass, concrete, wood, polymer films, heat-insulating materials, rubber. The main requirement for the selection of materials is the compatibility of structural materials with the working fluid under operating conditions. The peculiarity of the operation of solar collectors is the influence of the external environment on it. Therefore, the collector body, which incorporates such structural materials, i.e. a light-absorbing surface with tubes or channels for the heat carrier, a glass layer, thermal insulation, must reliably protect it from the effects of the external environment, in general, from the effects of harmful substances, dust, moisture.

The collector body can be made of galvanized steel, aluminum, glass fiber, wood, rubber, composite materials and other materials.

The selection of materials is carried out taking into account the availability of materials and in accordance with the design. Thus, for a solar collector with a vacuum tube, tubes made of high-quality borosilicate glass are required. All materials used to make collector elements must withstand the highest and lowest operating temperatures. In the inner part of the collector body, a free space for the expansion of the absorber must be provided, and the temperature in it can be 200 °C and higher (in the presence of a heat carrier).

Materials for the light-absorbing surface of the collector. When choosing structural materials for the manufacture of elements of solar systems, it is necessary to take into account their compatibility with working fluids. In this case, the following recommendations should be taken into account to prevent corrosion.

Aluminum - this material cannot be used in direct contact with water coming from the water mains ($pH = 5 \div 9$) without chemical treatment and without the addition of corrosion inhibitors; it can be used in direct contact with distilled water or water with a corrosion inhibitor, only it is necessary to avoid contact with copper or iron with a small positive electrode potential, which forms galvanic pairs with aluminum. In addition, aluminum can work with organic liquids that do not contain water. The speed of water solutions and water in the pipes should not exceed 1.25 m/s.

Copper and its alloys can be in direct contact with tap water with a low content of chlorides, sulfates and sulfides or with ionized, distilled water. Copper cannot be used in the following cases:

- 1) In contact with aqueous solutions containing high concentrations of hydrogen sulfide, chlorides and sulfates;
- 2) In direct contact with water or solutions with a velocity of more than 1.25 m/s and a pH of up to 5.

It is recommended that steel be in direct contact with distilled, settled water or water containing a small salt content with a corrosion inhibitor ($pH = 8 \div 12$). It cannot be used in direct contact with untreated tap water, distilled, deionized water with a pH higher than 12 or a pH of up to 8.

Galvanized steel (iron) should be used for internal lining of non-aqueous organic liquids and cathodic protection heat accumulators. It cannot be used in direct contact with aqueous solutions containing copper ions with a pH of up to 8 or a pH of more than 12, as well as with water temperatures above 55 °C. Stainless steel should have high resistance to corrosion pitting and intergranular corrosion, pitting corrosion in working environments. It can be used in contact with non-aqueous organic liquids. In all other cases, the choice of stainless steel grade should be based on its compatibility with a specific liquid medium. Plastic, rubber, caoutchouc, composite materials are well compatible with water and other liquids as liquid heat carriers. However, their scope of application in the field of solar technology is not yet wide.

The materials must have the following characteristics:

- Good resistance to atmospheric factors - precipitation, pollution, etc., and ultraviolet radiation;
- Ability to withstand temperature fluctuations from - 25 °C to 150 °C;
- Sufficient mechanical strength and fire safety;

Disadvantages of plastics: they degrade under the influence of ultraviolet radiation and cannot withstand high temperatures that can develop in the absence of a heat carrier in the collector. If plastics, rubber, rubber and composite materials are used, the mass of the collector is reduced to 5–10 kg/m². In Germany and France, solar collectors are made and developed from synthetic rubber and resistant plastic pipes that are not affected by ultraviolet radiation, have low cost, high efficiency, withstand low (up to –30°C) and high (up to 110–140°C) temperatures. The cost of a plastic solar collector producing 1 kW·h of thermal energy is 10–12 times cheaper than a metal structure, and 4–5 times less than the energy from burning liquid fuel.

It is clear that it is necessary to focus on the creation and production of new types of solar collectors using modern polymer and composite materials. It is advisable to use plastic collectors for low-temperature solar installations for hot water supply in

individual residential areas, settlements, rural agricultural facilities, as well as for heating swimming pools.

Materials for transparent insulation of solar collectors. Transparent insulation is used in solar collectors to reduce heat loss and prevent fires from entering them. Usually, one or two layers of transparent insulation are used in solar collectors, but a solar collector design without transparent insulation, sometimes a three-layer insulation design, is also used.

A transparent insulation material must have a high transmittance (coefficient) for the solar radiation flux density (wavelength from 0.3 to 2.5 μm) and at the same time be practically opaque for long-wave radiation (more than 3 μm) - thermal radiation, i.e. radiation emanating from the absorber surface.

Usually, frame glass is also used, preferably glass with a low iron oxide content. The glass should be insulated from the metal surface with a Π -shaped gasket, which prevents the formation of cracks and damage to the glass due to thermal stresses. An alternative material for transparent insulation is a polymer film, unfortunately, its significant disadvantage is its degradation under the influence of ultraviolet radiation. However, given its low cost, it is advisable to use it. The polymer film (compared to glass) transmits solar radiation well, so a polymer film can be used instead of a glass layer. To increase the shelf life of the film, it is necessary to specially process it to increase its stability to ultraviolet radiation.

Thermal insulation materials for collectors. Thermal insulation material must meet the following requirements. It must have a low thermal conductivity coefficient λ , low density ρ , high melting point, high resistance to weather conditions and various external influences. The best thermal insulation materials are polyurethane foam and polystyrene, but mineral wool and glass wool can also be used.

Table-3.10.1

Thermal insulation materials

Material	$\lambda, W/(m \cdot K)$	$T, ^\circ C$	$\rho, \text{kg}/\text{m}^3$
Polystyrene	0,035	80	0,02
Polyurethane foam	0,028	100	0,035
Penoplast	0,03-0,08	150	0,0-0,7
Polyvinyl chloride	0,035	130	0,04-0,08
Polymethacrylamide	0,029-0,035	160	0,03-0,2
Mineral wool	0,038	200	0,145
Fiberglass	0,036	300	0,12

Some of the most commonly used characteristics in solar collector designs are listed in Table 3.10.1.

Sealing materials and gaskets. In most cases, silicone rubber is suitable for sealing the glass in the solar collector housing. Sealing gaskets should be placed on

both sides of the glass. In this case, the gasket should have a Π -shape with a gap for the glass.

Materials for selective coatings

The absorber of the solar collector must have a high absorption capacity in the solar radiation spectrum. At the same time, energy is lost due to thermal radiation of the absorbing surface, therefore, in order to reduce losses, the emissivity of this surface should be as small as possible in the long-wave region. In most flat-plate collectors, the temperature of the absorbing surface does not exceed 200°C (473 K), while the effective temperature of the Sun's surface is about 6000 K. This means that the spectral range of the incident solar radiation and the long-wave radiation emitted by the surface (98% of the solar radiation spectrum outside the atmosphere in the wavelength range of not less than 3 μm is not less than 1% of the radiation of a black body at a temperature of 200°C). This situation requires the creation of selective surfaces with high absorption capacity in the long-wave region and with respect to solar radiation.

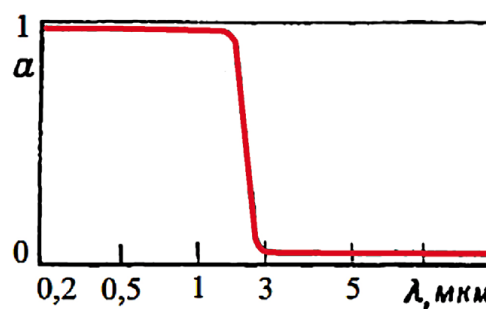


Figure-3.10.1. An ideal selective surface with a threshold wavelength of 3 μm .

The essence of the concept of a selective surface can be understood from the graph presented in Figure 3.10.1. The ideal surface represented there is called half-gray, since it can be considered gray in the solar spectrum (i.e., in the wavelength range less than 3.0 μm). Such an ideal surface has a very small spectral reflectance in the long-wavelength range.

The absorption in the solar radiation spectrum and the emissivity in the long-wavelength radiation range are determined from the spectral reflectance data obtained by integrating over the corresponding wavelength range.

The absorption with respect to solar radiation, as defined in the literature, is expressed in the form α , and the emissivity is expressed in the form ε .

It is known that solar collectors usually operate at sufficiently low temperatures, practically all the emitted radiation falls on the wavelength range greater than 3 μm .

α – the absorptivity is defined as the fraction of solar radiation energy absorbed by the surface. This quantity can be increased to a maximum value of one, independent of the operating mode.

ε – emissivity indicates how much of the radiant energy emitted from a surface is relative to the energy radiated by a black body at the same temperature.

The α/ε ratio is often used as a criterion for the efficiency of a selective surface. The radiation concentration in this ratio determines the maximum achievable temperature and, therefore, the maximum efficiency. However, to determine the overall efficiency of the system, the values of α and ε must be given separately.

A coating with a high transmittance coefficient in the long-wave region and a high absorption coefficient relative to solar radiation can be applied to a surface with a low emissivity. As a result, the coating absorbs solar radiation, while the substrate is an emitter in the long-wave region of the spectrum (with poor properties).

The coating can have a homogeneous or fine-grained structure, as a result of which its properties are determined by the optical properties of the coating material. Metal substrates are coated with a coating consisting of metal oxides, for example, copper oxide on aluminum, copper oxide on copper, nickel zinc sulfide on galvanized iron, etc. A coating consisting of black chromium is widely used in solar collectors. The base is nickel deposited on copper or steel. The coating is formed by electroplating in a galvanic bath consisting of chromic acid and other reagents. In laboratory conditions, values of the absorption coefficient up to 0.95 and the reflection coefficient up to 0.14 have been obtained. It was found that the wavelength interval at which the transition from low reflectivity to high occurs is from 1.5 to 5 μm . In this case, as the coating thickness increases, the transition shifts to longer wavelengths. The coating consists of a set of particles and voids.

To obtain a selective coating of black copper on the surface of the copper plate, it is carefully processed for various periods of time in solutions of sodium hydroxide or sodium chloride oxide heated to 140°C, and a layer of copper oxide is created. A similar patented darkening process is also used in the USA under the name "Ebanol". The absorption capacity of the coating can be increased by using interference. The creation of selective surfaces in vacuum tube collectors has been studied by foreign researchers using vacuum sputtering methods. The sputtering method for obtaining a metal coating can be carried out in an inert atmosphere (Argon) or in an atmosphere (Argon, with a 1–2% mixture of methane) to create a metal-carbide metal coating. It is known that these coatings have a moderate absorption capacity ($\alpha = 0.8$) and a very small emissivity ($\varepsilon = 0.03$). For most applications, they must have very high absorption capacity.

In order to obtain an operating temperature of 300 to 400 °C, the deposition method is used to create selective metal-ceramic coatings on the receiver surfaces of collectors equipped with parabolic trough concentrators. Four layers are deposited on the receiver surface of the steel pipe: an anti-diffusion oxide layer that prevents molecular diffusion of the steel substrate in the coating; a metal-ceramic layer that

reflects infrared radiation and provides a low emissivity; an absorbing metal-ceramic layer that makes the oxide layer transparent: Such a coating also works in vacuum insulation, has an absorption capacity of 0.96 for solar radiation, a calculated emissivity of 0.16 in the long-wave radiation range at 350 °C and is extremely stable at receiver temperatures of around 400 °C.

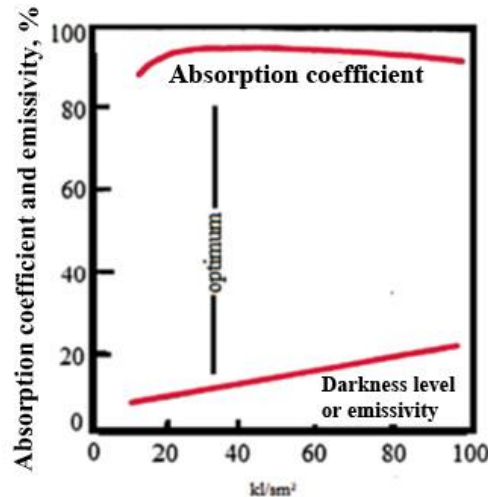


Figure-3.10.2. Variation of α and ε with annealing time and current density of black chrome.

In flat-plate solar collectors, the high absorptivity of the receiving surface is more important than the emissivity. For most surfaces, the relationship between α and ε shown in Figure 3.10.2 is characteristic. For the black chrome considered above, the optimum deposition time (coating thickness) is clearly visible. For other selective coatings, the optimal value of the mass per unit area or other quantitative indicators characterizing their physical properties is not known.

In general, conclusions can be drawn based on the assessment of the influence of α and ε on the annual operation of the solar device. However, the general rule is that the greatest efficiency is achieved in the range of α values close to the maximum. Table 3.10.2 shows the values of the emissivity in the infrared radiation range and the absorptivity of surfaces with selective coatings in relation to solar radiation.

Table-3.10.2

Properties of selective surfaces

Coating	Absorption capacity in the solar radiation range, α_s	Emissivity in the infrared radiation range, ε_T	Selectivity of the coating, α_s/ε_T
Black paint:			
Matte	0,95-0,98	0,9-0,97	1
Silicone	0,94	0,4	2,35
Acrylic	0,92-0,97	0,84-0,9	1,1
Black chrome	0,87-0,93	0,1	9
Black chrome on nickel	0,92-0,94	0,07-0,12	8213

Black nickel on nickel	0,93	0,06	15
Black zinc	0,9	0,1	9
Copper oxide on aluminum	0,93	0,11	8,5

Table-3.10.3

Properties of materials for solar systems

Material	S, kJ/kg · K)	ρ , kg / m ³	λ , W/(m · K)	α	ε
Building materials					
Concrete	0,834	1920-2240	0,8-1,73	0,6-0,98	0,88-0,97
Brick	0,921	1920-2080	0,6-1,3	0,26-0,89	0,93
Wood	2,51-2,93	350-740	0,1-0,16	0,6	0,9
Ceramic tile	0,8		1,7-2,9	0,02-0,08	0,6-0,9
Steel	0,5	7830	45	0,8-0,9	0,85
Oxidized iron	0,5	7-----10	55	0,8-0,94	0,94
Aluminum:					
polished	0,88	2740	202	0,1-0,4	0,03
oxidized	0,88	2740	202	0,4-0,65	0,09

Table 3.10.3 above lists the properties of materials used for solar systems.

Heat carriers for solar collectors. In solar collectors, water or a non-freezing liquid antifreeze is mainly used as a heat carrier in liquid heating and hot water supply systems. In air collectors, the heat carrier is air itself. Water has clear advantages and disadvantages compared to air as a heat carrier. Water has good thermal physical properties (heat capacity, heat transfer coefficient, viscosity, density, etc.), but at low external temperatures, water freezes in other elements of the solar system and in water pipes. Necessary measures must be taken to prevent it from freezing. For example, to prevent the cold water pipe coming to the solar collector from freezing, it must be wrapped with heating cables supplied from an electric source and covered with thick thermal insulation materials. A serious problem is the corrosion activity of this water with most construction materials. Air does not freeze and does not cause corrosion, but heat, due to its physical properties, gives way to water. The dimensions of the air passages and channels for air flow are much larger than the dimensions of the structural elements of liquid systems. In addition, fans in air systems consume a large amount of energy and create noise. To ensure effective heat exchange with the absorbing surface, the air flow must cross the entire surface of the absorber. Along with water, non-freezing heat carriers are used in liquid systems - ethylene and propylene glycol, etc. Their disadvantage is the short shelf life (up to 3-5 years). Propylene glycol, unlike ethylene glycol, is not considered toxic. The toxicity of water depends on the type of corrosion inhibitor used. The use of freons is not recommended, as they damage the ozone layer of the Earth's atmosphere.

Manufacturing and installation of solar collectors

Solar collectors are usually manufactured in factory conditions, and at the same time the supporting structures are also manufactured. However, simple collectors can be manufactured locally, but it should be borne in mind that in this case their thermal efficiency will not be high.

The main structural element in solar collectors is the absorber, that is, the light-absorbing surface. The structural forms of various types of absorbers for the most commonly used flat solar collectors are presented in Figure 3.10.3. The most commonly used absorber in liquid collectors consists of structural elements consisting of a row of small-diameter (10–15 mm) tubes connected by a flat sheet.

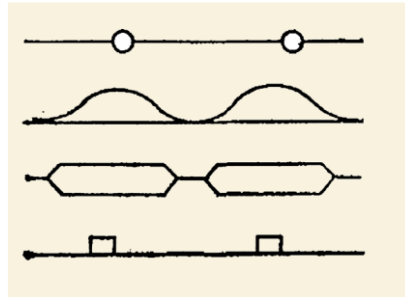


Figure-3.10.3. Schemes of absorbers in flat liquid collectors: a – pipe on a flat sheet; b – combination of corrugated and flat sheets; c – stamped absorber; d – sheet welded in the shape of a rectangle.



Figure-3.10.4. Assembly instructions.

Assembling and installing a vacuum tube solar collector: At least 2 people are required to assemble a solar collector, one of whom must be a specialist. The specialist must have all the tools. In addition, liquid soap and a sponge are also used. The assembly and installation of one device takes 2-3 hours. The assembly and installation process is carried out in accordance with the instructions. The base frame is made using the photos in accordance with the instructions. During assembly, the base frame is fastened mainly with bolts and nuts attached to the assembly.

3.11. Tower-type solar power plants and their energy characteristics

The idea of the technological cycle of tower-type solar power plants (TSP) was proposed more than 370 years ago. The practical development of TSP began in the 1960s of the 20th century, and in the 1980s this type began to develop much more than other types of TTSP (Table-3.11.1).

TSP is based on the well-known thermodynamic cycle, in which, instead of a steam boiler due to the combustion of organic fuels (gas, oil, coal, peat, etc.) in a TPP, a similar boiler is used, using various vapor and liquid heat carriers due to the heat of solar energy (Figure-3.11.1).

Table-3.11.1

The world's first tower-type solar power plant built at the end of the 20th century

TTSP	Location area	Country	Launched year	Electricity N (MW)	Heat carrier
SSPS	Algeria	Spain	1981-year	0,5	Liquid sodium
EURELIOS	Sicily	Italy	1981-year	1,0	Water vapor
SUNSHINE	Nio Town	Japan	1981-year	1,0	-
CESA-1	Algeria	Spain	1983-year	1,0	-
THEMIS	Targassonne	France	1982-year	2-2,5	Salt solution
Solar One	Barstow	USA	1982-year	10	Water vapor
Solar Two	Barstow	USA	1999-year	10	Liquid sodium
SES-5	Crimea	Russia	1986-year	5,0	Water vapor

The solar energy receiver (boiler) is located on a tower high above the ground, onto which solar radiation is reflected using many automatically controlled mirror reflectors (heliostats). In other words, the inhabitants of the city of Syracuse are reminded of the ancient idea of Archimedes. According to him, enemy ships in the port of Syracuse were burned by reflecting mirrors.

Solar radiation is concentrated and concentrated at one point and fed to the heat-absorbing surface of the boiler, where steam is formed, and then the steam directly enters the heat exchanger or steam turbine. A generator rotor is mounted on the shaft of the steam turbine, which produces electrical energy of a certain frequency and

voltage. Often, the capacity of a tower-type solar power station is determined by the height of the tower on which the boiler-receiver is located. In this case, the high height of the tower prevents the risk of heliostats shading each other. For example, in a tower-type solar power station with a capacity of $50 \div 100$ MW, the height of the tower should be $200 \div 300$ m, and the area of the used heliostats should be $2 \div 3$ km² (15 $\div$ 25 thousand). For a tower-type solar power station with a capacity of $150 \div 200$ MW, a tower with a height of $350 \div 400$ m is required, but in practice it is very difficult to build such stations. Based on this, it can be said that the capacity of the built tower-type solar power station is limited to $5.0 \div 10.0$ MW and the height of the tower is $70 \div 100$ m. For a tower-type solar power station, the main disadvantage is the cost of expensive land for installing multiple heliostats.

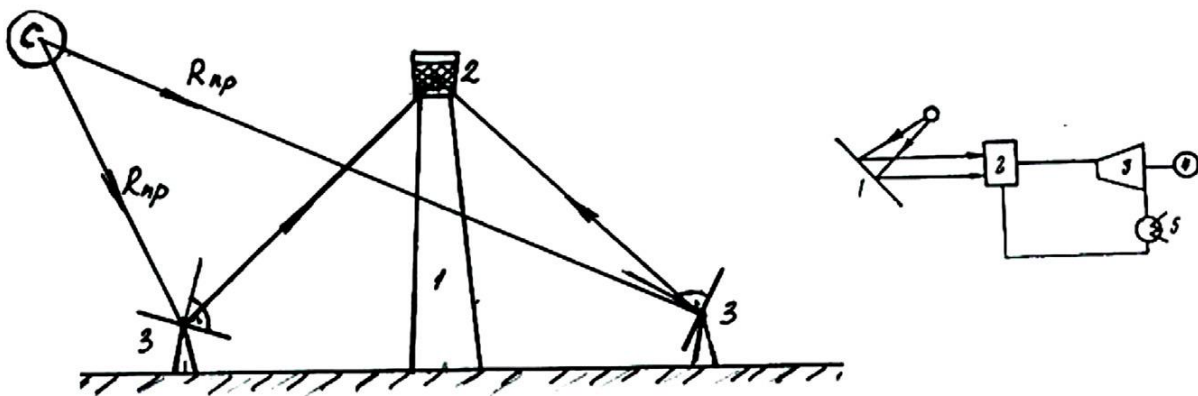


Figure-3.11.1. Main structures of a tower-type solar power plant: 1 – tower, 2 – solar radiation heat receiver-boiler; 3 – heliostats (left); Principle diagram of converting solar energy into electricity: 1 – heliostats, 2 – boiler, 3 – turbine, 4 – generator, 5 – condenser (right).

Since 1981, the BSES EURELIOS tower solar power plant with a capacity of 1 MW and a tower height of 50 m has been operating on the island of Sicily (Italy). In its heat receiver, water vapor with a temperature of 600°C is generated, which is used in a conventional steam turbine. Also, in Alceria, southern Spain, a 0.5 MW solar power plant has been operating since 1981, where liquid sodium is used as the primary heat carrier, which transfers its heat to water vapor through a heat exchanger. The sodium heat carrier is also a time-based heat accumulator. Such a solar power plant, taking into account the random and cyclical nature of solar radiation, provides an additional source of power in the energy system, which allows saving on scarce organic fuels. To increase the efficiency of such solar power plants, energy collectors can be added to their technological schemes, which will help redistribute the solar energy falling during the daytime during the brightest sunlight.

The above is illustrated in Figure 3.11.2, where in cases a and b the two most developed tower-type solar power plants with heat accumulators are depicted. In Figure

3.11.2, a) A heat accumulator is connected in series to the general technological circuit of converting solar radiation in a tower-type solar power plant. b). Only a part of the working body heated in the tower of the solar power plant is allocated to the heat accumulator. The useful thermal power of the tower-type solar power plant N_{QES} is determined by the following formula:

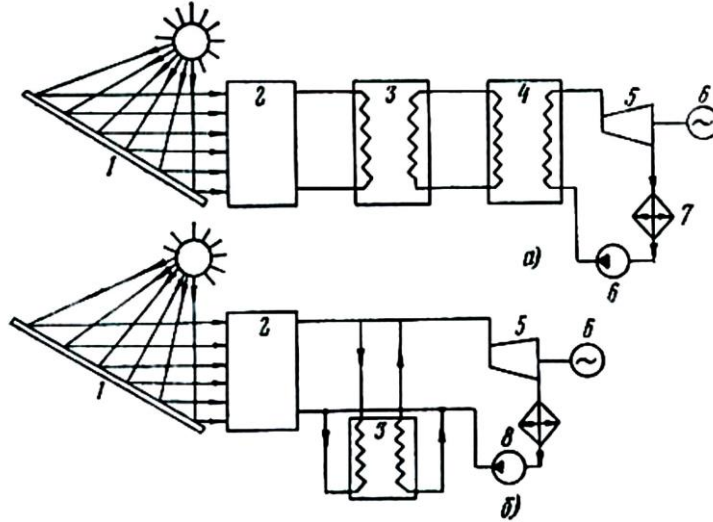


Figure-3.11.2. Technological scheme of a tower-type solar power plant with an accumulator: 1 – heliostats, 2 – receiver (boiler), 3 – heat accumulator, 4 – heat exchanger, 5 – steam turbine, 6 – generator, 7 – condenser, 8 – pump.

$$N_{QES}(t) = R_{\Sigma}(t)F_g r_{aks.} \sin \theta K_{soya.} K_{bl} K_{i.yu.} K_{chang.} r_k \quad (3.11.1)$$

where, $N_{QES}(t)$, kW; $R_{\Sigma}(t)$ – Direct solar radiation incident on a surface of 1 m² (in kW/m²); F_g – area of heliostats (m²); $r_{aks.}$ – the reflecting ability of heliostats (0,75); $\sin \theta$ – 0,75–0,8 – The actual angle of incidence of solar radiation on heliostats; $K_{chiq.}$ – Shadow formation coefficient in heliostats; $K_{blok.}$ – heliostat blocking coefficient (usually $K_{chiq.} K_{blok.} = 1$); $K_{i.yu.}$ – heat loss coefficient 0,85; $K_{chang.}$ – pollution coefficient 0,95; r_k – solar radiation absorption coefficient by the boiler heat sink 0,93 ÷ 0,95.

Taking into account all types of energy loss in $\eta_{QES}^{term.}$, it can be written as follows:

$$N_{QES}(t) = R_{\Sigma}(t)F_g \eta_{QES}^{term.} \quad (3.11.2)$$

where, $\eta_{QES}^{term.}$ – Tower-type Solar Power Plant Overall Efficiency.

In 1985, the first experimental tower-type solar power plant “QES-5” with a capacity of 5 MW was launched in the Kerch Peninsula, Shelkino settlement, Crimea region of the former USSR. Solar energy is concentrated in an 89 m high open cylindrical tower serving as a steam generator. The heating surface of the boiler is 154 m², which produces 28 tons of saturated steam per hour at a pressure of 4 MPa and a temperature of 250 °C. The heat flux density of solar radiation is 130 kW/m², which is realized by 1600 flat glass squares with an area of 25.5 m² and a reflection coefficient

of 0.71 heliostats. The designed operating hours of this tower-type solar power plant are 1920 hours per year. The ratio of the total area of the heliostats to the boiler surface is 211. It is planned to install a water-steam heat accumulator with a capacity of 500 m³ at “QES-5”.

3.12. Solar pools and their energy properties

In ordinary reservoirs of drinking water, the absorbed solar energy mainly heats the upper layer, and this energy is quickly lost, especially at night, when the air is disturbed, due to water evaporation, and changes in the ambient temperature. In reservoirs containing saline solution NaCl table salt or magnesium chloride MgCl₂, the mechanism of solar energy absorption occurs completely differently. In this case, depending on the salinity of the water, it is divided into layers, and the temperature gradient of the salt content from top to bottom divides the entire liquid volume into three zones, the salt content in which increases from the surface to the bottom. The initial thin upper layer (10–20 mm) is almost drinking water, and is limited by a second non-convective layer of greater thickness, the salt content in it gradually increases with depth and reaches a maximum of 15–25% for NaCl and 30% for MgCl₂. The thickness of this layer is 2/3 of the total depth of the reservoir. In the third, lower convective layer, the salt content reaches its highest level and is evenly distributed throughout the liquid part. The gradient of the salt solution composition along the depth of the arm exerts pressure on the free - convective - passage of the liquid from the bottom of the suspended layer to the surface, thereby ensuring that heat accumulates near the bottom. The high density of the salty lower layers of water allows for better absorption of solar energy, as a result of which the lower layers heat up more than the surface. For this reason, in some natural salt lakes, the water temperature at the bottom rises to 70°C. This indicates a high salt content.

In cases where solar energy passes through the entire water and is absorbed by the black-painted bottom, the water near the bottom heats up to 90–100°C, while the surface temperature of the layer remains at 20°C. The energy utilization of a solar artificial lake (reservoir) section and the change in the temperature of the liquid with respect to the height of the lake are shown in Figure 3.12.1. The utilization of the thermal gradient energy is used to pump water with a temperature of 60–90°C in the lower layers of the lake into 4 heat exchangers and evaporate liquids such as freon and ammonia that boil at low temperatures. This liquid vapor drives a turbogenerator in a conventional steam turbine section. The liquid vapor is cooled by the much colder surface of the water, condensed, and reused. Solar artificial lakes with large amounts of water are good heat sources, which makes it relatively easy to solve the heat storage device. For example, a 2 m deep lake can provide continuous operation of an electric generator for up to a week if the insulation fails. When creating lakes of the appropriate

depth, it is even possible to provide for seasonal operation of heat pumps. Electricity generated on the basis of solar lakes is relatively cheap and costs \$ 0.1 per 1 kWh. The efficiency of renewable energy sources, which are solar lakes, is several percent. From one hectare of lake area, up to 200–300 kW of electricity can be obtained.

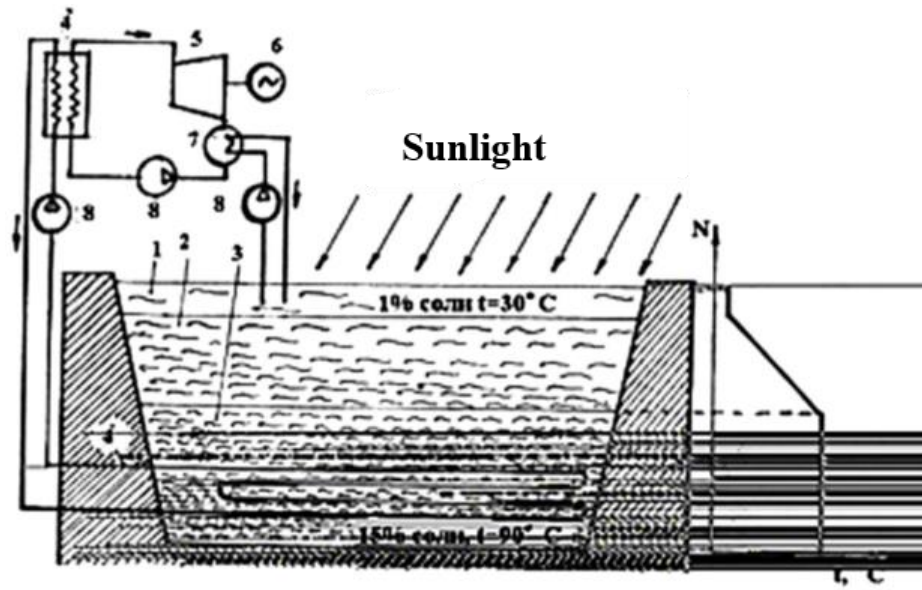


Figure-3.12.1. Solar artificial lake (reservoir) cross-section with energy utilization and temperature change of the liquid according to the height of the lake: 1 - fresh water; 2 - protective layer; 3 - hot mixture layer; 4 - heat exchanger; 5 - turbine; 6 - generator; 7 - condenser; 8 - pump.

Renewable energy sources in the form of solar lakes exist in a number of countries: Israel has a capacity of 300 kW and 5 MW, and the USA has a capacity of 5 MW. Effective research is being conducted in Australia, India, Italy, Japan, and Egypt on their construction and use. Good work is also being done in Uzbekistan, where natural salt lakes that have formed in large quantities in the Aral Sea zone of Karakalpakstan are being used.

3.13. Heat exchange during condensation

For the laminar flow regime ($Re = \frac{\omega \cdot \delta_x}{\gamma}$), the local heat transfer coefficient in film condensation can be approximately determined using the Nusselt formula.

$$\alpha_x = \sqrt[4]{\frac{r \rho^2 g \lambda^3}{4 \mu (t_n - t_c) x}}; \quad (3.13.1)$$

$$\alpha_x = \frac{\lambda}{\delta_x}. \quad (3.13.2)$$

The heat transfer coefficient for a vertical surface with laminar film motion ($Re_f=5...7$) is determined by the following formula:

$$\alpha_{sep} = 0,943 \sqrt[4]{\frac{r \rho^2 g \lambda^3}{4 \mu (t_h - t_c) H}}; \quad (3.13.3)$$

when, $7 < Re_j < 400$

Average value

$$\alpha_{ver} = 1,15 \sqrt[4]{\frac{r \rho^2 g \lambda^3}{\mu (t_h - t_c) H}}; \quad (3.13.4)$$

For horizontal gutter,

$$\alpha_{gor} = 0,728 \sqrt[4]{\frac{r \rho^2 g \lambda^3}{\mu (t_h - t_c) d}}; \quad (3.13.5)$$

where, r – heat of vaporization, J/kg (t_h accepted at condensation temperature);

ρ – condensate density, kg/m³;

g – acceleration of free fall, m²/s;

λ – thermal conductivity of condensate, W/(m·K);

μ – dynamic viscosity coefficient, Pa·s;

d – pipe diameter, m;

H – wall height, m;

w – local height of the film flow and average shear velocity, m/s.

δ_x – film thickness at a distance x from the film surface, m;

t_h and t_c – saturation and wall temperature, respectively, °C;

γ – kinematic viscosity coefficient, m²/s.

During the condensation of freons in copper pipes, α can be determined using the formulas: (3.13.3), (3.13.4), (3.13.5) where $H=d_{gh}$ and $d=d_{gh}$.

Heat transfer coefficient of freon for steam condensation conditions inside horizontal tubes of a flat coil

$$\alpha_{3M} = 0,195 \alpha \cdot q^{0,15}, \quad (3.13.6)$$

where, α (3.13.5) is determined by the formula.

Example-1. Dry saturated water vapor condenses on the outer surface of a vertical pipe with a diameter $d_h = 20$ mm and a height of 2 m under a pressure $p = 100$ kPa. The pipe surface temperature $t_c = 94.5$ °C.

Determine the average heat transfer coefficient from steam to the pipe and the mass of steam that condenses on the pipe surface.

Solution. Based on the application at a pressure of $P=100$ kPa, we find the saturation temperature $t_h=99,6$ °C and $Z=2258$ kJ/kg; $\lambda_{wc}=68,3$ W/(m·K); $\mu_{wc}=282,5 \cdot 10^{-6}$ Pa·s; $\nu=0,295 \cdot 10^{-6}$ m²/s; $\rho_{wc}=959$ kg/m³.

We determine the reduced height of the vertical surface

$$Z = \frac{H(t_h - t_c)}{z \rho \gamma} \left(\frac{q}{\gamma} \right)^{1/3} = \frac{2(99,64 - 94,5) \cdot 68,3 \cdot 10^{-2}}{2258 \cdot 10^3 \cdot 959 \cdot 0,295 \cdot 10^{-6}} = \left(\frac{9,81}{(0,295 \cdot 10^{-6})^2} \right)^{1/3} = 525$$

525 < 2300

Laminar flow mode.

The heat transfer coefficient averaged along the perimeter during film condensation of dry saturated steam in horizontal pipes can be determined by the formula:

$$Re = 0,81 \cdot z^{0,75}, \quad (3.13.7)$$

where, $Re = \frac{\alpha(t_h - t_c)\pi R}{z\rho v}$ - Reynolds number;

$$z = \frac{\lambda(t_H - t_c)\pi R}{z\rho v} \left(\frac{q}{v^2} \right)^{\frac{1}{3}};$$

R – pipe radius, m.

The thermal and physical parameters of the condensate in formulas (3.13.6) and (3.13.7) are selected depending on the saturation temperature.

The average heat transfer coefficient during condensation in a smooth pipe bundle is determined by the formula

$$\alpha_n = \alpha \cdot \varepsilon_n = \alpha \cdot n^{-0,167} \quad (3.13.8)$$

where, n - the average number of vertical pipes in the apparatus for a half row of a corridor or staggered set;

ε_n - coefficient taking into account the effect of the number of pipe rows.

n	1	3	5	10	15	20
ε_n	1	0,83	0,765	0.68	0,635	0,605

v - kinematic viscosity of condensate, m^2/s .

In equations (3.13.3), (3.13.4) and (3.13.5), the physical parameters of the condensate film ρ , λ , μ , v are determined at the average temperature $t_{cp} = 0,5(t_H + t_C)$, and r is determined at the saturation temperature t_H .

The average height heat transfer coefficient during film condensation of dry saturated steam for laminar flow of condensate film on a vertical surface and in vertical pipes can be determined by the formula

$$Re = 0,95 Z^{0,78} \cdot \varepsilon_t, \quad (3.13.9)$$

where, $Re = \frac{\alpha(t_H - t_C)H}{Z\rho v}$ - Reynolds number;

$$Z = \frac{H(t_H - t_C)\lambda}{z\rho v} \left(\frac{d}{\gamma^2} \right)^{\frac{1}{3}} - \text{reduced vertical surface height};$$

The formula is valid for $Re < Re_{kp} \leq 1600$

$Z_{kp} = 2300$. If $Re > Re_{kp}$ so, then the calculation equation is

$$Re = (253 + 0,069 \left(\frac{Pr_{\infty}}{Pr_c} \right)^{0,25} Pr^{0,5} (Z - 2300))^{\frac{4}{3}}, \quad (3.13.10)$$

where, $Pr_{\text{жс}}$ and Pr_c - are the values of the Prandtl number for the condensate at temperatures t_H and t_C , respectively,

$$\varepsilon_t = \left(\left(\frac{\lambda_c}{\lambda} \right)^3 \left(\frac{\mu}{\mu_c} \right) \right)^{\frac{1}{8}}$$

$$h_p = \frac{\pi}{4} \left(\frac{D_p^2 - d_H^2}{D_p} \right),$$

where, D_p – rib diameter, m;

d_H - pipe outer diameter, m;

A_e – the area of the horizontal surface between the ribs and the ends of the ribs.

$$A_e = 0,5\pi(D_p^2 - d_H^2)/S_p; \quad A_e = \pi d_H(1 - \delta_0/S_p) + \pi D_p \delta_m / S_p,$$

where, S_p – rib steps;

δ_0, δ_m – The thickness of the rib at the base and tip.

The value of E_p depends on both the geometry and material of the fin and the heat transfer conditions.

The E_p value for a flat rib is calculated using the formula:

$$E_p = (th(mh)/mh), \quad (3.13.11)$$

where, $th(mh) = (e^{mh} - e^{-mh}) / (e^{mh} + e^{-mh})$ – hyperbolic tangent in the argument mh ;

$m = \sqrt{\frac{2\alpha}{(\lambda_p \delta)}}$ - auxiliary quantity;

h and δ - rib height and thickness, m;

λ_p - thermal conductivity of the rib material, W/(m·K).

Depending on mh , the magnitude E takes on the following values:

mh	0	0,1	0,2	0,3	0,4	0,5	0,6	0,8	1,0
E_p	1	0,097	0,987	0,971	0,950	0,924	0,895	0,830	0,762
mh	1,2	1,4	4,6	1,8	2,0	2,5	3	3,5	4,0
E_p	0,695	0,32	0,576	0,525	0,482	0,394	0,332	0,285	0,250

The E_p coefficient for round, square and plate pipes of constant thickness is found using the following formula:

$$E_p = (th(mh')) / mh' ; \quad mh' = h' / (\delta 2Bi) \quad (3.13.12)$$

where, h' - height of a given rib, m;

$Bi = \alpha \delta / \lambda_p$ - Bio criterion.

The given height of the ribs is calculated:

$$\text{for round ribs:} \quad h' = h(1 + 0,805 \lg \frac{D_p}{d_H}), \quad (3.13.13)$$

where, $h = 0,5(D_p - d_H)$ – rib height.

The heat transfer coefficient during ammonia condensation in horizontal steel pipes can be determined by the formula

$$\alpha_m = 8980 q^{-0,2} d_{\text{вн}}^{-0,33} = 194007^{-0,167} d_{\text{вн}}^{-0,275} \quad (3.13.14)$$

Average heat transfer coefficient for condensation in a stacked tube bundle

$$\alpha_{np} = \alpha \cdot \varepsilon_p \cdot n^{-0,167} \quad (3.13.15)$$

where, ε_p the rib coefficient is determined by the following formula:

$$\varepsilon_h = 1 / (1 + (F_D' / F_{jh}) T_h^{0,75} (d_y' h_h)^{0,25} + (F_u' / F_{jh})) \quad (3.13.16)$$

where, ε_p - rib efficiency coefficient,

A_B - vertical surface area of the rib, m²,

A_{op} - the total area of the outer surface of the pipe, taking into account the ribs, m²

A rectangle with sides a and b , $a > b$.

$$h' = 0,5 d_H \left(1,28 \frac{b}{d_H} \sqrt{\frac{a}{b} - 0,2} - 1 \right) \left(1 + 0,805 \lg \left(1,28 \frac{b}{d_H} \sqrt{\frac{a}{b} - 0,2} \right) \right) \quad (3.13.17)$$

When $a = b$,

$$h' = 0,5 d_H \left(1,15 \frac{b}{d_H} - 1 \right) \left(1 + 0,805 \lg 1,15 \frac{b}{d_H} \right) \quad (3.13.18)$$

Lamellas:

The corridor light is calculated using formulas (3.13.17) and (3.13.18).

Chess piece in $a > b$

$$h' = 0,5 d_H \left(1,27 \frac{b}{d_H} \sqrt{\frac{a}{b} - 0,3} - 1 \right) \left(1 + 0,805 \lg \left(1,27 \frac{b}{d_H} \sqrt{\frac{a}{b} - 0,3} \right) \right) \quad (2.71)$$

When $a = b$,

$$h' = 0,5 d_H \left(1,065 \frac{b}{d_H} - 1 \right) \left(1 + 0,805 \lg 1,065 \frac{b}{d_H} \right) \quad (2.72)$$

TASKS

1. In a horizontal shell-and-tube heat exchanger, film condensation of dry saturated steam occurs on the outer surface of the tubes at a saturation pressure of 0.3 mPa. The outer diameter of the tubes is 18 mm. The wall surface temperature is $t_{cm} = 20^\circ\text{C}$.

Determine the heat transfer coefficient from the condensed steam to the surface of the tubes located in the first, third, and ninth rows of the staggered stack.

2. In a horizontal two-pipe condenser, dry saturated steam is condensed in an inner pipe with a diameter of R12 dBH/dH=14/16 mm. The length of the working part of the condenser is 1 m. The saturation pressure $p_H = 5,76 \cdot 10^5$ Pa.

Determine the temperature of the inner wall of the pipe at $m_t = 0.558$ kg/(sm) of condensed steam.

3. Condensation of dry saturated vapor of refrigerant R12 occurs in a horizontal shell-and-tube condenser with external fins. The saturation temperature is 30°C , the average temperature of the fin surface and inter-fin spaces is 28°C . The layers are

made of round copper with a thickness of $\delta = 2$ mm and a fin spacing of $S = 4$ mm. The fin diameter is $D_p = 30$ mm, the tube diameter $d_H = 20$ mm. The number of vertical rows of tubes is $n = 10$. Determine the heat transfer coefficient.

4. In a finned shell evaporator of a refrigeration machine, the boiling point of the refrigerant R12 must be above $t_h = -20$ °C and the heat flux density $q = 6500$ W/m².

Determine the temperature of the evaporator wall.

5. In the horizontal evaporator of a refrigeration machine, the refrigerant R12 boils in a flowing film. The saturation temperature $t_H = -30$ °C, the wall surface $t_{cm} = -28$ °C. The outer diameter of the pipe $d_H = 18$ mm.

Determine the linear heat flux density.

6. Film condensation of dry saturated water vapor occurs on the surface of a vertical pipe at a height of 3 m. The vapor pressure is $p = 0,04 \cdot 10^5$ Pa. The pipe surface temperature is $t_c = 20$ °C.

Determine the value of the linear heat transfer coefficient at a distance of $x = 2$ m from the upper end of the pipe if the condensate film flow is laminar along the entire height of the pipe.

7. Dry saturated water vapor condenses on the outer surface of a horizontal pipe with a diameter of $d = 32$ mm and a length of 2 m at a pressure of $p = 100$ kPa. The temperature of the pipe surface is 94.5 °C.

Determine the average heat transfer coefficient from the steam and the pipe and the amount of steam condensed on the surface, in m_t kg/h.

8. A steam boiler with a total heating surface $A = 15$ m² produces moist saturated steam with a pressure of 0.3 mPa and a dryness fraction $x = 0.95$. The wall temperature is 145 °C.

Determine the amount of water vapor produced in the boiler in 1 hour.

9. Ammonia boils in a vertical pipe with an internal diameter of $d = 35$ mm. The saturation temperature is $t_H = 0$ °C and the wall temperature is $t_{cm} = 3$ °C.

Determine the heat transfer coefficient from ammonia to the wall.

10. R12 boils in the evaporator tubes of a refrigerator. Saturation temperature $t_H = -12$ °C, tube surface $t_{cm} = -7$ °C. Mass flow of refrigerant $m_t = 85$ kg/h; surface heat flux density $q = 5220$ W/m².

Determine the inner diameter of the pipe.

11. Boiling water moves at a speed of $w = 1$ m/s in a pipe with an internal diameter of $d_{eH} = 32$ mm. The water is under a pressure of $p = 500$ kPa.

Determine the heat transfer coefficient from the wall of the pipe to boiling water if the internal surface temperature is 154 °C.

12. In a tubular water heater, water moves through the pipes at a speed of $w = 1$ m/s. At the same time, the water temperature rises from 20 to 80 °C. The inner diameter of the pipe is 18 mm, the length is 25 m. The average wall temperature is $t_c = 82$ °C.

Determine the heat transfer coefficient α from the pipe surface to the moving water.

13. The heat exchanger for cooling water is a tank-accumulator with a coil inside. The sodium chloride solution circulates through the spiral. The radius of the coil is 0.7 m, the diameter of the coiled pipes is 18 mm. The average temperature of the inner surface of the pipe is $t_c=0^\circ\text{C}$ and the solution $t_m=-3^\circ\text{C}$.

The speed of movement of the solution is $w=1.1$ m/s, the concentration of the solution is $\varepsilon=11\%$.

Determine the total length of the coil if the heat flux transferred to the solution is $\Phi = 10$ kW.

14. Calculate the surface heat flux density from the surface of a horizontal cylindrical heat exchanger when cooled by a free air flow. The outer diameter of the heat exchanger housing is $d_H=500$ mm. The surface temperature is 150°C , and the air temperature in the room is $t_{\text{room}} = 20^\circ\text{C}$.

15. A rectangular conductive tire with a cross section of 100x3 mm is located at one end and is cooled by a free flow of air with a temperature of $t_{\text{room}}=25^\circ\text{C}$. The temperature of the tire under long-term load conditions should not exceed 70°C .

Determine the heat transfer coefficient α from the tire surface and the permissible current in the bus under the specified conditions, if the resistivity of the tire material is $\rho = 0,13$ Ohm $\cdot$ mm²/m.

16. If the cylinder head temperature is 90°C and the cover wall temperature is 20°C , determine the equivalent heat transfer coefficient and surface heat flux density between the cylinder head and the engine cover.

17. Milk with a fat content of 3.5% is cooled in a cylindrical tank with a height of $h = 1.5$. The temperature of the tank walls is $t_{cm}=5^\circ\text{C}$, and the temperature of the milk is 35°C . There is no mixing.

Determine the heat transfer coefficient and surface heat flux density.

18. The temperature in the refrigerator compartment is $t_{cm}=-20^\circ\text{C}$. The loss of cold occurs only through the ceiling, its temperature is $t_c=-17^\circ\text{C}$. The air in the room is at rest.

If the ceiling size is 4x5 m, determine the loss of cold through the ceiling. Thermal radiation is not taken into account.

The reduced height of the edge of the circle h' is found using the formula (2.66).

$$h' = \frac{D_p - d_H}{2} \left(1 + 0,35 \ln \frac{D_p}{d_H} \right) = \frac{26 - 20}{2} \left(1 + 0,35 \ln \frac{28}{20} \right) = 3,787 \text{ mm}$$

We find the auxiliary quantity m using the following formula

$$m = \sqrt{\frac{2\alpha}{\lambda_m \delta}} = \sqrt{\frac{2 \cdot 2533}{386 \cdot 0,002}} = 81$$

Then $mh' = 81 \cdot 0,003787 = 0,307$

We find $thmh' = 0.292$ by application.

$$E_p = \frac{0,292}{0,307} = 0,95$$

Using (3.13.11), we determine the efficiency coefficient of the rib

$$h_p = \frac{\pi}{4} \left(\frac{D_p^2 - d^2}{D_p} \right) = \frac{3,14}{4} \left(\frac{0,026^2 - 0,02^2}{0,026} \right) = 8,33 \cdot 10^{-3}$$

$$A_b = 0,5 \cdot 3,14 \frac{0,026^2 - (0,020)^2}{0,004} = 0,1083 m^2$$

$$A_e = 3,14 \cdot 0,002 \left(1 - \frac{0,002}{0,004} \right) + \frac{3,14 \cdot 0,0026 \cdot 0,002}{0,004} = 0,072 m^2$$

$$E_p = 1,1 \left(\frac{0,108}{0,189} \right) \cdot 0,95^{0,75} \left(\frac{0,020}{8,33 \cdot 10^{-3}} \right)^{0,25} + \left(\frac{72 \cdot 10^{-3}}{0,18} \right) = 1,19$$

The heat transfer coefficient α is found using the formula (3.13.10)

$$\alpha_{np} = \alpha \varepsilon_p n^{-0,167} = 2533 \cdot 1,19 \cdot 10^{-0,167} = 2052 \text{ W/(m}^2 \cdot \text{K)}$$

$\delta = 2$ mm, rib step $S_p = 4$ mm. Rib diameter $D_p = 26$ mm, pipe diameter $d = 20$ mm.

The number of vertical rows of pipes is $n = 10$.

Determine the heat transfer coefficient.

Solution. The average heat transfer coefficient during condensation in a layered tube bundle is determined by the formula (3.13.10).

$$\alpha_{np} = \alpha \varepsilon_p n^{-0,167}$$

$$\text{where, } \alpha = 0,728 \sqrt[4]{\frac{\lambda_{\text{жс}}^3 \rho_{\text{жс}}^2 g z}{\mu_{\text{жс}} (t_H - t_c) d}}$$

The thermophysical properties of R22 at $t_H = 81^\circ \text{C}$ are determined by the formula $z = 1,7602 \cdot 10^5$ J/kg; other parameters $\mu_{\text{жс}} = 2,295 \cdot 10^{-4}$ Pa·s; $\lambda_{\text{жс}} = 0,0826$ W/(m·K);

We determine $\rho^1 = 1176$ kg/m³ at $t_{cp} = 0,5(t_H + t_c) = 0,5(81 + 29) = 30^\circ \text{C}$.

Thermal conductivity of copper

$$\alpha = 0,728 \sqrt[4]{\frac{0,0826^3 \cdot 1176^2 \cdot 9,81 \cdot 1,7602 \cdot 10^5}{2,295 \cdot 10^{-4} (31 - 29) \cdot 0,02}} = 2533 \text{ W/(m}^2 \cdot \text{K)}$$

The efficiency coefficient of the rib is found using the formula $E_p = thmh' / mh'$.

We determine the thermophysical properties of R22 using $t_H = 20^\circ \text{C}$. $Z = 188,41$ 10³ J/kg $\delta = 90,12 \cdot 10^{-4}$ N/m $\rho^1 = 1213$ kg/m³; $\rho^4 = 38,760$ kg/m³; $\lambda_{\text{жс}} = 0,082$ W/(m·K); $\mu_{\text{жс}} = 2,383 \cdot 10^{-4}$ Pa·s; $\gamma_m = 0,197 \cdot 10^{-7}$ m²/s.

$$\alpha = 0,728 \sqrt[4]{\frac{0,082^3 \cdot 1213 \cdot 9,81 \cdot 18841 \cdot 10^3}{2,383 \cdot 10^{-4} \cdot \Delta t \cdot 0,020}} = 3066 \cdot \Delta t^{\frac{1}{4}}$$

$$q = \alpha_t \Delta t; \quad q = \frac{Q}{\alpha}$$

$$We = \frac{90,12 \cdot 10^{-4}}{9,81(1213 - 38,760) \cdot 0,020^2} = \frac{90,12 \cdot 10^{-4}}{4,608} = 1,955 \cdot 10^{-3};$$

$$\alpha_t = 0,2 \cdot 1,955^{-0,25} \cdot 10^{-3} \cdot 3666 \cdot \Delta t^{-0,25} = 2866 \Delta t^{-0,25}$$

$$q = 128,95 \cdot \Delta t^{-0,25} \cdot \Delta t = 128,95 \Delta t^{0,75}$$

$$\Delta t^{0,75} = q = 950 = 0,325; \Delta t = 0,325^{1/0,75} = 0,223$$

$$t_H - t_C = 0,223; t_C = t_H - 0,223 = 20 - 0,223 = 19,76$$

Example-2. Condensation of dry saturated vapor of R22 occurs in a shell-and-tube horizontal condenser with external fins. Saturation temperature $t_H = 31^\circ\text{C}$, average surface temperature of the fins and inter-fin areas $t_{cm} = 31^\circ\text{C}$, average surface temperature of the fins and inter-fin areas $t_{cm} = 29^\circ\text{C}$. Copper fins, round, thickness $\mu_m = 9,483 \cdot 10^{-4} \text{ Pa}\cdot\text{s}$; $\rho_{жс} = 997,5 \text{ kg/m}^3$.

For the first row of a pipe bundle, α is determined by the formula (3.13.5).

$$\alpha = 0,728 \sqrt[4]{\frac{0,6017^3 \cdot 997,5^2 \cdot 9,81 \cdot 2,43 \cdot 10^6}{9,433 \cdot 10^{-4} \cdot (30 - 16) \cdot 0,019}} = 8786 \text{ Vt}/(\text{m}^2 \cdot \text{k})$$

We determine α_3 and α_{10} using formula (3.13.9)

$$\alpha_3 = \alpha \cdot \varepsilon_3 = 8786 \cdot 0,83 = 7292,68 \text{ Vt}/(\text{m}^2 \cdot \text{K})$$

$$\alpha_{10} = \alpha \cdot \varepsilon_{10} = 8786 \cdot 0,68 = 5974,48 \text{ Vt}/(\text{m}^2 \cdot \text{K})$$

Example-3. Condensation of dry saturated R22 vapor in the inner tube of a horizontal shell and tube apparatus occurs at a saturation pressure of $p_H = 916.93 \text{ kPa}$. The inner diameter of the tube is $d_{\text{вн}} = 20 \text{ mm}$.

If the surface heat flux density is $q = 950 \text{ W/m}^2$, determine the surface temperature of the pipe wall.

Solution. The average heat transfer coefficient depending on the total internal surface can be calculated using the formula $\alpha_m = 0,2 \alpha We^{-0,25}$. Here α is determined using the formula (3.13.5)

$$We = \frac{\sigma_m}{g(\rho' - \rho''z)d_{\text{вн}}^2} \quad - \text{Viber criteria.}$$

We determine $Re = 0,95z^{0,78} \varepsilon_t = 0,95 \cdot (525)^{0,78}$;

$$\varepsilon_t = \left(\left(\frac{\lambda_c}{\lambda_m} \right)^3 \left(\frac{M_m}{M_c} \right) \right)^{\frac{1}{8}} = \left(\frac{303 \cdot 236 \cdot 10^{-6}}{297,08 \cdot 10^{-6}} \right)^{\frac{1}{8}} = 125,73$$

Heat transfer coefficient α

$$\alpha = \frac{Re z \rho v}{(t_H - t_C)H} = \frac{132,35 \cdot 2258 \cdot 0,295 \cdot 10^{-6} \cdot 959}{(99,64 - 94,5) \cdot 2} = 7813 \text{ W}/(\text{m}^2 \cdot \text{k})$$

The amount of steam that condenses on the pipe surface

$$m_t = \frac{\pi d H \alpha \Delta t}{z} = \frac{3,14 \cdot 2 \cdot 10^{-2} \cdot 2 \cdot 7813 \cdot 5,14}{2258 \cdot 10^3} = 2,23 \cdot 10^3 \text{ kg/h.}$$

Example-4. In a horizontal shell-and-tube apparatus, film condensation of dry saturated water vapor is carried out on the outer surface of the tubes at a saturation pressure of 4.242 kPa. The outer diameter of the tubes $d_H = 19$ mm. The surface temperature $t_c=16^\circ\text{C}$.

Calculate the heat transfer coefficient from the condensed steam.

To the surface of the tubes located in the first, third, and ninth rows of the hardened set (counting from the top).

Solution. Based on the applications at the given saturation pressure, we determine the saturation temperature at $p_H=4,2442$ kPa, $t_H=30^\circ\text{C}$,

With $z=2,43 \cdot 10^6$ J/kg $t_{cp}=(30+16)0,5=23^\circ\text{C}$, we find $\lambda_{\text{жс}}=0,6017$ W/(m·K).

CHAPTER IV. PASSIVE SOLAR HEATING SYSTEMS

4.1. Passive solar heating systems and their analysis

The increase in energy consumption in connection with the growth of the world's population, as well as the development of the economy, the increase in the cost of purchasing energy resources, the depletion of reserves of fossil energy resources force us to think about creating energy-saving methods in construction. To do this, it is necessary to develop approaches aimed at rational use of energy consumption, as well as the transition to renewable energy sources, since the construction industry is one of the sectors of the economy with great potential in terms of implementing energy saving programs. In addition, greenhouse gases emitted into the atmosphere during fuel combustion lead to a change in the ecological state of our planet and, as a result, a significant deterioration in human living conditions. As a result, it is necessary to take effective and high-quality measures to protect the environment by reducing the production of pollutants. Based on the above, there can be no doubt about the usefulness and relevance of organizing the construction of buildings using environmentally friendly, renewable energy sources. The transition to this type of building allows for a significant reduction in fuel and energy consumption, energy costs, and greenhouse gas emissions.

Uzbekistan, along with many countries around the world, is taking active steps to reduce greenhouse gas emissions - a vivid example of this is the ratification of the Paris Climate Agreement in 2018. The Resolution of the President of the Republic of Uzbekistan №. PQ-4477 dated October 4, 2019 “On approval of the Strategy of the Republic of Uzbekistan for the transition to a “green” economy for 2019-2030” was adopted. At a conference held in Glasgow (Great Britain) in November 2021, Uzbekistan renewed its commitment to reduce greenhouse gas (GHG) emissions from the previously set 10% to 35% of the base 2010 level per unit of GDP by 2030. To achieve the ambitious goals mentioned above, it is necessary to create institutional, regulatory and financial mechanisms for investments in increasing the energy efficiency of buildings in Uzbekistan. Buildings are a significant source of energy savings and can have a positive and long-term impact on the entire energy sector through the large-scale implementation of energy efficiency programs.

The building stock is important for ensuring comfortable thermal conditions in buildings, reducing energy intensity, and increasing the level of thermal insulation due to its unused reserve. Another reason for the high energy consumption of buildings is that most buildings were built before 2011, when energy efficiency requirements were first made mandatory by KMK-2.01.18-2018.

According to the Ministry of Energy of the Republic of Uzbekistan, in 2010-2019, the absolute volume of greenhouse gas emissions in Uzbekistan increased by

30.6%, including emissions from power plants by 65.2%, and at the end of 2019, the share of the energy sector in the country's greenhouse gas emissions was 19.5%. In the context of the unfavorable situation with greenhouse gas emissions, the government has made efforts to promote a “green” agenda in the country, in accordance with which it has set a goal of doubling energy efficiency and reducing carbon dioxide emissions relative to gross domestic product.

Currently, there are about 8.4 million buildings and structures in Uzbekistan, of which 868 million m² are heated areas. The total construction stock of residential buildings accounts for 91%, and non-residential buildings account for the remaining 9%. Among residential buildings, individual private housing accounts for the largest share. Calculations and analyses show that the main potential for increasing energy efficiency in economic sectors is in the sector of public and private housing and communal services. In particular, more than 50% of the total energy resources in the Republic are consumed in heating and air conditioning systems of buildings and structures, and 10% in the hot water supply system. It follows that it is in the housing sector of the economy that there are huge opportunities for energy saving and the introduction of energy-efficient technologies.

As is known, solar heating systems are systems that use solar radiation energy as a heat source. What distinguishes these systems from other low-temperature heating systems is the use of a special element - a solar collector - designed to receive solar radiation and convert it into thermal energy.

Usually, solar thermal energy devices are used that are designed to receive solar radiation energy and partially replace the heating load (30-40%).

Solar heating systems are classified into the following main types - according to the technology of converting solar energy into heat; according to the type of load being replaced; according to the type of primary heat carrier in active systems; according to the volume of heat storage tanks; according to the capacity of the heat carrier.

Solar heating systems can be divided into the following groups: active and passive heating systems.

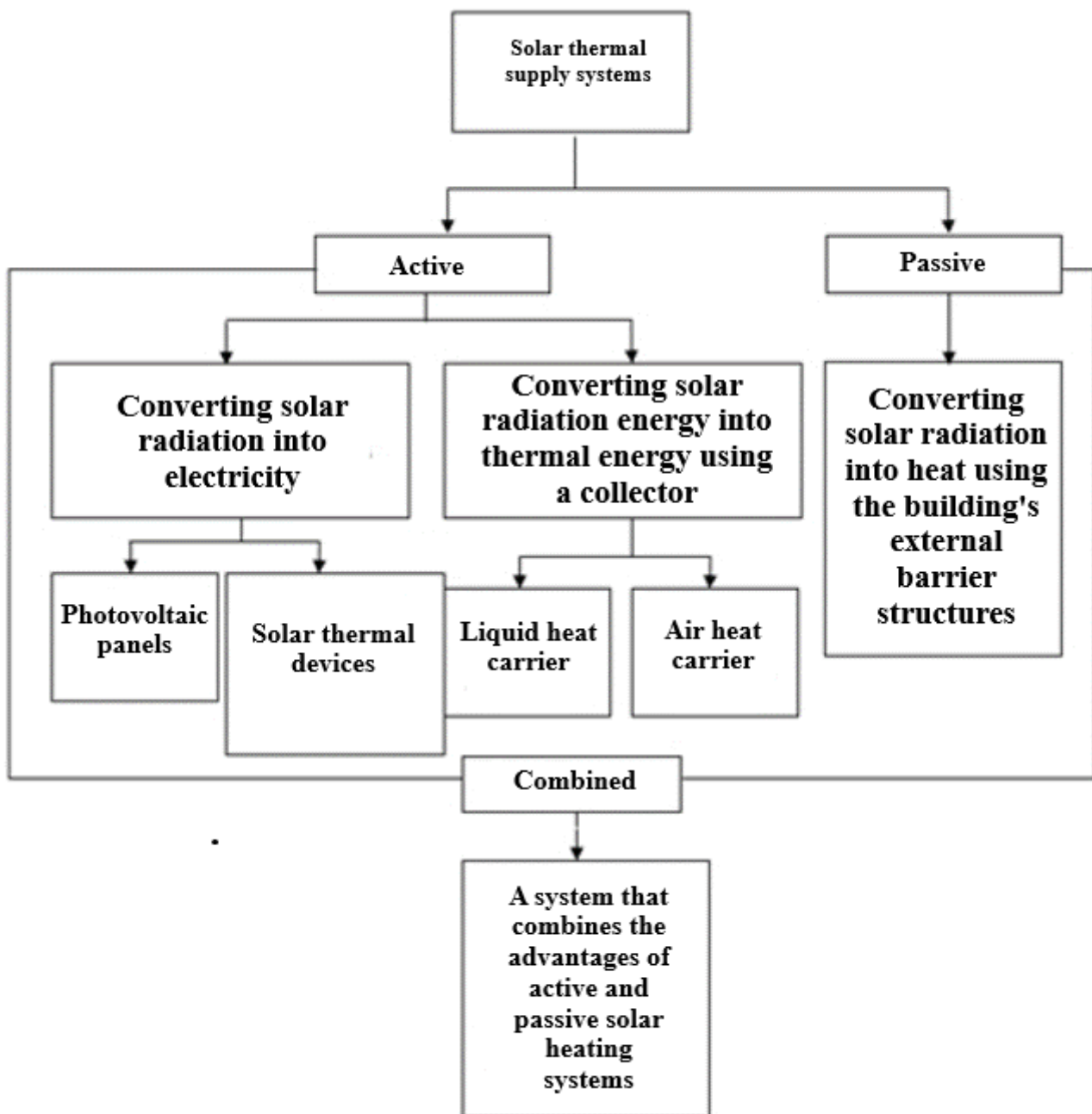
Passive solar heating systems are those in which the building itself or its individual barriers (collecting building, collecting wall, roof collector, etc.) serve as an element that receives solar radiation and converts it into heat.

The experience of developing, constructing and operating projects of structures with solar heating systems, the analysis and generalization of world achievements in this area show that one of the most effective systems is passive solar heating systems, which are mainly distinguished by their simplicity in terms of constructive solutions.

Passive solar heating systems combine the functions of receiving and transmitting heat directly through the walls of the building. Such buildings do not require additional costs, they automatically receive and accumulate solar radiation

energy. As is known, when designing simple heating systems, the system capacity is determined by the condition of providing heat in the coldest period of the year. At the same time, the system operates with a certain load, mainly compensating for real heat losses. In turn, such a load on the heating system can be reduced when designing the system by taking into account the solar radiation energy falling during the day. Designing a building in such a way as to receive maximum solar radiation heat can not only significantly reduce energy consumption for traditional heating, but in some cases, eliminate such costs.

Passive solar heating systems and their analysis



According to the European classification system for the classification of buildings and structures, energy-efficient buildings are divided into the following types:

- Energy-efficient house - houses with an average energy consumption for heating of not less than 60-70 kWh/(m²·year);
- Passive house - houses with an average energy consumption for heating of not more than 15 kWh/(m²·year) per year;
- Zero-energy building - a building with the same architectural standard as a passive house, but designed for self-generated energy consumption, with an average energy consumption for heating of 0 kWh/(m²·year).

An active (“Energy Plus” type) house is a building that, due to the use of special engineering systems, produces more energy than it consumes.

A passive house is a house with a relatively low negative impact on the environment, highly energy-efficient and equipped with all the amenities for living.

The main criterion for a passive house is to create a solid building shell with a high level of thermal insulation and a thermal conductivity of <0.15 W/(m²·K).

- The following criteria are taken into account when designing and building passive houses:
- “thermal bridges (heat circulation)” - prevention of heat loss;
- compactness of the structure;
- passive use of solar energy due to the south-facing orientation of the building and the absence of shading;
- special high-quality windows and window frames with a thermal conductivity coefficient of <0.8 W/(m²·K) and energy transfer - about 50 percent;
- air exchange rate: $n_{50} = 0.33x^{-1}$;
- heat recovery from recycled air, with a heat retention percentage of > 75%;
- highly efficient energy-saving devices for domestic use;
- water heating using solar collectors or a heat pump;
- passive air heating, for example, using a ground-to-air heat pump.

If we look at the global experience of passive houses, the first eco-house was built in Manchester, New Hampshire, in 1972, long before the first passive house was built and the energy crisis in the United States (Figure-4.1.1).



Figure-4.1.1. The first eco-home built in the United States.

The house is designed in the form of a cube, the architectural compactness of which allows to minimize heat loss. The glass layer accounts for only 10% of the wall surface, and solar collectors are installed on the roof of the house. The external walls consist of a two-layer structure, and the windows are equipped with sun-shielding barriers. The architectural solution of the building also allows for the optimal layout of the internal rooms, maximum use of natural lighting. The possibility of using recuperative ventilation and special tanks for storing cooled and heated water was also considered.

The second most energy-efficient house is the “EKONO-house” building, built in Otaniemi, Finland (1973-1979). The following main innovative energy-saving solutions were achieved in the design of the “EKONO-house” building:

- effective use of the internal volume of the building to reduce the area of enclosed structures and reduce heat loss through them;
- creation of an effective heat-shielding layer of barrier structures to reduce heat loss;
- use of barrier structures with high heat capacity to increase the thermal resistance of the building and accumulate heat; accumulation of solar radiation heat at the base of the structure to reduce the load on the heating system;
- use of windows with air ventilation properties to reduce heat flow in the summer season and heat loss to the outside in the winter season;
- minimizing air leakage in the ventilation system (building tightness) and external air flow consumption in order to reduce energy consumption in heating the building;
- efficient lighting in order to reduce electricity consumption; an automatic control system for climate and lighting equipment was used to optimize energy consumption and maintain energy consumption records.

The annual specific heat consumption of the first section of the EKONO-house was 124 kWh/m², which was 50% lower than that of office buildings in Finland. Similar buildings in the USA had even higher specific heat consumption values. The annual specific electricity consumption for heating the first section was 79 kWh/m² per year, which was lower than the figures in Finland and the USA. The annual specific heat consumption for heating the second section of the EKONO-house was 70 kWh/m² per year, and the specific electricity consumption for heating was 57 kWh/m² per year, which is 1/3 of the energy consumption of conventional houses of this type.

The first passive house was built in Darmstadt, Germany, in 1991. (Figure-4.1.2). This project confirmed the perfect functioning of all important systems in the operation of this type of house.

While the average energy consumption for heating has remained unchanged (less than 10 kWh/(m²·year)), the savings compared to a conventional house have been more

than 90 percent, and relatively high indoor air quality and comfortable temperature conditions have been noted.



Figure-4.1.2. General view of a passive house built in Germany in 1991.

The Passive House Institute was established to conduct and continue extensive research on the development and improvement of passive house performance. As of May 2011, more than 32,000 certified passive houses have been built, and from 2020 the passive building standard (average annual energy consumption for heating 15 kWh/m²) has been made mandatory in the European Union.

4.2. Environmental aspects of building passive houses

Today, the heat and electricity produced around the world is characterized by a high level of “carbon content”, since the share of renewable energy sources in the total volume of energy resources is still quite low. In 2018, the Paris Agreement proposed by the government of Uzbekistan to the UN Framework Convention on Climate Change was ratified, according to which Uzbekistan expressed its “readiness to contribute to eliminating the consequences of climate change”. Currently, the country faces important tasks in creating and implementing a regulatory framework for supporting renewable energy and mechanisms for entering the international trading market for quotas applied to emissions [21, 96-b, 14].

According to the Ministry of Energy of the Republic of Uzbekistan, in 2010-2019, the absolute volume of greenhouse gas emissions in Uzbekistan increased by 30.6%, including emissions from power plants by 65.2%, and at the end of 2019, the share of the energy sector in the country's greenhouse gas emissions was 19.5%. In the context of the unfavorable situation with greenhouse gas emissions, the government has increased its efforts to promote a "green" agenda in the country, and in accordance with it has set a goal of doubling energy efficiency and reducing carbon dioxide emissions relative to GDP (Table 4.2.1).

Table-4.2.1

The amount of CO₂ emissions released into the atmosphere as a result of fuel combustion (per 1 ton of fuel)

Fuel types	(CO ₂) emissions volume, t
Coal	2,29
Brown coal	1,45
Briquette	2.12
Coke	2,8
Natural gas	1,88
Fuel oil	3,16
Diesel fuel	3,12

The U.S. Environmental Protection Agency (EPA) tracks total greenhouse gas emissions in the United States by publishing its annual National Greenhouse Gas Inventory Report. This annual report estimates total domestic emissions and removals of greenhouse gases from anthropogenic activities across the United States. The most recent report available is from 2017, and the sources of greenhouse gas emissions are as follows:

- Transportation (28.9% of greenhouse gas emissions).
- Electricity generation (27.5%).
- Industry (22.2%).
- Commercial and residential buildings (11.6%).
- Agriculture (9.0%).

Greenhouse gas emissions in agriculture are associated with livestock (the main source of emissions is livestock), soil management in agriculture (the use of synthetic and organic fertilizers, the cultivation of nitrogenous crops, soil drainage), and rice cultivation.

The main feature of the “passive” system is that its efficiency almost entirely depends on the choice of an optimal design solution. This solution can only be implemented using analytical models that take into account the non-stationary and non-continuous nature of thermal processes. An important feature of the analytical solution of such problems is the possibility of obtaining an engineering approach, which is extremely necessary at this stage of the development of “passive” systems.

In a passive solar heating system, the building structure is designed to collect solar radiation. In such a heating system, the walls, ceilings or floor coverings of buildings and structures directly receive sunlight. There are also systems that provide special elements for heat storage built into the building structure (for example, stone boxes or glass containers filled with water). Such systems are also classified as passive solar systems. Passive solar buildings are comfortable to live in, harmoniously

combined with nature, and due to the abundance of natural light, not only heat is saved, but also electricity. A passive solar heating system includes a number of elements that perform the function of absorbing heat during the day and maintaining the temperature in the house for a certain period of time, such elements may include brick walls, stone floors, large windows and doors, etc. To properly use passive solar energy, active movement of warm air throughout the house is required. When the doors of the house are open, natural circulation is sufficient. Although passive system equipment design does not include mechanical devices, fans are sometimes included in the design. Glass, which easily transmits short-wave radiation, is a poor conductor of long-wave radiation. After solar energy enters through the glass and is absorbed by some object behind it, thermal energy is transferred to the outside by radiation. This phenomenon is called the “greenhouse effect”. The principle of operation of most devices that collect solar radiation and convert it into heat is based on the greenhouse effect.

Passive solar systems allow you to reduce the cost of electricity consumption by almost 50%. It is worth noting that, with full use of the principles of passive solar heating systems, the financial costs of building such a house will be the same as for a conventional house.

Passive solar heating systems are designed to collect solar heat in large-sized elements of buildings in a natural way - through south-facing windows. At the same time, energy savings due to the use of passive solar heating systems are 25-30% and directly depend on the climatic characteristics of the location.

The main characteristics of such solar heat supply systems are presented in Table 4.2.2.

Table-4.2.2

Description of passive energy systems

Type of solar installation	Type of solar power system	Description of planned technical activities	Methods of organizing solar energy supply
Passive systems			
Traditional construction buildings (without special equipment)	Skylights, greenhouses, sun windows	The main rooms are oriented to the south; the plan is elongated in width	Thermal protection layer. Large-area glazing. Auxiliary energy system. Natural air convection
Buildings modified using structural elements	Heat-collecting wall-collector; thermal pools	Keeping northern facades to a minimum, placing utility rooms on the north side of the building	Thermal insulation. Massive barriers. Large windows. Auxiliary energy system. Natural air convection.

The location of the buildings, the protective layer, the orientation of the windows and the heat load should be brought into a single system. To reduce internal temperature changes, protective coatings should be placed on the outside of the building, in internal heating areas, where insulation is required. Then the building structure will be suitable for any microclimate. If the right balance is achieved between the protective layer and the heat load in such buildings, energy and building materials will be saved, resulting in high economic efficiency.

Components of passive heating systems include:

- rooms illuminated by direct sunlight;
- water tanks located at the top of the building, designed to store energy;
- the use of pipes laid on the external surfaces of walls illuminated by sunlight;
- the use of systems used to fill window openings, barriers, and external walls of the building envelope that accumulate thermal energy;
- it is also possible to include thermosyphon ventilation systems to cool the air in heaters.

The effectiveness of any passive system directly depends on the type of glass used. Glass or other transparent materials retain short waves.

Windows regulate the flow of energy in two main ways:

- in winter, they provide the house with heat by transmitting solar energy into the building, when the air temperature inside the building exceeds the outside temperature;
- in summer, they help cool the building by reducing the level of sunlight penetration by properly positioning the window and shading it, and also use ventilation to cool the house. If the heat of solar radiation is used, it is necessary to ensure that it enters the room at the most beneficial time.

Typically, in the winter season, sunlight should enter the room between 9:00 and 15:00, when there should be almost no obstacles in their path. The shade of trees in the yard can block the rooms of the house from sunlight, and such aspects must be taken into account during construction. In design, the walls, floor and ceiling of the building are more important than the location of the internal components.

The glass materials from which windows are made usually transmit solar radiation in the range of 0.4-2.5 μm . As a result of the absorption and re-emission of light by opaque objects located in the room, its wavelength increases to 11 μm . Glass is considered a barrier that does not transmit electromagnetic waves of this length, and the light entering the room does not escape.

Passive systems have collectors and accumulators designed to concentrate and store solar radiation, which are external barriers with a high heat capacity. The conversion of solar energy into thermal energy can be achieved by heating it directly with sunlight and storing the heat in a large external casing that collects this heat and gradually returns it to the room.

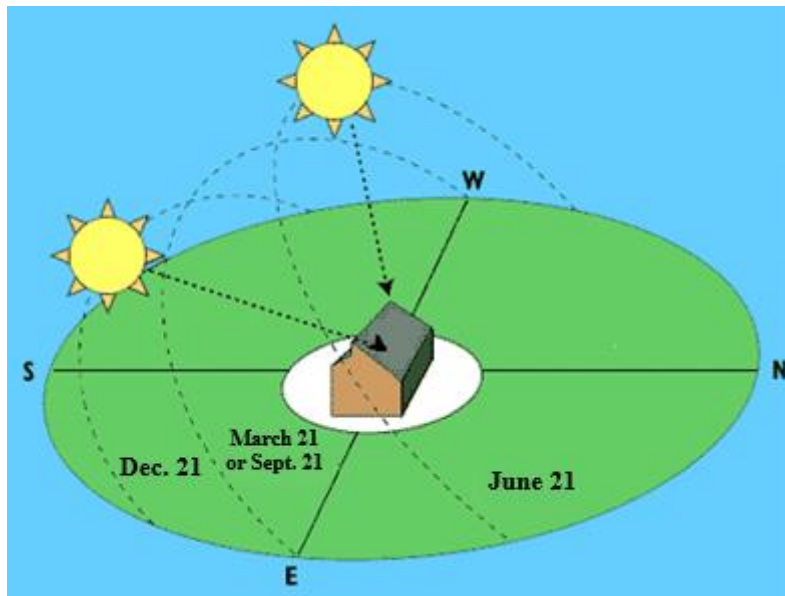


Figure-4.2.1. Recommended orientation of buildings in the Northern Hemisphere.

4.3. Disadvantages of Passive Solar Heating Systems

Sunlight can fade furniture and carpets.

If a room is exposed to direct sunlight for several hours at a time, it can overheat during the day. By trying to keep the room warm longer during the dark hours of the day, you can increase the temperature of some rooms during the day.

Some buildings are not at all suitable for passive solar heating systems. These may be buildings that are shaded from the sun by various objects, facing north, with few or no windows, or facing south. In such cases, passive solar heating systems are not suitable at all.

Windows that let in sunlight during the day also heat the room, but if they are not well designed, they lead to heat loss during the dark hours of the day. In this case, heat loss protection, i.e., sealing the windows, installing double-glazed windows, will help.

Passive solar heating systems are designed to collect solar heat in large-sized elements of buildings in a natural way - through transparent barriers facing south. At the same time, energy savings due to the use of passive solar heating systems are 35-45% and directly depend on the climatic indicators of the location.

The first experiments on heating through an insolation passive solar heating system were carried out in Uzbekistan 85 years ago under the leadership of the famous solar scientist and engineer V.B. Weinberg. According to his project, a small building of a solar laboratory was built in Kaplanbek (now in the Shymkent region of the Republic of Kazakhstan) in 1931. The results of observations showed that in cold sunny (open) weather, the laboratory room did not require heating, but in cloudy, windy weather, the building was additionally heated. In summer, heating was not observed in the laboratory building, since in the summer season of the year, due to

the high position of the sun above the horizon, the amount of solar radiation entering the laboratory room is 2-3 times less than in the winter season of the year.

In the climate of the state of Indiana, USA, when heating a house with an insolation passive solar heating system, with a double-layer glass thickness of 6 mm and a transparent barrier with an air layer thickness of 12 mm, without using any additional heating systems, the average room temperature was maintained within the range of $14\div 15^{\circ}\text{C}$ during the day at an outdoor temperature of $0\div 1^{\circ}\text{C}$. During the dark hours of the day, the indoor air temperature decreased by $+2^{\circ}\text{C}$, and during the day it increased by $+27^{\circ}\text{C}$. However, it should be noted that the total heat loss of this house was 16% higher than that of a house with a traditional heating system.

It was concluded that the use of passive solar heating systems during the heating season in the climatic conditions of Uzbekistan is one of the most optimal modern heating systems for the following reasons:

1) Uzbekistan is located in a geographical latitude with a favorable sunny temperate climate, from October 15 to April 15, that is, during the heating season, there are about 100 sunny days;

2) The cost of buildings adapted to passive solar heating systems is almost the same as that of conventionally built buildings;

3) In the climatic conditions of Uzbekistan, the design, construction and operation of buildings using low-temperature passive solar heating systems is one of the most modern and efficient heating methods, and in the future there are wide opportunities for researchers to put many inventions and scientific developments into practice.

4) The high economic, environmental and energy efficiency of heating buildings using passive solar heating systems for the climatic conditions of Uzbekistan, for example, in Europe the fuel replacement coefficient is 25-30%, while in Uzbekistan this coefficient is 35-80%.

The rational use of energy for heating, ventilation and air conditioning of buildings is ensured by the following methods:

- increasing the thermal resistance and air permeability of building envelopes, as well as reducing heat loss through enclosing structures by eliminating excess light gaps;

- reducing heat gain during the hot season using sun-shielding devices;

- reducing heat and cold losses from equipment, pipes and air ducts located outside heated rooms;

Use of heating, ventilation and cooling equipment with high energy efficiency;

Use of economically viable secondary and renewable energy resources.

4.4. Active solar heating systems and their analysis. Active solar heating systems.

- Active solar heating systems are heating systems that provide for the equipment of buildings with special technological devices that convert solar energy into thermal energy. Active low-temperature solar heating devices are installed separately and are not part of any building equipment. Currently, solar heating devices are divided into two types:
 - solar collectors;
 - flat solar devices.

Solar collectors are made in the form of spherical or paraboloid mirrors, and in the form of a paraboloid cylinder.

Flat solar devices are divided into two types:

- flat collectors;
- flat absorbers.

An active solar heating system provides more heat than a passive heating system and, by its very nature, is more technologically complex. Typically, such a solar heating system includes three components: a solar collector to absorb solar radiation, a system to store the energy received, and a heat exchange system to provide heat to residential buildings. Active systems are divided into air- or liquid-based types. The main characteristics of active solar heating systems are presented in Table 4.4.1.

There are also differences in active heating systems, which are reflected in the way in which the solar energy is concentrated in the solar collector. In liquid-based systems, the liquid performs this task, while in air-based systems, the energy is absorbed by the air. Typically, solar collectors are installed on the roofs of buildings, so it is advisable to place the outlet for the hot air in the ceiling. The installed system acts as a ceiling fan, mixing the air in the house.

Table-4.4.1

Description of active energy systems

Type of solar installation	Type of solar power system	Description of planned technical activities	Methods of organizing solar energy supply
Active systems			
Buildings with special devices in their structure	Flat plate collectors (water and air) accumulators	Compact size; Placing collectors on south-facing walls or south-facing roofs	The presence of special heat accumulators: storing water in containers for water heating; using gravel for air heating
A building with separate devices	Flat plate collectors, solar collectors, batteries	Compact in size; simple, unconstrained houses with solar heating	The presence of boilers, a special heat distribution system. Mechanical approach to heat distribution

Active solar heating systems, in turn, can be classified as follows:

- 1) by intended use: hot water, heating and combined hot water and heating systems;
- 2) by time of use during the year: seasonal and year-round;
- 3) by coverage of consumers: individual, grouped and centralized;
- 4) by the number of circuits: single-, double- and multi-circuit;
- 5) by the presence of a backup source;

Active systems involve the equipment of buildings with special technological devices that convert solar energy into thermal energy. Active devices designed to convert solar energy into heat using liquid-based collectors can be divided into the following main groups:

- evacuated tube collectors;
- flat-plate collectors equipped with single-layer selectively coated glass;
- flat-plate collectors equipped with single-layer non-selectively coated glass;
- absorbers without a transparent coating.

One of the most important indicators of devices that convert sunlight into thermal energy is their energy efficiency. It is expressed as the ratio of the energy (heat) flux received from the collector to the energy flux of solar radiation falling on its surface.

The value of energy efficiency is most affected by optical and thermal losses. Optical losses are determined by the transmittance of the transparent coating and the absorption capacity of the absorber. They are practically independent of temperature. Heat losses arise from heat exchange between the surfaces of the collector and absorber and the environment, and depend on the temperature difference of the surrounding air.

If the collectors are used in the warm season to heat a low-temperature ($Q=800 \text{ W/m}^2$) heat carrier, for example, water in open systems (the temperature of the heat carrier in the collector does not differ much from the outside air temperature), then collectors with selective coating and other types are covered with a single-layer glass coating with approximately the same efficiency - about 80%. In such conditions, the efficiency and the useful work coefficient of the most efficient absorbers without a transparent coating are about 90%, which is due to the absence of optical losses (reflection) in the transparent coating [43, 22-b, 44]. Evacuated-tube collectors are considered the least efficient (60%) due to the significant reflection of solar radiation when passing through the curved surface of the glass tubes. Since water heated by solar radiation energy can enter the accumulator at the time of drawing water from it to provide the heat load of the building, this system allows for independent control of the accumulator, additional energy source and the system, including the load. In such a system, a bypass line from the storage tank is designated for an additional energy source.

When using collectors for water heating in the hot season of the year, the temperature of the heat carrier may be higher than the ambient temperature (by 40-45°C). In such conditions, the efficiency of tubular, flat selective collectors, similar to the above type of collector, is about 55%. Similarly, the efficiency of selective uncoated collectors, in which heat transfer losses increase under these conditions, is also relatively small (40%). Due to the increased convective heat exchange between the heat carrier, which is not protected by a protective layer, and the surrounding air, uncoated open absorbers have the lowest efficiency, which can be reduced to 20%.

Figure 4.4.1 shows the principle scheme of active solar heating systems.

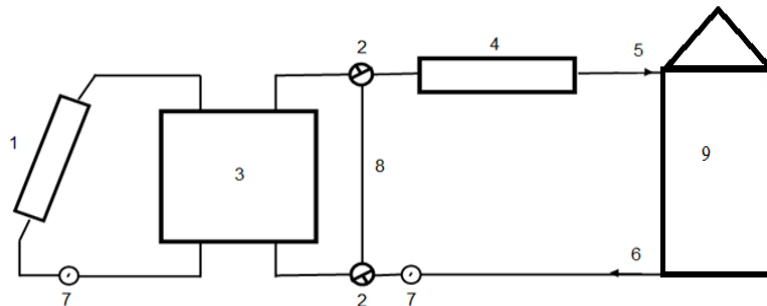


Figure-4.4.1. Schematic diagram of an active solar heating system. 1 – solar water heater; 2 – three-way valve; 3 – storage tank; 4 – additional heat source; 5 – entrance to the room; 6 – exit from the room; 7 – pump; 8 – bypass line of the storage tank; 9 – heat load receiver (Subscriber).

If, in the autumn-spring period of the year, the intensity of the solar radiation flux decreases to $Q = 400 \div 500 \text{ W/m}^2$, then even for the use of solar collectors for low-temperature heating systems and hot water supply systems, it is sufficient that the temperature difference between the outside air and the heat carrier is 40÷45°C. In such conditions, the highest efficiency (45%) is demonstrated by a vacuum-tube collector (due to the almost complete elimination of heat loss by convection due to the creation of a high vacuum in the pipes). In such conditions, collectors without selective coating and open absorbers made of single-layer glass with a selective coating of only 28% are not used.

From the above, it can be concluded that open absorbers can be effective even in conditions of high solar radiation intensity and low heat carrier temperature (seasonal water heating in the collector). Collectors with absorbers without selective coating can be used for seasonal water heating in hot water systems in Uzbekistan. Collectors with high-quality selective coating can be used for permanent (year-round) water heating for IST (hot water supply) systems and for low-temperature heating systems in the autumn-spring period of the year. Vacuum-tube collectors can be used for heating systems all year round. In liquid-based collectors, a liquid heat carrier is heated - water or a special non-freezing liquid (Figure 4.4.2).

An example of the use of this type of system is a single-circuit system with passive circulation of heat carrier: collectors, storage tanks and connecting pipes of the system are filled with liquid. Solar radiation passing through the transparent coating of the collector heats its absorber panel and the liquid in the channels.

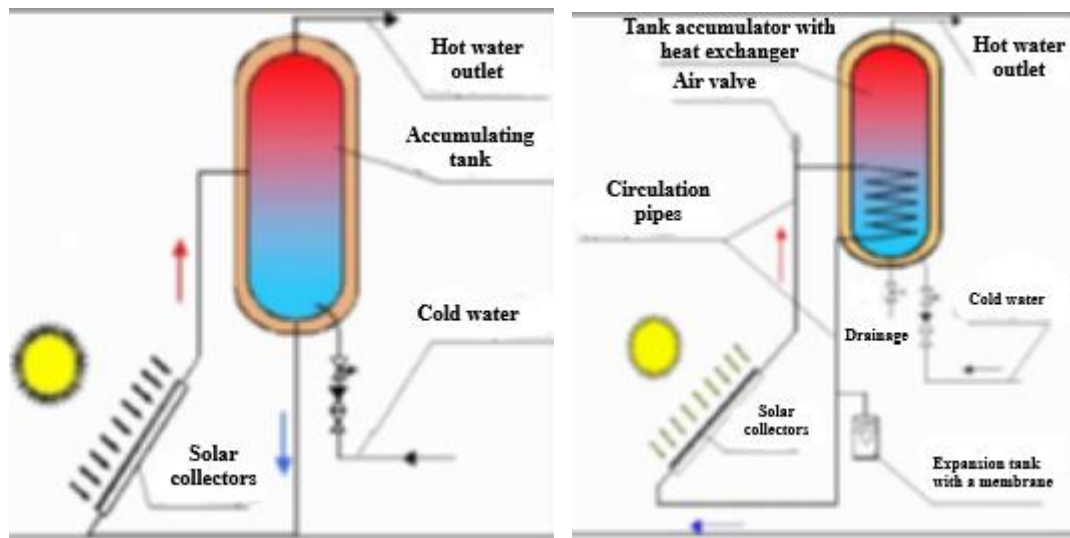


Figure-4.4.2. Liquid-based solar collector. Single-circuit and double-circuit system.

When heated, the density of the liquid decreases and the heated liquid begins to move to the upper point of the collector, and then through the pipe to the tank-accumulator. The heated liquid in the tank reaches the upper point, and the liquid with a lower temperature falls to the lower part of the tank, that is, the process of occupying layers depending on the temperature is observed. The cold liquid from the lower part of the tank flows along the pipe to the lower part of the collector. Thus, with sufficient solar radiation, a constant circulation process occurs in the collector circuit, the speed and intensity of which depend on the density of the solar radiation flux. Gradually, during the daylight hours, the tank heats up completely, and water for use must be taken from the layers located in the upper part of the tank. This is usually done by forcing the heated hot water from the tank by pumping cold water under pressure from below.

The principle of operation of a two-circuit system is similar to that of a single-circuit system, the system has a separate closed circuit consisting of a collector, pipelines and a tank-accumulator, a heat exchanger. This circuit is filled with a special non-freezing heat carrier. After being heated in the collector, the heat carrier passes to the upper part of the heat exchanger, gives off its heat to the water in the tank-accumulator and cools down, then moves to the lower part of the collector, ensuring constant circulation with the participation of solar radiation.

According to the operating conditions, by installing a three-way barrier in appropriate cases, the system can operate in the following four modes:

Mode 1: if the building is receiving solar energy and there is no need to heat the building, then all the heat received from the solar heater is stored in the heat accumulator;

Mode 2: if the building is receiving solar energy and there is a need to heat the building, all the heat received from the solar heater is used for the building's heat load;

Mode 3: if the building is not receiving solar energy and there is a need to heat the building, and there is a heat reserve in the heat accumulator (accumulator), the building's heat load is provided with the stored heat;

Mode 4: If the building is not receiving solar energy and there is a need to heat the building, and the heat reserve in the heat accumulator (heat storage tank) has been used up, an additional energy source is used to provide the heat load.

In real systems (for example, in the warm season), a fifth mode may also occur, i.e. the accumulator is saturated with energy, there is no heat load, and the collector is generating heat. In such conditions, it is not possible to use or accumulate the absorbed energy and it must be directed directly to another consumer, i.e. additional operating modes, such as the operation mode for hot water supply, should be provided. The advantages of water heating systems are represented by the fact that they involve the accumulation (accumulation) of a common heat carrier for heat transfer systems, and relatively low energy consumption when pumping heat carrier for a small-capacity accumulator.

However, the use of water in heating systems in the cold season poses a number of difficulties, such as the possibility of freezing of water during the dark hours of the day, the risk of corrosion in the system.

It should be noted that in the cold season of the year, if the water in the solar heater is not drained, antifreeze or special liquids that do not freeze to a certain temperature are usually used to prevent the water from freezing. A water-air heat exchanger is used to transfer heat from the accumulator to the building. When the reserve energy source in the accumulator runs out, a conventional boiler supplies the necessary heat.

As a result of the studies conducted in this section, the following conclusions were drawn:

- 1) There are sufficient legal bases for the use of active solar heating systems in the Republic of Uzbekistan for the current period.

- 2) An analysis of scientific and practical work carried out in Uzbekistan and around the world showed that there are many opportunities for scientific research and practical work in our country in the use of active solar heating systems.

- 3) The effectiveness of the use of active solar heating systems in the climatic conditions of Uzbekistan is reflected in the following chapters of the scientific research

work. The use of solar-fueled heating systems is considered the most suitable active solar heating system for the climatic conditions of Uzbekistan.

The location of solar collectors should be determined taking into account the capabilities of the construction site, landscape and climatic conditions, and the location of the building. The calculation of the supporting structures supporting the solar collectors should be carried out taking into account the loads on the support due to wind and snow.

4.5. Combined (active and passive) solar heating systems: advantages and disadvantages

To convert solar energy into thermal energy, a solar collector covered with glass or plastic surfaces, which is used in the “greenhouse effect”, is used. However, it is not always advisable to use only active or passive systems. In order to reduce heat loss and the energy needs of the building, it is advisable to use combined systems that take into account the most beneficial and efficient qualities of passive and active systems.

Scientific research in the field of using Solar Energy (SE) for heating purposes in the world is mainly carried out on the development and study of passive solar heating systems, which differ from active systems, primarily due to their simplicity and low cost. In temperate climates, the most common among passive solar heating systems are insolation systems based on the direct flow of solar radiation (SR) energy into heated rooms through (usually enlarged) light slots in the south wall.

Since the conversion of solar radiation energy into thermal energy in insolation systems occurs after it enters heated buildings, their thermal efficiency is significantly (1.5-2 times) higher than that of traditional active and passive solar heating systems. To increase the efficiency of these systems, three-layer flat transparent coatings are used, which are installed sequentially inside, and the air environment of the room is ventilated, and the inner and middle layers are moistened with barriers.

Panel-radiant heating systems, unlike traditional radiator systems, usually use a heat carrier with a temperature of 35-40°C. In the winter season, the thermal efficiency of the solar panel for heating water to a temperature of 35-40°C is almost twice as high as when heating water to a temperature of 70-80°C. It is clear that the use of panel-radiant systems in solar heating systems can significantly reduce the cost of the solar panel.

Panel-radiant solar heating systems can accumulate sufficient solar radiation to heat buildings during the heating season, which significantly reduces the cost of maintaining the heating system.

The desire to comprehensively address the issues of heating and hot water supply of buildings, the specific characteristics of the equipment used, climatic conditions and a number of other factors leads to the solution of their systems in a combined solution.

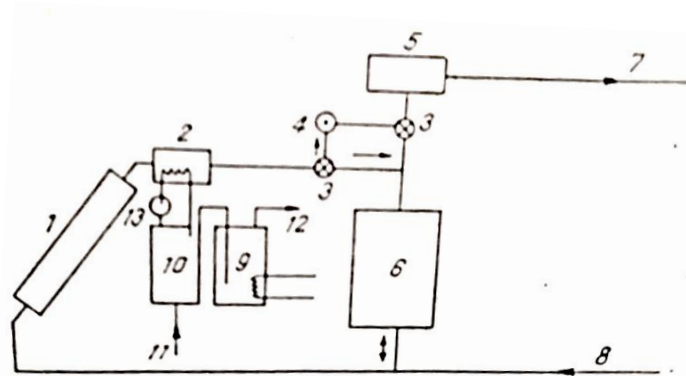


Figure-4.5.1. Schematic diagram of a solar heat supply system with an integrated active air heat carrier: 1-solar air heater; 2-water-air heat exchanger; 3-barrier; 4-fan; 5-additional heat source; 6-gravel heat accumulator; 7-to the room; 8-from the room; 9-water heater; 10 - water heating tank; 11 - cold water; 12-hot water.

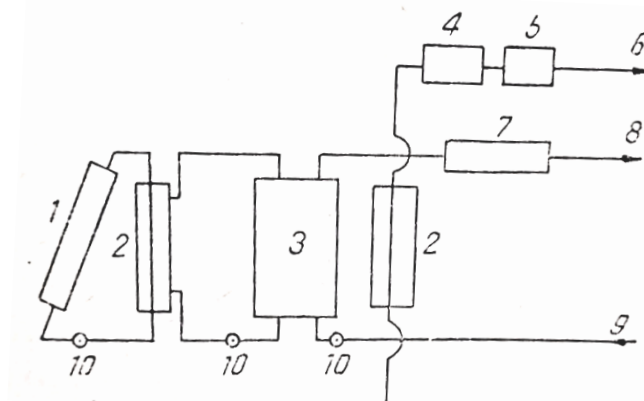


Figure 4.5.2. Principal diagram of a solar heat supply system with a combined active liquid heat carrier: 1-solar water heater; 2-water-water heat exchanger; 3-tank-accumulator; 4-hot water storage tank for hot water supply; 5-additional heat source for the hot water supply system; 6-hot water; 7-additional heat source for the heating system; 8 - inlet to the room; 9 - outlet from the room; 10 - pumps.

Figures 4.5.1 and 4.5.2 show schematic diagrams of integrated active heat supply systems using air and liquid as heat carriers in a solar heating device, respectively. Integrated systems are calculated by combining certain methods of calculating individual subsystems into a common model, and then substituting a general purpose function. In active solar heating systems, the solar collector, which includes a heating element and a heat carrier, is designed to directly absorb sunlight and transfer thermal energy to a storage tank or consumer.

Table 4.5.1 compares the classifications of Low Temperature Solar Heating Systems and their important aspects.

The results of the analysis showed that the use of passive and active solar heating systems (coordinated) with acceptable performance allowed saving up to 60-85% of fuel in TSSs.

Table-4.5.1

Classifications of low temperature Solar thermal supply systems (TSS)

Low Temperature Solar Thermal Supply Systems	Fuel economy coefficient in systems	System achievements	Disadvantages of the system	System advantages	System vulnerabilities
<i>Low Temperature Passive Solar TSS</i>	35-45%	1. There are design solutions for using the proposed system. 2. No large expenditures are required for the construction of buildings and structures	1. The possibility of accumulating and directing excess heat is not always available. 2. Must be planned during the design phase	The system's operating life is the same as the useful life of the building, so there are no unnecessary costs.	The system has a low capacity to provide heat on cloudy, cold days as an additional heat source (backup).
<i>Low Temperature Active Solar TSS</i>	20-35%	Designing solar-fueled heat supply systems leads to an increase in the fuel efficiency of the system, and it is a widespread system with many years of experience in design, construction, and operation.	1. Solar water heating collectors, heat storage tanks and other equipment require a lot of money and space. 2. Engineering solutions for a heat supply system based solely on solar energy are insufficient, since solar insolation during the heating season is insufficient and not always economically justified.	Ease of operation of the system (start-up, shutdown, adjustment), availability of a backup system, heating indicators for objects in the cold season of the year are suitable for radiant panel heating systems (30-35°C)	The system requires constant monitoring during its operation, and optimal operating modes depend on predictions of climate indicators.
<i>Low-temperature hybrid solar TSS</i>	60-80%	It provides guaranteed thermal energy during the heating season, while at the same time it is possible to create an autonomous heating system in the climatic conditions of the Republic. High energy efficiency and economic efficiency	High initial cost of the system	Using modern technologies, it is possible to design it as an autonomous and backup heat supply system, and there are possibilities for automation and control of the system.	Climate indicators are designed based on predictions, and it is difficult to make changes to the system without a qualified specialist.

The architecture of buildings built using solar energy reflects the essence of solar engineering ideas incorporated into the solutions of the structural elements of the building and, accordingly, determines the spatial structure of residential and public buildings based on the functional process implemented in the building. The specifics are determined by the location and design of solar engineering structures. “Solar architecture” causes significant changes in the design, planning and appearance of residential buildings in the field of using solar energy for electricity supply. As a result, a change in the psychological and social attitude towards the natural resources of the Earth, a rational approach to their consumption and protection of the natural environment is emerging.

4.6. Energy-efficient transparent barrier insulative passive solar heating supply system

The thermal protection properties of any transparent barrier, including those designed for passive solar heating systems, are determined by the difference between the total solar radiation entering the heated room and the heat loss through this transparent barrier.

The main purpose of studying Transparent Barriers is to select designs for passive solar heating systems, since the optical and thermal characteristics of the Transparent Barrier largely determine the efficiency of the solar heating systems of the type under consideration.

The combination of the optical and thermal characteristics of the Transparent Barrier can be expressed as their thermal efficiency (η). The expression for determining the instantaneous efficiency, obtained from the equations of the optical and thermal balances of the Transparent Barriers, is as follows:

$$\eta_c = \tau_c - \frac{K_{c_{k-o}}}{q_{nad}}(t_k - t_o), \quad (4.6.1)$$

where, τ - Total solar radiation transmittance of a transparent barrier; $K_{c_{k-o}}$ - Heat transfer coefficient through a transparent barrier; q_{pad} - The total solar radiation flux density falling on the front surface of a transparent barrier; t_k , t_o - respectively, the temperature of the heated room and the ambient (i.e., outside air)..

The results of approximate calculations for expression (4.6.1) show that at $t_k=18^\circ\text{C}$, $t_o=5^\circ\text{C}$ and $q_{pad} = 700 \text{ W/m}^2$ (daytime), and $\tau = 0.8; 0.65, 0.51$ and $K_{c_{k-o}} = 6.35; 3.25$ and $2.0 \text{ W/(m}^2 \cdot ^\circ\text{C)}$, the values of η_s for a single-, double- and triple-layer transparent barrier made of ordinary glass are 0.49 and 0.59; 0.65, respectively. It should be noted that in the winter season of the year, the duration of daylight hours is 7-8 hours, and heat loss through a transparent barrier is observed throughout the day (i.e., 24 hours), and to solve the problem of choosing a transparent barrier structure for a specific insolation passive heating system, it is more convenient to use a comparison of their heat-shielding properties.

The heat-shielding properties of any transparent barrier, including those intended for passive solar heating systems, are determined by the difference between the total amount of sunlight entering the heated room and the heat loss through this transparent barrier. It should be noted that the heat-shielding properties of transparent barriers, all other things being equal, depend on the direction of their location, the climatic conditions of the area and can be determined per unit of time (instantaneous value), per day or for the heating season.

Dusyarov A. S. The work was carried out in relation to the solution of the problem of choosing the type of transparent barrier for passive solar heating systems, determining and comparing their instantaneous heat-shielding properties.

This study was carried out for a transparent barrier made of two and three layers of glass (Fig. 4.6.1).

In solving this problem, the authors set the following boundary conditions that do not cause a change in the physical picture of the thermal processes occurring in the transparent barrier elements under consideration:

- the absorption of transmitted solar radiation by the glass is not taken into account;
- the thermal resistance of the glass is not taken into account;
- the coating surface does not affect the thermal regime of the glass layer;
- the number of transitions does not change over time.

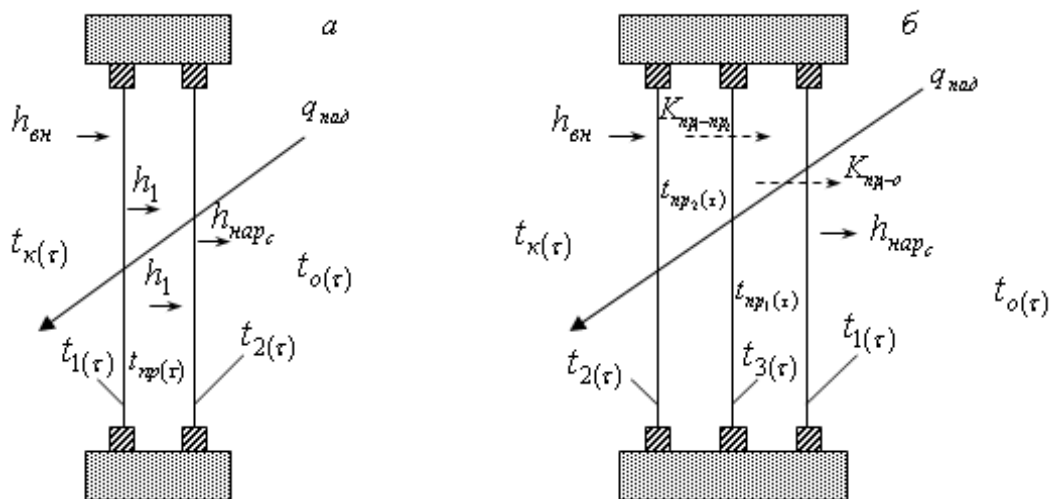


Figure 4.6.1. Schematic diagram of thermal calculation of two (a) and three-layer (b) transparent barriers of insolation passive heating systems.

Q_{tush} - the intensity of the total solar flux falling on the front surface of a transparent barrier; h_{ichki} - the sum (convective and radiant) heat transfer coefficient between the inner surface of the inner layer of the transparent barrier and the environment in the heated room; h_{tashqi} - the sum (convective and radiant) heat transfer coefficient between the outer surface of the outer layer of the transparent barrier and the environment in the heated room; F_1 and F_2 – surfaces of the inner and outer layers of the transparent barrier, respectively; h_1 - The convective heat transfer coefficient between the outer surface of the inner layer of the transparent barrier and the air in the closed interlayer installed between the inner and outer layers of the transparent barrier; t , t_2 , t_k and t_{pr} - the temperature of the inner and outer layers of the transparent barrier, the heated room environment, and the air layer, respectively.

As a result of the formulation and joint solution of equilibrium equations for the inner and outer layers of two- and three-layer transparent barriers, taking into account the non-stationary thermal processes occurring in them, the authors obtained the corresponding solutions (in the form of a function of discrete-continuous values) for determining the daily temperature changes of individual layers of the transparent barrier and the air layer between them. The data obtained in this way served as the basis for comparing the thermal protection properties of two- and three-layer transparent barriers, as well as two-layer transparent barriers coated with a selective film of zinc oxide on the inner surface of the inner layer of the barrier in passive solar heating systems (Fig. 4.6.2).

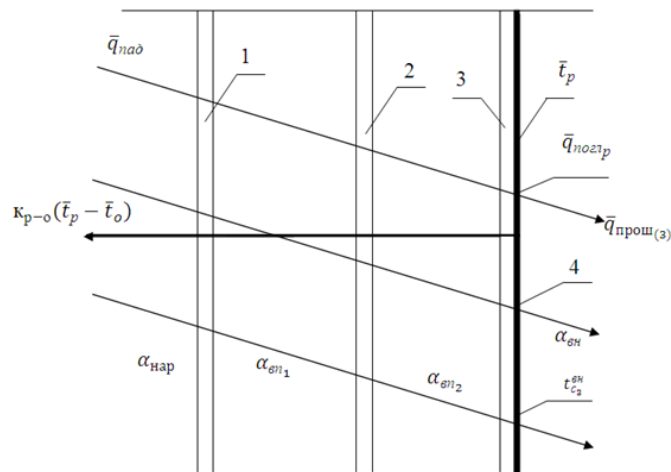


Figure-4.6.2. Calculation-heating principle diagram of the heat balance of the inner surface of the inner layer of a three-layer transparent barrier with a partially light-absorbing film coating. 1, 2 and 3 – respectively, the outer, middle and inner layers of the three-layer transparent wall; 4 – partially light-absorbing variable film coating; a_{vp1} and a_{vp2} – respectively, the closed air layers between the outer and middle and middle and inner layers of the wall.

From the analysis of the graphs in Figure 4.6.3, it can be seen that the three-layer barrier has the greatest thermal insulation properties. As can be seen from the graphs, the thermal insulation properties of a two-layer transparent barrier with a selective film coating are almost equal to half the sum of the thermal insulation properties of two- and three-layer transparent barriers made of ordinary glass.

A comparison of the thermal insulation properties of a single- and two-layer transparent barrier (made of ordinary glass) oriented to the south and east for the climatic conditions of Bracknell (Great Britain) during the heating season is given. As expected, the thermal insulation properties of transparent barriers oriented to the south are much higher than those oriented to the east or west.

Marakaev R.Yu., Permyakov S.N., Fedorov V.K. proposed an approximate method for calculating the possible heating level of the heat-absorbing layer of a two-layer thermal insulation layer located in the outer row. This method is based on solving

the one-dimensional differential equation of stationary heat transfer for the heat-absorbing layer of the thermal insulation layer under the boundary conditions of the third type, for the case when there is an internal heat source in it.

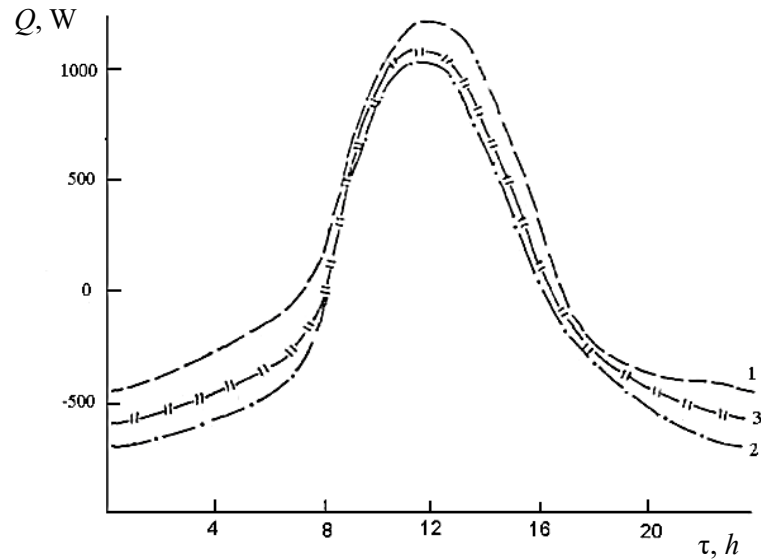


Figure 4.6.3. Comparison of thermal insulation properties of transparent barriers in passive solar heating systems. 1 and 2, respectively, for a two- and three-layer transparent barrier made of ordinary glass; 3 for a two-layer transparent barrier with a selective film coating on the inner surface of the inner layer of glass.

Despite the fact that this method is known to many, when solving the above differential equation, the power of the internal heat source in the considered layer was incorrectly determined. It is known that the specific power of the internal heat source in the considered layer of a transparent barrier, formed due to the partial absorption of the transmitted solar radiation, is determined as follows:

$$q_V = \frac{q_{noz\lambda}}{\delta}, \quad (4.6.2)$$

where

$$q_{noz\lambda} = \alpha_{noz\lambda} q_{nad} \quad (4.6.3)$$

- The surface density of the total (direct and diffuse) solar radiation flux absorbed by the heat-absorbing layer of the barrier; q_{nad} - surface density of the total solar radiation flux incident on the front surface of the heat-absorbing layer;

$$\alpha_{noz\lambda} = 1 - e^{-\beta L} \quad (4.6.4)$$

integral absorption coefficient of the heat-absorbing layer material; β - The extinction coefficient of solar radiation in the visible range of the heat-absorbing layer material; L - the length of the radiation path through the thickness of the heat-absorbing layer.

By substituting (4.6.3) and (4.6.4) into (4.6.2), we obtain the following expression:

$$q_v = \frac{1 - e^{-\beta L}}{\delta} q_{nad} . \quad (4.6.5)$$

In the works of Avezov R.R., Avezova N.R., Abdukhamidov D.U., Kuralov M.A., the value of the term that takes into account the presence of an internal heat source in the heat-absorbing layer when solving the heat equation is determined from the following expression:

$$q_v = \frac{q_{nad} \cdot e^{-\beta L}}{\delta} . \quad (4.6.6)$$

It should be noted that in expression (4.6.6) does not represent the heat power of the internal source, which is formed as a result of the partial absorption of solar radiation passing through the layer, according to $\frac{e^{-\beta L}}{\delta}$ Bouguer's law, but the decrease in the intensity of the transmitted solar radiation with respect to the thickness of this layer under consideration. Therefore, the calculation method for determining the temperature of the heat-absorbing layer of transparent barriers in insolation passive heating systems may lead to incorrect results.

In addition to comparing the thermal protection properties of various passive solar heating systems, it is of great practical interest to study the temperature regime of objects heated by them. One of the ways to solve this problem is to conduct appropriate experimental studies in natural conditions. The main feature of passive solar heating systems is that their efficiency almost entirely depends on the choice of design solutions for their elements and the climatic conditions of the area where the facility is built. Due to the diversity of these two factors, solving the problem under consideration by means of experiments carried out in natural conditions requires significant financial costs and a lot of time. Existing numerical methods make it much easier to solve this problem by calculation.

The solution of the equilibrium equations for the daily change in air temperature inside the heated room $t_{\kappa(\tau)}$ for the climatic conditions of Tashkent city using numerical methods, based on the discrete change in time of the ambient temperature (t_o) and the solar radiation (q) entering the room through a transparent barrier, has the following form:

$$t_{\kappa(i+1)} = t_{\kappa(i)} + 0.265[t_{o(i+1)} - t_{\kappa(i)}] + 0.018q_{(i+1)} , \quad (4.6.7)$$

- after sunset

$$t_{\kappa(i+1)} = t_{\kappa(i)} + 0.24[t_{o(i+1)} - t_{\kappa(i)}] , \quad (4.6.8)$$

The steady-state heat balance equation for the processes of converting solar energy into heat in insolation passive solar heating systems is as follows:

$$Q_{tush_{oyna}}^{\Sigma} = Q_{kayt_{oyna}}^{\Sigma} + Q_{yut_{oyna}}^{\Sigma} + Q_{yut_{pq}}^{\Sigma} + Q_{o'tuv_{oyna}}^{\Sigma}, \quad (4.6.9)$$

In this $Q_{tush_{oyna}}^{\Sigma}$ – the amount of heat energy falling on the window; $Q_{kayt_{oyna}}^{\Sigma}$ – amount of heat returning from the window; $Q_{yut_{oyna}}^{\Sigma}$ – amount of heat absorbed by the glass; $Q_{yut_{pk}}^{\Sigma}$ – The amount of heat absorbed by the film coating; $Q_{o'tuv_{oyna}}^{\Sigma}$ – amount of heat passing through the glass;

The non-stationary heat balance equation for the processes of converting solar energy into heat in insolation passive solar heating supply systems is as follows:

$$(\rho SV)_x \frac{dt_x}{dz} = Q_{o'tgan_{oyna}}^{\Sigma} + Q_{r-x}^r + G_f - Q_{iy}^{devor} - Q_{iy}^{pk} - Q_{iy}^{shift} - Q_{iy}^{eshik}, \quad (4.6.10)$$

In this $(\rho SV)_x \frac{dt_x}{dz}$ – processes of converting solar radiation energy into heat; $Q_{o'tgan_{oyna}}^{\Sigma}$ – amount of heat passing through the glass; Q_{r-x}^r – The energy of the solar radiation stream entering a heated room through a transparent barrier; G_f – The total convective and radiant heat flux falling on the surface of a layer of light-absorbing film installed on the inner layer of a three-layer transparent barrier to the heated room; Q_{iy}^{devor} – heat loss from walls; Q_{iy}^{pk} – Heat loss from the film coating; Q_{iy}^{shift} – Heat loss from the ceiling; Q_{iy}^{eshik} – heat loss through doors.

In this case, these balance equations were developed using methods such as the mean square method, the iterative method, and integration.

From the results of calculations for the five months of the heating season, it can be concluded that in the cold months of the year (December, January, February), the air temperature of the heated object from 10:00 to 16:00 in the morning is equal to or higher than the design temperature (18°C), which allows the traditional heating system to be turned off for six hours. In November and March, the traditional system can be turned off for ten hours (from 8:00 to 17:00 in the evening).

The result of computational studies on determining the specific mass of a short-term phase-transition accumulator based on the previously given conditions.

The total solar radiation flux falling on the surface of the transparent barrier under consideration during the day (8 hours) is 11,000 kJ/m². The average daily thermal efficiency of the transparent barrier is 0.6. The average daily value of the outdoor air temperature is + 5°C, and the normalized air temperature of the heated room is +18°C (according to ShNQ 2.01.04-97 Construction Heat Engineering).

Table 4.6.1 presents the thermal performance of various building materials and structures.

Table-4.6.1.

Thermal and technical characteristics of building materials and structures

S/n	Name	Density kg/m^3	Surface m^2	Melting temperature $^{\circ}C$	latent heat of fusion kJ/kg	Volume m^3	Heat capacity of the mixture, $kJ/kg^{\circ}C$	Thermal conductivity coefficient $W/(m^2^{\circ}C)$	Thickness (mm)
1	Floor		15			45	1,47	0,35	4
2	Transparent barrier		3,75				0.84	0.76	4
3	$(SN_3)_2SO$ dimethyl sulfoxide	1100		20±2	150-250		1,97	1.2	

The heat capacity of a short-term (daily) heat accumulator in insolation-passive heating systems is determined from the ratio of daily changes in the temperature differences at the end and beginning of the thermal charging process of the battery, $Q_{pol}^{spo}(\tau)$ and $Q_{tp}^{op}(\tau)$ i.e.:

$$(MS_r)_{akk} = \frac{\sum_i(Q_{pol}^{spo}\Delta\tau)_i - \sum_i(Q_{tp}^{op}\Delta\tau)_i}{\Delta t_{akk}}. \quad (4.6.11)$$

Specific heat capacity ($q_{foy}^{sht}(\tau)$) of the battery in question (per unit surface area of the transparent barrier):

$$q_{foy}^{sht}(\tau) = \frac{Q_{foy}^{sht}(\tau)}{F_{sht}}, \quad (4.6.12)$$

In this: $Q_{foy}^{sht}(\tau)$ – useful energy of a transparent barrier; F_{sht} – transparent barrier surface;

$$q_{iy}^{ix}(\tau) = \frac{q_o V}{F_{sht}} [t_{vn} - t_o(\tau)], \quad (4.6.13)$$

In this: $q_o V$ – specific heat characteristic of the heated room volume; F_{sht} – transparent barrier surface.

The specific gravity (m_{akk}) of a heat accumulator is determined as follows:

$$m_{akk} = \frac{M_{akk}}{F_{sht}}, \quad (4.6.14)$$

In this: M_{akk} – heat accumulator mass;

The specific heat capacity and mass $(mc_p)_{akk}$ of the excess heat storage accumulator for passive solar heating systems is calculated by the following equation:

$$(mc_p)_{akk} = \frac{\sum_i(\eta_{sht}q_{tush}^{sht}\Delta\tau)_i - \sum_i(q_{iy}^{ix}\Delta\tau)_i}{\Delta t_{akk}}. \quad (4.6.15)$$

In this, η_{sht} – coefficient of efficiency of a transparent barrier; $q_{tush}^{sht}\Delta\tau$ – the specific amount of heat that enters a transparent barrier over a given period of time; $q_{iy}^{ix}\Delta\tau$ – specific heat loss of a heated room over a certain period of time. Since passive solar heating systems are characterized by a lower temperature of the heat carrier, it is possible to use phase-transition materials with a melting point of $\approx 20^\circ\text{C}$ to accumulate excess solar energy during the daytime.

When using low-temperature phase-transition materials to accumulate heat in insolation passive heating systems, the formula for the specific mass of the accumulating material is used, established according to expression (4.6.15):

$$m_{akk} = \frac{\sum_i(\eta_{sht}q_{tush}^{sht}\Delta\tau)_i - \sum_i(q_{iy}^{ix}\Delta\tau)_i}{r_{akk} + (s_r\Delta t)_{akk}}, \quad (4.6.16)$$

In this, r_{akk} – latent heat of phase transition of a heat-storing substance phase-storing substance; $s_{r_{akk}}$ and Δt_{akk} , the specific heat capacity and daily temperature difference of a liquid heat-storing substance, respectively.

Table 4.6.2 presents short-term phase-shifting heat accumulators to improve the operating mode and efficiency of the insolation passive solar heat supply system.

Table-4.6.2

Phase transition materials used as short-term heat accumulators

T/r	Name	Melting temperature, °C	Latent heat of fusion, kJ/kg
1	(SN_3) ₂ SO - dimethyl sulfoxide	20±2	150-250
2	Paraffin 5913	22÷24	190
3	Glauber's salt and table salt mix ($\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O} \cdot 9\text{NaCl}$)	23	

The following phase-transition substances are promising for accumulating low-potential heat in insolation passive heating systems: (SN_3)₂SO -dimethyl sulfoxide (melting point 20±2°C, latent heat of fusion 150-250 kJ/kg), paraffin 5913 (melting point 22÷24°C, latent heat of fusion 190 kJ/kg), as well as a mixture of Glauber's and table salts ($\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O} \cdot 9\text{NaCl}$) with a melting point of 23°C.

We present the results of computational studies to determine the specific mass of a short-term phase-transition battery based on (SN_3)₂SO with a melting temperature of 18.6°C, (latent heat of fusion 173 kJ/kg, specific heat capacity (in liquid state) 1.97 kJ/(m²·°C)) and a density of 1100 kg/m³ for an experimental building with a floor surface of 15m², a volume of 45m³, and a transparent barrier surface of 3.75m² and a specific heat capacity of 1.2 W/(m²·°C)). The total solar radiation flux falling on the surface of the transparent barrier during the day (8 hours) is 11000 kJ/m². The average

daily thermal efficiency of the transparent barrier is 0.6. The average daily value of the outdoor air temperature is $+5^{\circ}\text{C}$, and the normalized air temperature of the heated room is $+18^{\circ}\text{C}$.

The calculated value of the total daily heat loss from a heated room is equal to:

$$Q_{iy}^{ix} = 1,2 \cdot 45(18 - 5) \cdot 8 \cdot 3600 J = 20217,6 \text{ kJ}$$

and the heat generation capacity of the daily transparent barrier:

$$Q_{foy}^{sht} = 0,6 \cdot 11000 \cdot 3,75 \text{ Dj} = 24750 \text{ kJ}.$$

The value of excess solar radiation in the room where energy must be concentrated:

$$Q_{foy}^{sht} - Q_{iy}^{ix} = 24750 - 20217,6 = 4532,4 \text{ kJ}.$$

The mass of the phase-transition heat accumulator used in a passive solar heating system, with an air temperature of 40°C at the outlet and 35°C at the inlet to the liquid accumulator, is as follows for the object under consideration:

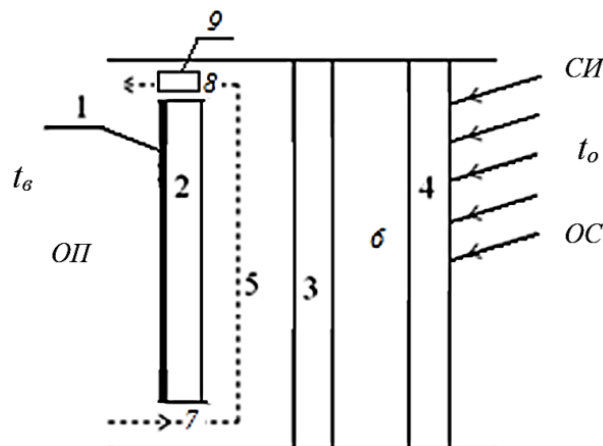
$$M_{akk} = \frac{4532,4}{173 + 1,97(35 - 20)} = 22,4 \text{ kg},$$

where the specific mass of the heat accumulator is $m_{akk} = 22,4 / 3,75 = 6 \text{ kg/m}^2$, which allows it to be placed in the air outlet of the heat exchanger, as shown in figure 4.6.4.

Thus, the temperature regime of a facility with a passive solar heating system for the climatic conditions of Tashkent was solved using numerical methods.

From the results of calculations for five months of the heating season, it can be concluded that in the cold months of the year (December, January, February) the air temperature of the heated facility from 10:00 to 16:00 is equal to or higher than the design temperature (18°C), which allows the traditional heating system to be turned off for six hours.

In November and March, the traditional system can be turned off for ten hours (from 8:00 to 17:00).



ОП-heated room; ОС - environment; СН-solar radiation;

t_g -air temperature of the heated room; t_o - OC temperature; 1-partially light-absorbing thin polymer film coating; 2, 3, and 4, respectively, the inner, middle and outer layers of three-layer semi-insulated thermal insulation; 5 and 6, respectively, ventilated (product) and closed air layers; 7, 8 - ventilation ducts; 9-short-term heat accumulation

Figure-4.6.4. The principle scheme and operation of a three-layer thermal insulation in passive solar heating systems.

4.7. Static model of an active solar heating system without a heat accumulator

The study of the thermal and technical performance of active solar heating systems in buildings was carried out by modeling for stationary and non-stationary cases of an active solar heating system without heat storage. The following figure 4.7.1 shows the principle diagram of an active solar heating system, based on which its mathematical model is presented, taking into account the thermal protection of buildings:

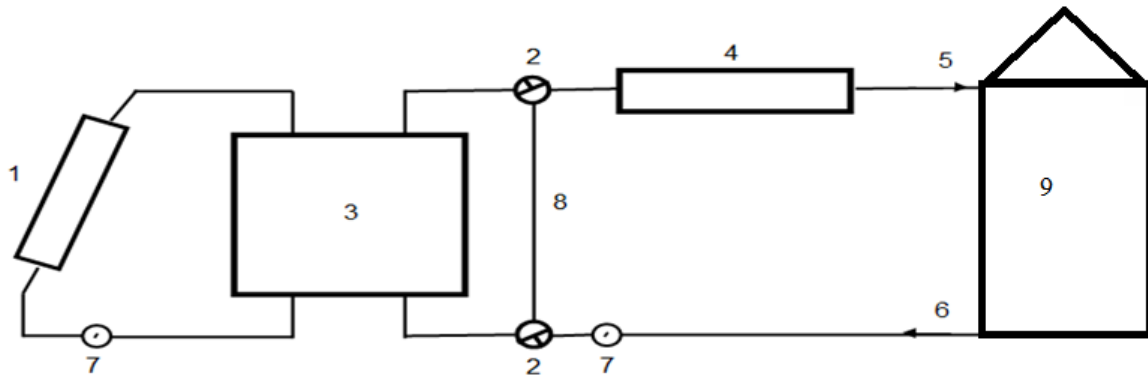


Figure 4.7.1. Schematic diagram of an active solar heating system. 1 – solar water heater; 2 – three-way valve; 3 tank-accumulator; 4 – additional heat source; 5 – entrance to the room; 6 – exit from the room; 7 – pump; 8 – accumulator bypass line; 9 – heat load receiver (Subscriber)

$$K_{m.tr1} = \beta_h \frac{\frac{n_w A_{fw}}{R_{wall1}} + \frac{n_c A_{hcp}}{R_{roofsum1}} + \frac{n_f A_{hcp}}{R_{floor1}} + \frac{n_{win} A_{win}}{R_{win1}}}{A_{c.sum}} \quad (4.7.1)$$

$$K_{m.tr2} = \beta_h \frac{\frac{n_w A_{fw}}{R_{wall2}} + \frac{n_c A_{hcp}}{R_{roofsum2}} + \frac{n_f A_{hcp}}{R_{floor2}} + \frac{n_{win} A_{win}}{R_{win2}}}{A_{c.sum}} \quad (4.7.2)$$

$$K_{m.tr3} = \beta_h \frac{\frac{n_w A_{fw}}{R_{wall3}} + \frac{n_c A_{hcp}}{R_{roofsum3}} + \frac{n_f A_{hcp}}{R_{floor3}} + \frac{n_{win} A_{win}}{R_{win3}}}{A_{c.sum}} \quad (4.7.3)$$

where: $K_{m.tr1}$ – heat transfer coefficient given to the building, $W/(m^2 \cdot ^\circ C)$; β_h - the coefficient is adopted to take into account the additional consumption of the heating system due to the discreteness of the nominal heat flow of the nomenclature of heating devices, additional heat loss through their barriers, increased temperature in corner

rooms, and heat loss from pipes passing through unheated rooms: $\beta_h = 1$; n_w, n_c, n_f - position of individual barriers (walls, ceilings, floors) relative to the outside air, coefficient of reduction of temperature difference for a barrier not in contact with the outside air; $A_{devor}, A_{shift}, A_{pol}, A_{deraza}, A_{kirish\ eshigi}$ - the area of the external surface, respectively walls, ceilings, floors, windows, entrance doors to the building, m^2 ; $R_{devor}, R_{shift}, R_{pol}, R_{deraza}, R_{kirish\ eshigi}$ - resistance to heat transfer, respectively walls, ceilings, floors, windows, entrance doors to the building, m^2 ; A_{bttum} - the total area of all external barriers of the building taken into account in the formula, m^2 .

In the next step, to determine infiltration, the average density of the supply air during the heating period is found from the following formula:

$$\rho_{h.o'z} = \frac{353}{273+0.5(t_{ichki}-t_{tashqi})} \frac{kg}{m^3}, \quad (4.7.4)$$

where: $\rho_{h.o'z}$ - average density of supply air during the heating period, kg/m^3 ; t_{ichki} - calculated indoor air temperature, $^{\circ}C$, t_{tashqi} - outside temperature, $^{\circ}C$. is determined as the average temperature of the heating period for the average conditions of the heating period: $t_{tashqi} = t_{o'rtacha}$ K_{uibk} - overall heat transfer coefficient of the building, $W/(m^2 \cdot ^{\circ}C)$.

$$K_{m.inf} = \frac{C_a n_a \beta_v V_h \rho_{ho'z} k_1}{A_{bttum}}, \quad (4.7.5)$$

where: C_a - heat capacity of air, $J/(kg \cdot ^{\circ}C)$; n_a - average air exchange rate in the building during the heating season, 1/hour; β_v - coefficient of reduction of air volume in a building, $\beta_v = 0.85$; V_h - heated volume of the building, m^3 ; $\rho_{ho'z}$ - average density of supply air during the heating period, kg/m^3 ; k_1 - coefficient for taking into account the effect of counter-heat flow on transparent barriers; A_{bttum} - the total area of all external barriers of the building taken into account in the formula, m^2 ; K_{uibk} - overall heat transfer coefficient of the building, $W/(m^2 \cdot ^{\circ}C)$.

$$K_{m2} = K_{m.tr2} + K_{m.inf} \quad (4.7.6)$$

$$K_{m3} = K_{m.tr3} + K_{m.inf} \quad (4.7.7)$$

The total heat loss of the building during the heating season $Q_{h1,2,3}$ is as follows:

$$Q_{h1} = K_{m1} HDD A_{bttum} \quad (4.7.8)$$

$$Q_{h2} = K_{m2} HDD A_{bttum} \quad (4.7.9)$$

$$Q_{h3} = K_{m3} HDD A_{bttum} \quad (4.7.10)$$

where: $K_{m1,2,3} - K_{uibk}$ - total heat transfer coefficient of the building according to thermal protection levels (I, II, III), $W/(m^2 \cdot ^{\circ}C)$; HDD - Calculated indicators of

heating season degree days for the city of Tashkent, °C; A_{bttum} – the total area of all external barriers of the building taken into account in the formula, m².

When calculating the thermal energy obtained from solar collectors, the thermal energy entering the active solar heating system is determined using the following formula:

$$Q_{qite} = 0.44H_o A_{qkuyu} \quad (4.7.11)$$

In this, H_o – The flux of solar radiation outside the atmosphere, kW·s/m²; A_{qkuyu} – Total surface area of solar collectors, m².

Figure 4.7.2 below illustrates the change in the specific heat consumption of a building with different levels of thermal protection.

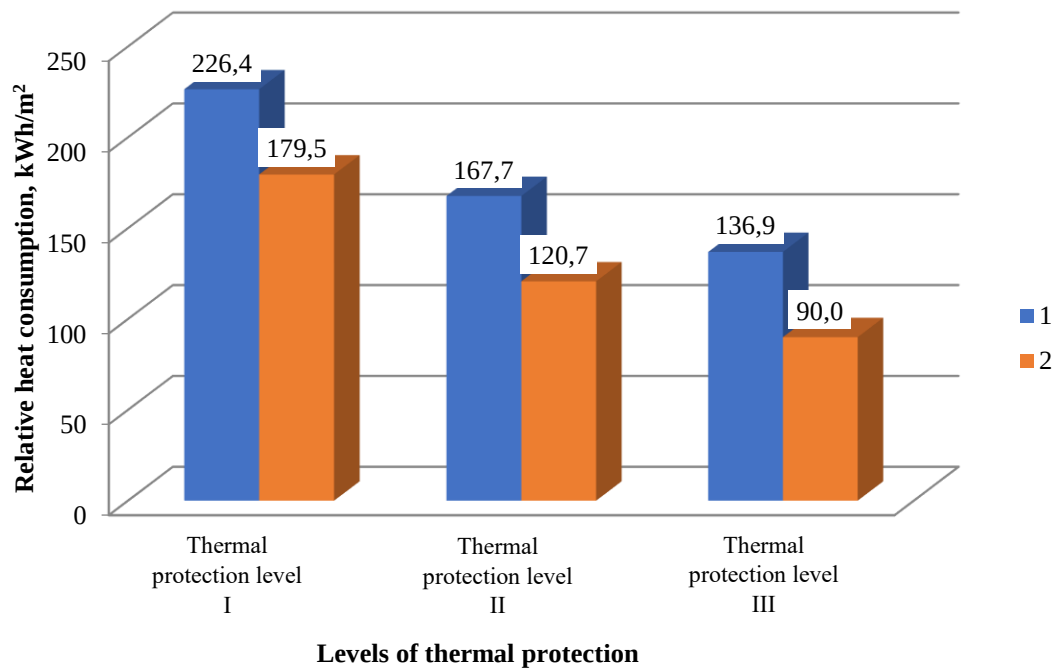


Figure 4.7.2. Variation of comparative heat consumption at different levels of thermal protection: 1- for the case where an active solar heating system is not used; 2- for the case where an active solar heating system without a tank is used.

Table-4.7.1

Comparison of computational and experimental results

Degrees of thermal protection	Annual specific heat consumption of the building,, kW·h/m ²	
	Experimental	Calculated
Thermal protection level I	218.9	226.4
Thermal protection level II	151.8	167.6
Thermal protection level III	123.4	136.9

The calculation results show that for the city of Tashkent, when using an active solar heating system without a tank, the specific heat consumption can be reduced by 179.5 kWh/m² (20.7%) at the first level of thermal protection, 120.7 kWh/m² (28%) at

the second level of thermal protection, and 90.0 kWh/m² (34.3%) at the third level of thermal protection.

4.8. Dynamic model of an active solar heating system with a heat accumulator

To determine the dynamic thermal properties of a building, we use the degree-hour method. In this case, the total heat loss of the building is determined as follows:

$$Q_{uiy} = K_m HDH A_{uyu}, \quad (4.8.1)$$

In this, HDH – is the number of degree-hours of the heating period, and is determined as follows:

$$HDH(t_{ix}) = \sum_{i=1}^N (t_{ix} - \bar{t}_{tx})^+, \quad (4.8.2)$$

In this, t_{ix} - required temperature inside the building; $\bar{t}_{tx}$ - hourly average ambient temperature; N - the number of hours in the period under consideration (month, heating season or year).

The geometric dimensions of the heated building are calculated as follows:

$$A_{as} = \frac{\pi D_{to.ich}^2}{4} \quad (4.8.3)$$

$$A_{yon} = \pi D_{to.ich} H_{to} \quad (4.8.4)$$

$$V_{tp} = A_{as} H_{to} \quad (4.8.5)$$

$$U_{as} = \left(\frac{\delta_{to}}{\lambda_{to}} + \frac{\delta_{to.ich}}{\lambda_{to.ich}} + \frac{1}{H_{tash}} \right)^{-1} \quad (4.8.6)$$

$$U_{yon.l} = \left[\frac{1}{2\lambda_{to}} \ln \left(\frac{D_{to.ich} + \delta_{to}}{D_{to.ich}} \right) + \frac{1}{2\lambda_{to.ich}} \ln \left[\frac{D_{to.tash}}{D_{to.ich} + \delta_{to}} \right] + \frac{1}{h_{nar} D_{to.tash}} \right]^{-1} \quad (4.8.7)$$

$$UA = U_{as} A_{as} + U_{yon.l} + H_{to} \quad (4.8.8)$$

The calculated heat flux density q_{hioz} , W/m² is calculated using the following formula:

$$q_{hioz} = q_{nom} \left(\frac{\Delta t_{sr}}{70} \right)^{l+nl} \left(\frac{G_{s.s} \frac{hr}{kg}}{360} \right)^{pl}, \quad (4.8.9)$$

where: q_{nom} – nominal heat flux density, W/m²; $\left(\frac{\Delta t_{sr}}{70} \right)^{l+nl}$ – temperature difference when the heat carrier is water, °C; $G_{s.s}$ – water consumption, kg/h; n, p – experimental numerical values.

For the selected surface A_{ia} , the temperature of the water leaving the heating devices t_{chiq} , °C is as follows:

$$t_{chiq} = t_{hh} + \frac{1}{\left[\frac{1}{(t_{kir} - t_{hh})^{nl}} + \frac{q_{pr} nl \left(\frac{G_{pr} hr}{0.1 kg} \right)^{N_{seksiya Apr}}}{70^{1+nl} G_{pr} C_{suv}} \right]^{\frac{1}{nl}}} \quad (4.8.10)$$

where: t_{kir} – temperature of the heat carrier entering the heating device. °C; t_{hh} – calculated air temperature in the heated room. °C; q_{pr} – q_{nom} - nominal heat flux density, W/m²; n, p – experimental numerical indicators that represent the influence of the structural and hydraulic properties of a heating device on the heat transfer coefficient; $G_{s.s}$ – actual water consumption in the heating device, kg/h; C_{suv} – specific heat capacity of water J/(kg °C).

Table 4.8.1 presents the geometric dimensions of the heat accumulating and storing tank accumulator for the system under consideration, which serve as the basis for determining the values of formula (4.8.5).

Table-4.8.1

Geometric dimensions of the heat storage tank accumulator

Naming	Outer diameter, mm	Inner diameter, mm	Length mm	Insulation mm	Weight kg	Volume m ³
Tank-accumulator	2 506	2 300	5 300	200	3 800	22

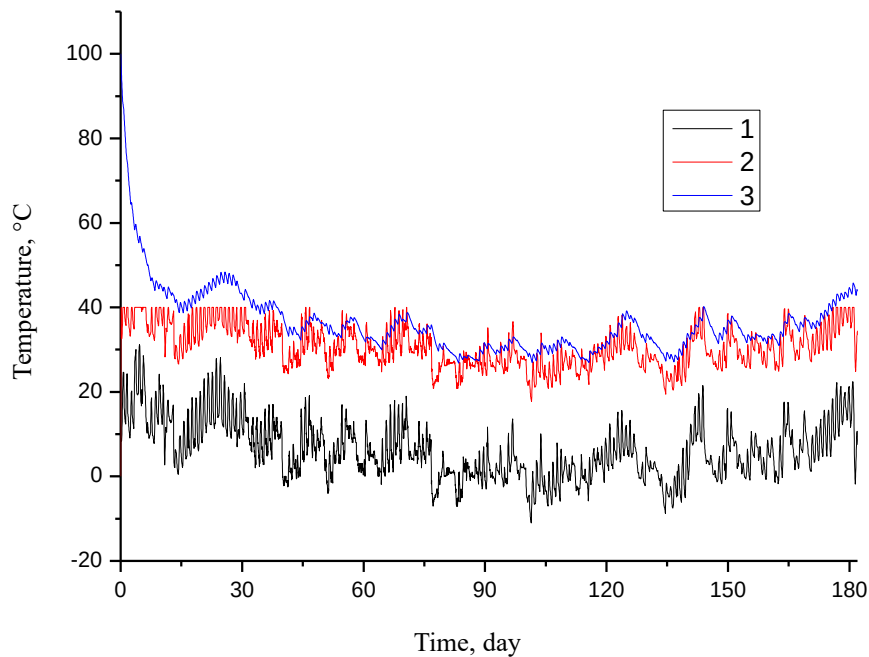


Figure 4.8.1. 15.10.2019 - 15.04.2020. Temperature changes over time: 1-ambient temperature; 2-temperature of the heat carrier leaving the solar collector; 3-temperature of the water in the tank-accumulator.

Table-4.8.2

Thermal and technical characteristics of the protective layer of the heat-storage tank accumulator

	Density kg/m ³	Relative heat capacity, kJ/kg ⁰ C	Thermal conductivity coefficient W/m ² ⁰ C	Thickness mm
Galvanized steel sheet	7800	0,45	55	0.5
Mineral wool	125	0.84	0.056	200
Paint	1100		0.233	0.1
Steel sheet	8000	0.468	50	4

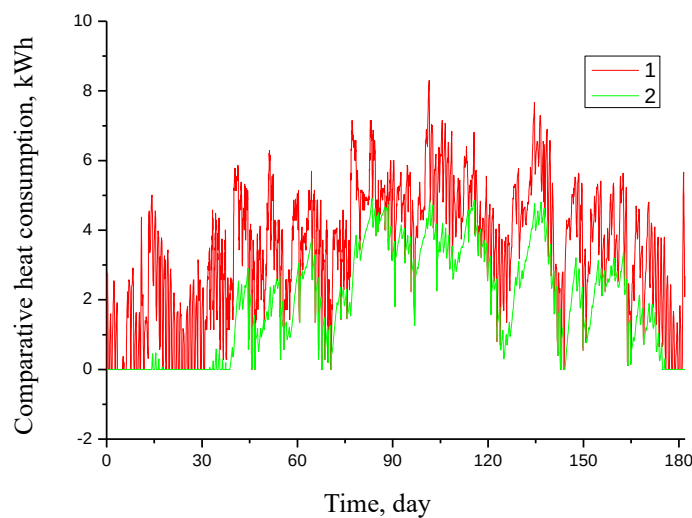


Figure 4.8.2. 15.10.2019 - 15.04.2020. Hourly change in the heat load of a conventional heating system: 1- without an active solar heating system; 2- with an active solar heating system.

Based on the values presented in Figures 4.8.1 and 4.8.2, and Table 4.8.2, the dynamics of the changes in temperatures inside the building over time and the hourly changes in the heat load of the traditional heating system when using an active solar heating system with a heat accumulator are presented, and Figure 4.8.3 analyzes the active solar heating system with and without a tank accumulator according to the levels of thermal protection.

Figure 4.8.3 shows the change in the specific annual heat energy consumption for heating a building by the levels of thermal protection. Calculations conducted for a building with an active solar heating system with a tank-accumulator show that for the city of Tashkent, it is possible to reduce the specific heat consumption by 78.81 kW·h/m² (33.4%) at the first level of thermal protection, by 69.17 kW·h/m² (39.79%) at the second level of thermal protection, and by 63.6 kW·h/m² (44.7%) at the third level of thermal protection.

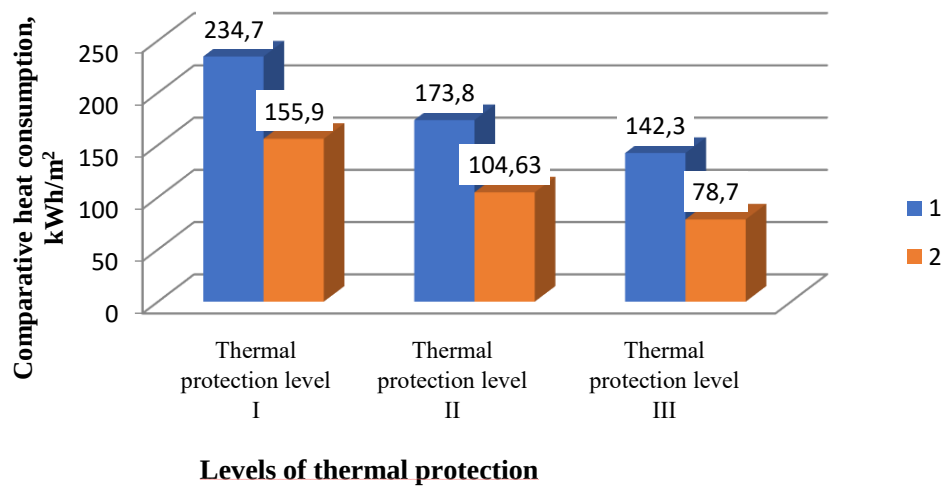
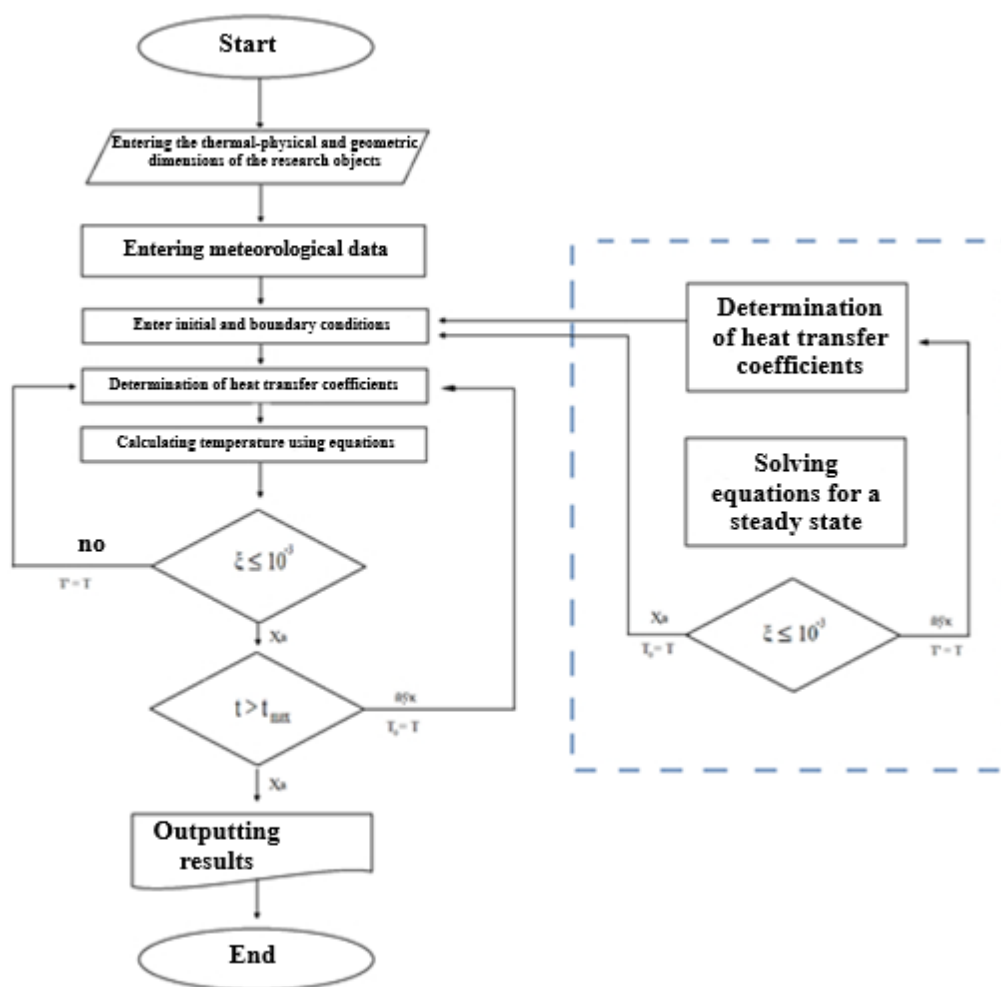


Figure 4.8.3. Variation of comparative heat consumption by levels of thermal protection. 1- for the case where an active solar heating system is not used; 2- for the case where an active solar heating system with a tank-accumulator is used.



T' – estimated temperatures; T_0 – initial temperatures

Figure 4.8.4. Block diagram of the algorithm of the mathematical model developed for an active solar heating system.

For the numerical implementation of the proposed model (Figure 4.8.4), an algorithm and software were developed in the MathCAD 2015 professional package.

With the help of this program, the specific thermal properties, energy, environmental and economic indicators of administrative and residential buildings were determined using the degree-hour and degree-day methods. In particular, the specific energy consumption for heating in administrative and residential buildings, the amount of energy saved when using solar collectors and heat accumulators, the annual amount of reduced emissions of toxic gases into the atmosphere, the annual material savings, the hourly average values of specific heat indicators, and the payback period were calculated.

The software product is built in the Python programming language based on static and dynamic mathematical models of thermal processes occurring in administrative and residential buildings, and the average hourly and annual energy, environmental and economic indicators of administrative and residential buildings are calculated based on climate data.

4.9. Determination of heat capacity and selection of materials for short-term accumulation of daily excess heat from solar radiation in insolation passive heating systems

The main tasks of heat accumulation in passive heating systems are to determine the heat capacity of the short-term heat accumulation device of the heating system and to select the appropriate material.

The analysis of the research results shows that even small amounts of heat accumulation in solar heating systems can significantly increase the fuel exchange coefficient of these systems. At the same time, increasing the specified capacity of the heat accumulator (HA) to $400 \div 600 \text{ kJ}/(\text{m}^2 \cdot ^\circ\text{C})$ per unit surface of the solar heater does not give a significant change in the fuel exchange coefficient. The alternative capacity of solar heating systems for residential buildings in regions with temperate climatic conditions is $300 \text{ kJ}/(\text{m}^2 \cdot ^\circ\text{C})$. When the air temperature in the heated room is $18\text{-}20^\circ\text{C}$ and the fuel conversion coefficient of the heating system is below $0.30\text{-}0.35$, the use of heat accumulators in solar heating systems during the midday period, which lasts only $1.5\text{-}2$ hours in the heated room, is not practical.

For passive solar heating systems, the heat capacity of the short-term heat accumulator can be determined by looking at the daily graphs of the useful solar radiation heat energy entering the heated room through the three-layer transparent barrier in the system (Q_{pol}^{sht}) and the environment (Q_{iyu}^{op}). Naturally, if $Q_{pol}^{sht}(\tau) > Q_{iyu}^{op}(\tau)$, then there is a need to accumulate excess solar heat during the time interval $\Delta\tau = \tau_2 - \tau_1$ of the daylight part of the day (Figure 4.9.1).

where: $t_{x_n} = 18^\circ\text{C}$ - normalized air temperature in heated buildings; $t_o(\tau)$ - daily change in outdoor air temperature.

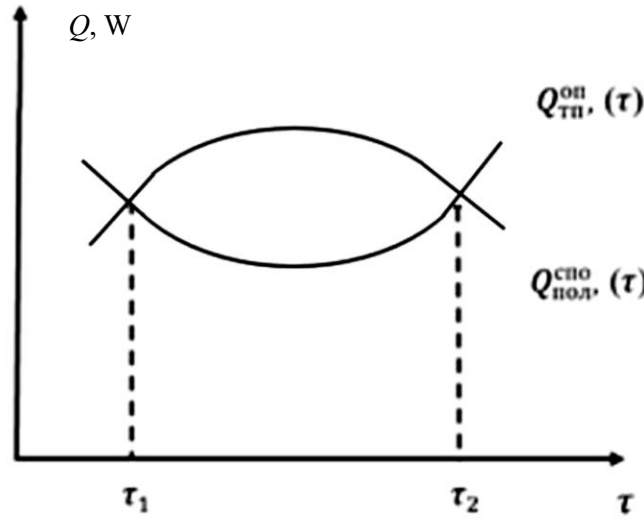


Figure 4.9.1. Daily variation of $Q_{pol}^{sht}(\tau)$ and $Q_{iyu}^{op}(\tau)$ on a typical clear and warm day.

The value of the daily change of Q_{pol}^{sht} depends on the surface area (F_{sht}) of the Transparent Barrier of the heated building, as well as its design. In particular, it depends on the ventilated air layer (i.e., produx) between its inner and middle layers, and on the surface of its inner layer and the inner layer, which is considered the most effective for a three-layer Transparent Barrier with a light-absorbing laminate film between the partially light-absorbing layer (Fig. 4.6.4). The value of $Q_{pol}^{sht}(\tau)$ can be determined from expression (4.9.1).

The value of the daily change of Q_{iyu}^{op} is determined from the following expression, depending on the specific heat capacity (q_o) and the volume (V) of the heated building in terms of external dimensions:

$$Q_{iyu}^{op}(\tau) = q_o V [t_{x_n} - t_o(\tau)], \quad (4.9.1)$$

$$Q_{pol}^{sht}(\tau) = Q_{o'tuvchi}^{sht}(\tau) + Q_{np}^{\Sigma}(\tau) + Q_p^{gr}(\tau), \quad (4.9.2)$$

where:

$$Q_{o'tuvchi}^{sht}(\tau) = \alpha_{eff}^{op} \tau_{eff}^{spo}(\tau) F_{sht} q_{tush}^{sht}(\tau); \quad (4.9.3)$$

- Daily variation of the total solar radiation flux entering the heated room through a three-layer energy-efficient transparent barrier (ETB),

$$Q_{np}^{\Sigma}(\tau) = \alpha_{np}^{\Sigma}(\tau) F_{sht} [t_{np}(\tau) - t_x(\tau)], \quad (4.9.4)$$

- The daily variation of the total (convective and radiant) heat flux emitted from the inner surface of the light-absorbing laminate film of the translucent barrier and received by the heated room;

$$Q_p^\Sigma(\tau) = G_v(\tau) S_{rv} [t_{x_{tash}}(\tau) - t_{x_{ichki}}(\tau)], \quad (4.9.5)$$

- Daily variation of heat flow entering the heated building from the inner and middle layers of the Transparent Barrier based on natural gravity;

α_{eff}^{op} – The ability of a heated room to effectively absorb the total solar radiation,

τ_{eff}^{spo} – The total effective solar radiation transmission capacity of the three-layer Transparent Barrier;

q_{pad}^{spo} – The total solar radiation flux incident on the front surface of the transparent barrier is the surface density.; α_{lp}^Σ – The combined (convective and radiant) heat transfer coefficient between the inner surface of a partially absorbing laminate film and the interior of the building;

$t_{lp}(\tau)$ – Daily temperature variation of the partially light-absorbing layer in contact with the inner layer of the transparent barrier; $t_v(\tau)$ – air temperature in the heated room; $G_v(\tau)$ – Daily air flow through a transparent barrier product;

S_{rv} – specific heat capacity of air; $t_{v_{vx}}(\tau)$ and $t_{v_{vbx}}(\tau)$ respectively, daily changes in air flow temperature at the entrance and exit of the Transparent Barrier product.

The daily variation of the total (optical and thermal) efficiency of the Transparent Barrier under consideration is determined from a certain expression:

$$\eta_{spo}(\tau) = \frac{q_{pol}^{spo}(\tau)}{q_{pad}^{spo}(\tau)}, \quad (4.9.6)$$

and expressions (4.9.2) and (4.9.3) can be expressed in the following form:

$$\eta_{spo}(\tau) = \alpha_{eff}^{op} \tau_{eff}^{spo}(\tau) + \frac{\alpha_{lp}^\Sigma}{q_{pad}^{spo}(\tau)} [t_{lp}(\tau) - t_v(\tau)] + \frac{m_v(\tau) S_{rv}}{q_{pad}^{spo}(\tau)} [t_{v_{vbx}}(\tau) - t_{v_{vx}}(\tau)], \quad (4.9.7)$$

bu yerda:

$$m_v(\tau) = \frac{G_v(\tau)}{F_{spo}}, \quad (4.9.8)$$

- Daily change in air flow rate (per unit of the front surface of the air handling unit) passing through the product.

CHAPTER V. HEAT ACCUMULATION SYSTEMS

5.1. Heat storage in solar systems

The need for heat storage in solar systems is based on the daily variation of solar energy flux throughout the year. The energy reserve in the accumulator is calculated for a day or several hours in short-term accumulation processes, and for several months in seasonal accumulation. In general, the use of heat accumulators increases the efficiency of solar systems and the reliability of heat supply. Low-temperature heat storage systems cover the temperature range from 30 to 100°C and are used in air (30°C) and water heating systems (30–90°C), as well as in hot water supply (45–60°C). As a rule, the heat storage system consists of:

- Reservoir;
- Heat storage material for storing and harvesting thermal energy;
- A heat exchanger device for supplying and transferring heat during battery charging and discharging;
- Thermal insulation.

Depending on the nature of the physicochemical processes that occur in the heat-accumulating material (HAM), batteries can be divided into the following classes:

- Capacitive type accumulators, i.e., they use the heat capacity of the accumulating material that is heated (cooled) without changing its aggregate state (natural stone, shell, water, solutions of salts in water, etc.);
- Batteries consisting of substances with a spatial transition state, in which the heat of fusion (solidification) of the substance is used;
- Energy storage devices based on the absorption and release of heat in reversible chemical and photochemical reactions.

In the first group of accumulators, the processes of cooling or heating the heat-accumulating material through a heat exchanger using solar energy occur simultaneously or sequentially. This method of heat accumulation is widespread.

The disadvantage of this type of accumulators is that they have a large mass, which requires a large area, and for 1 gJ of accumulated heat, an area of the same size as a building is also required.

A comparative table of various heat-accumulating materials is given.

Table-5.1.1

Comparison of some heat-accumulating materials

HAM characteristics	Granite, slate	Water	Glauber's salt		Paraffin
Density capacity	1600	1000	1460 ^j	1330 ^j	786 ^t
Heat	0,84	4,2	1,92 ^t	3,26 ^j	2,89 ^t
Thermal conductivity coefficient	0,45	0,6	1,85 ^t	1,714 ^j	0,498 ^t

Mass of HAM for accumulating 1 TJ of heat at $\Delta T = 20$ K, kg	59500	11900	3300	3750
Relative mass of HAM to water mass, KG/KB	5	1	0,28	0,32
Volume of HAM for 1 TJ heat accumulation at $\Delta T = 20$ K, m ³	49,6 ^j	11,9	2,26	4,77
Relative volume of HAM to water mass m ³ /m ³		1	0,19	0,4

Note: 1. The levels are defined as follows: t – solid state; j – liquid state; – assuming a void volume of 25%.

2. Heat of fusion and temperature: paraffin – 47°C and 209 kJ/kg; Glauber's salt – 32°C and 251 kJ/kg.

5.2 Features of the heat storage system

Temperature range

Temperature is a crucial characteristic of thermal energy storage systems. The temperature at which a thermal energy storage system is charged and discharged usually determines its application area. The different temperature ranges for thermal energy storage systems are as follows. High-temperature devices require temperatures between 300 and 1400 °C. In the renewable energy sector, solar thermal power plants are key applications in this range. Large storage systems using molten salt at temperatures up to 560 °C are charged during the day and then discharged at night to ensure continuous electricity generation, even at night when there is no sunlight. Thermal storage systems can also be combined with compressed air energy storage (CAES) systems, with advanced adiabatic compressed air energy storage (AA-CAES) systems increasing their efficiency from about 50% to 70% of that of CAES, at maximum temperatures of about 600 °C. Examples of industrial process heat applications that exist at temperatures above 1000 °C include solutions in the steel industry, glass industry, and industrial air purification systems.

Process heat at temperatures between 100 and 250 °C is mainly used to generate steam for industrial processes. These processes include, for example, pasteurization and sterilization. Steam accumulator technology (also called “pressure water storage” or “Ruths”) is the most modern technology for heat storage and is used in process heat applications. “Variable storage” or “thermocline” storage systems can also be used for pressurized water. Hot and cold water are combined in a single vessel to provide pressurized water at up to 180 °C (about 30 bar). Thermal storage temperatures for building heating are from 25 °C for surface heating systems (underfloor heating) to 90

°C for surface radiators. For hygienic reasons, a minimum temperature of 60 °C is required for hot water supply in households and businesses. Cold storage in the refrigeration and air conditioning area usually requires a temperature of 5 to 18 °C. Food is usually cooled to a temperature of 5 to 8 °C. The air conditioning temperature is 6 to 18 °C. Freezing food is an additional area and requires a temperature below -18 °C.

Heat transfer fluid (HTF)

Thermal energy storage systems vary widely not only in terms of temperature, but also in terms of the heat carrier, also known as the heat transfer fluid. Heat transfer fluids can be of a single-phase type (e.g. water, air, thermal oil, molten salt) or of a two-phase type with condensation and evaporation processes in the system (e.g. water/steam). The storage material and heat transfer fluid can be the same (e.g. water). In the more simple direct storage concept, an additional volume of hot working fluid is used to store the heat. Many working fluids cannot be stored directly; the energy must be transferred to a separate storage medium. Here, two different media (as heat transfer fluid and storage medium) are used. This concept is often referred to as the indirect storage concept.

Indirect storage concepts require sufficient heat transfer rates and minimizing temperature gradients from the heat transfer fluid to the storage medium. Separate heat transfer fluid and storage medium may be used for the following reasons:

1. The heat transfer fluid is expensive (e.g., synthetic thermal oil).
2. The heat transfer fluid has a low volumetric energy density (e.g., gas, steam).
3. The heat transfer fluid is pressurized (e.g., steam, water, thermal oil), and the larger storage volume is at (or close to) ambient pressure for economic reasons.
4. The heat transfer fluid is incompatible with the storage medium.

For indirect storage concepts, the storage medium and the heat transfer fluid may be in direct contact (e.g., air as the heat transfer medium in the steel industry and ceramics as the storage medium).

In some cases, direct contact between the heat transfer fluid and the storage material is not economical or practical, for example, if the heat transfer fluid is pressurized (e.g., steam) or incompatible with the storage medium.

In this case, a design with indirect contact between the carrier and the storage medium can be used. Indirect communication can be achieved by an external heat exchanger or a heat exchanger built into the storage volume.

Heat losses

Important aspects of heat losses include the temperature of the thermal energy storage at the nominal temperature, thermal insulation, surface-to-volume ratio (or

size), environmental conditions (e.g. temperature, wind speed), number of charge/discharge cycles per year, and operating time. Typically, the thermal insulation aspect is economically optimized in relation to heat losses. Thermal energy storage systems can have low heat losses. For example, molten salt tanks with efficiencies of up to 99% have been built.

Number of tanks

The number of tanks or storage units is another classification criterion. The single tank concept is characterized by different zones with charged and discharged storage materials. An example is a smart thermal storage system with hot and cold zones (e.g. a water tank). Alternatively, storage systems can use two tanks. For example, a smart thermal storage system with molten salt consists of two separate tanks with different temperatures and liquid levels. The two-tank concept allows for the separation of thermal power (storage capacity of the two tanks) and thermal power (additional external heat exchangers).

Long/Short-term storage

As with all energy storage technologies, a key characteristic of thermal energy storage systems is the time interval between charging and discharging. Short-term storage is used for storage lasting from a few minutes to a few days. Long-term storage stores energy for a period of weeks to a year. It is important to note that short-term storage systems typically have higher charging and discharging capacities than long-term storage systems, but they store less energy.

Centralized/decentralized storage

The location in the power system is also an important criterion for describing an energy storage system. Storage systems are classified as centralized or decentralized. There are no formal or precise definitions of these terms, so they can only be defined by examples. Large solar thermal power plants have large molten salt storage facilities located within the power plant. Such storage sites can be called centralized storage sites. Buffer storage for heating in a single-family home or latent heat ice storage for air conditioning in an office building are typical examples of decentralized storage facilities. The closer the storage is to the consumer, the more decentralized it is.

Stationary/mobile storage

Stationary energy storage systems are devices that are installed in a specific location. They are different from mobile storage systems. Since energy density is the decisive factor here, these are usually latent heat storage or thermochemical storage

systems. The energy is stored in containers at a charging point, for example, if the energy cannot be used there. The stored energy is then delivered to the consumer.

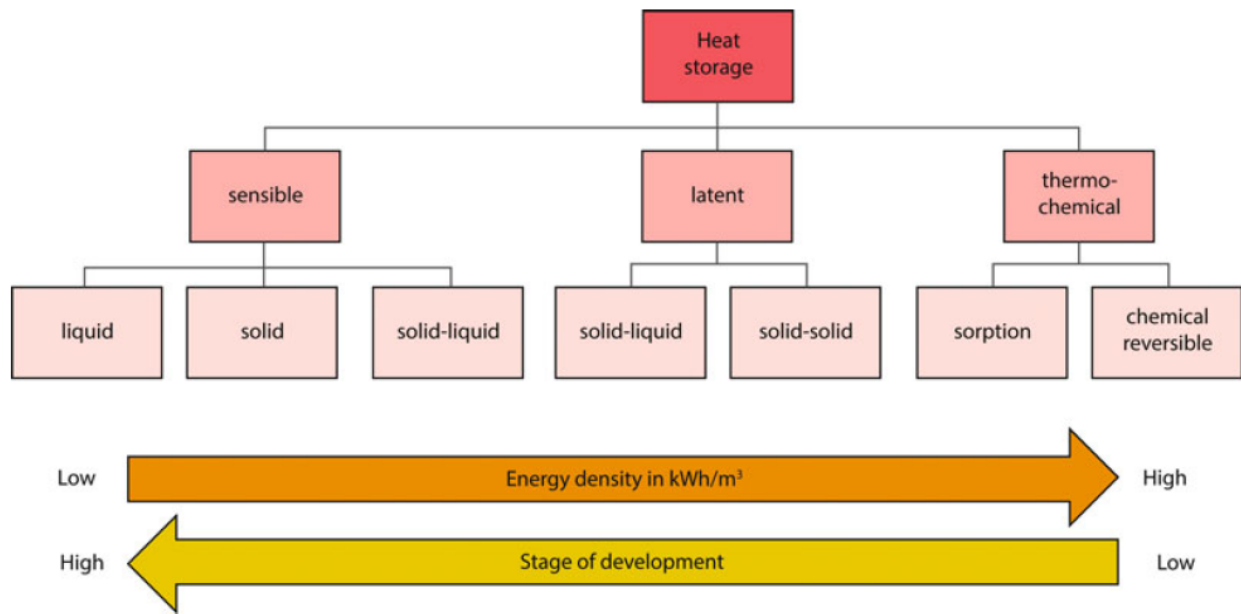


Figure-5.2.1. Overview of various thermal energy storage technologies.

5.3 Storage technologies

Thermal energy can be stored in various ways. This section explains the terms sensible, latent, and thermochemical thermal storage (Figure 5.2.1).

Sensible-heat storage

The rational storage of thermal energy requires a significant change in temperature. The storage medium is heated or cooled. The amount of energy stored is determined by the specific heat capacity (cp-value) of the material. Since the temperature differences between the storage medium and the environment are often larger in rational energy storage systems than in other types of storage, the thermal insulation used is very important. Since water has a high specific heat capacity, as well as other advantages such as environmental friendliness, ease of use and low cost, many rational thermal storage systems use water. Typical applications for such storage are in heating systems for buffer storage. However, rational thermal storage systems are also used for medium and long-term (seasonal) storage. If water is used as the heat carrier here, it can be modified to provide protection against frost by adding alcohol. If necessary, it can be replaced with another material. For example, solid materials such as gravel or iron oxide rock can be used in sensible heat storage. Compared to latent or thermochemical heat storage systems, sensible-heat storage facilities have a lower energy density if the limited temperature range is taken into account. However, sensible-heat storage technology is standardized and significantly lower in cost.

Latent-heat storage

Latent heat storage systems store the energy required for a phase change along with sensible heat. In practice, most technologies use a solid-to-liquid phase change. This phase change typically results in a volume change of about 10%. This must be taken into account when designing the system. The enthalpy of transition or melting is an important factor. The phase change from liquid to gas results in a large volume change that is difficult to control. As a result, it is rarely used. Theoretically, there are three types of this type of storage system. The first option is isochoric storage. Here, the pressure increases with temperature and, consequently, with the evaporation temperature. The second option is isobaric storage. Since this requires a large condensation chamber, this option is not used in practice. The third option is storage in an open system at ambient pressure using a liquid-to-gas phase change. This type of storage is essentially a type of chemical storage, and is therefore discussed in the adsorption section. Latent storage has a number of advantages over sensible storage. Significantly more thermal energy can be stored at smaller temperature differences than in conventional storage. This means that these systems have a higher energy density. In addition, the temperature remains constant for a long time during charging and discharging. As a result, phase change materials (PCMs) that are incorporated into the building structure with a melting temperature of 20 to 25 °C can be used (for example). This keeps the room temperature within a comfortable range, as it takes longer to make the room too cold or too hot. Due to their high storage capacities, the design of latent-heat storage structures can be significantly more compact. The melting temperatures of the various types of materials used range from -40 °C to 1000 °C. An interesting option that is particularly useful for transporting thermal energy are slurries, also known as phase change slurries (PCS). The advantage of these is that they can always be pumped, regardless of their state of aggregation. This is achieved by adding the phase change material to a carrier fluid with a low solidification temperature. At the same time, only a fraction of the thermal power of the phase change material is used, so that a sufficient liquid phase remains. Slags are used to build air conditioners and are also used in industrial facilities. Here, ice solutions can be used.

Thermochemical-heat storage

The term thermochemical energy storage refers to chemically reversible reactions where the reaction products can be separated and stored for long periods of time. This separation eliminates storage losses. During charging, an exothermic reaction releases the stored energy. Thermochemical energy storage systems allow for very high energy storage densities, but are rarely used in practice. This is because the technology is still in the research and development phase. Sorption storage facilities

are also often classified as thermochemical storage systems. Currently, research on sorption storage is focused on the use of industrial waste. This technology is particularly well suited for industrial drying processes. The absorption and adsorption processes are more physical than chemical. However, since this is common practice, sorption storage systems are classified as thermochemical energy storage systems in this chapter.

Thermophysical properties and requirements for materials

Heat storage materials for sensible, latent and thermochemical heat storage systems must meet the following requirements:

1. Large gravimetric storage capacity (high heat capacity, latent heat or heat of reaction);
2. Large volumetric storage capacity (high density and gravimetric values listed above);
3. Long service life, non-toxic, non-flammable, no explosive phases, simple in handling (e.g. hygroscopic);
4. Non-corrosive in terms of heating, heat exchanger and heat transfer, reinforcing structures (e.g. fins), use of inexpensive structural materials;
5. Ability to perform multiple cycles of charging/discharging without loss in performance and storage capacity (high cycling and thermal stability);
6. Reasonable material costs, high availability;
7. High heat dissipation and thermal effusivity values (high heat transfer rate);
8. Small density variation with temperature (e.g., to minimize thermomechanical stress);

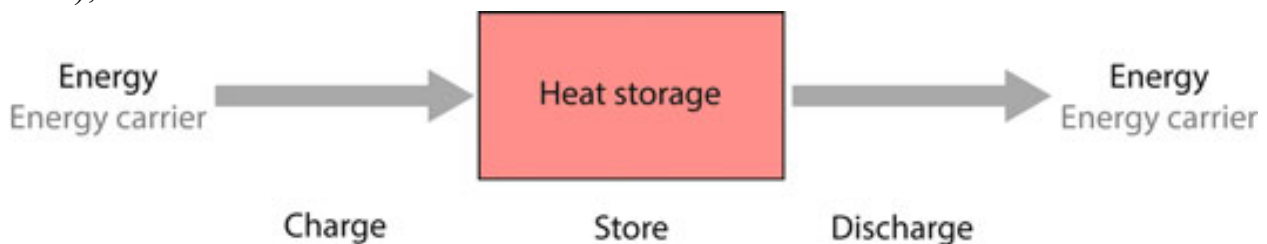


Figure-5.3.1. Energy storage processes.

5.4. Fundamentals of thermodynamics

Thermal energy

Thermal energy storage systems increase the energy content (charge) by adding energy or an energy carrier. They then store this energy content for a certain period of time with as little loss as possible (storage) and release the energy or energy carrier when needed (recharge). Release reduces the energy content of the storage system (Figure-5.3.1).

Thermal energy - heat and cold - is not a state variable that describes the state of the system, but only a process variable that is used during changes in state or processes (Figure-5.4.1). This is especially true for the storage capacity (or energy storage density), that is, the thermal energy stored relative to the mass or volume of the storage object. This means that the storage space does not contain a certain amount of heat or cold, but rather it contains the thermodynamic potential to transfer that amount of heat or cold to another medium (Figure-5.4.2).

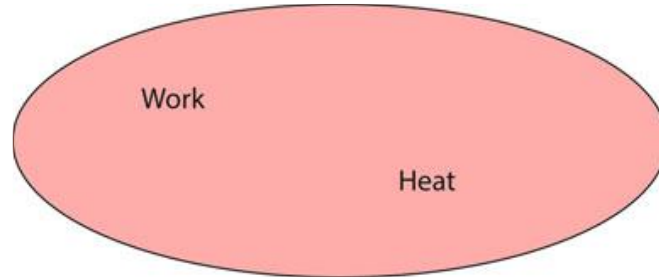


Figure-5.4.1. Process variables.

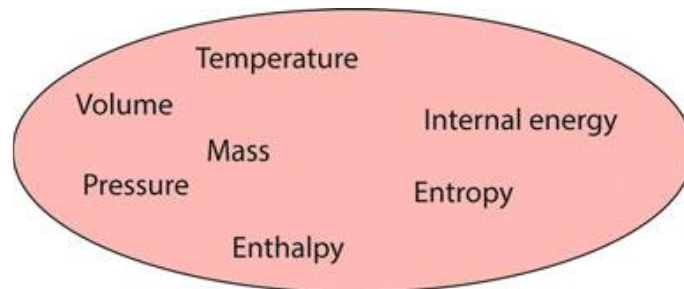


Figure-5.4.2. State variables.

As a result, the term “energy consumption” is physically incorrect. This is because energy is not consumed; it is simply converted into another form. Similarly, the term “energy storage” is not entirely correct, because although it can no longer be used, the energy is certainly unchanged. The term exergy is then used to describe the part of the energy that can be converted into other forms of energy in the environment. The greater the difference between the storage temperature and the ambient temperature, the greater the amount of exergy and the importance of heat. The term energy is used here to describe the part of the energy that is released to the environment and is no longer usable. If the charging, storage, and discharging processes of a thermal energy storage system result in losses, the amount of energy increases and the amount of energy decreases. However, the sum of the energy and exergy contents remains the same. Equation 5.4.1 shows how the exergy content is calculated:

$$E_{exergy} = Q \cdot \frac{T_S - T_U}{T_S} = Q \cdot \left(1 - \frac{T_U}{T_S}\right) \quad (5.4.1)$$

This shows that the exergy content Q is determined by the heat quantity, the storage temperature T_S , and the ambient temperature T_U . This fraction is equal to the Carnot efficiency limit, which assumes the maximum useful work available in the heat.

For simplicity, in the following discussion, the terms energy storage, heat quantity, etc. will be referred to by their common dictionary definitions.

Heat transfer

There are three types of heat transfer: convection, thermal radiation, and thermal conduction. Convection refers to the movement of particles carrying thermal energy in a flowing fluid. As a result, it is also called heat flow. For example, gravity heating systems transfer heat from a heat storage tank to radiators through circulating water, and therefore use convection. This type of convection is also called natural or free convection. With forced convection, the particles are transported by mechanically driving the fluid using pumps or fans (such as circulating pumps in hot water heating). Convection also occurs on the outside of radiators. The air near the radiators is heated and expands. It then rises and is replaced by cold air from below. The fluid can be controlled by temperature, density, concentration, or pressure differences within the system.

Thermal radiation is electromagnetic radiation emitted by materials due to their temperature. Thermal radiation, which stores thermal energy, is in the infrared range. At higher temperatures, the maximum radiation of thermal radiation shifts to shorter wavelengths. Thermal radiation does not depend on the material. According to the Stefan-Boltzmann equation, it is determined by the temperature T , the surface area A , the emissivity of the radiating body, and the Stefan-Boltzmann constant $\sigma=5.67\times 10^{-8}$ $\text{W}\cdot\text{m}^{-2}\text{K}^{-4}$:

$$\dot{Q} = \frac{\partial Q}{\partial t} = \varepsilon \sigma A T^4 \quad (5.4.2)$$

Since there is (more) heat radiation from a hot environment to a cold environment and (less) heat radiation from a cold environment to a hot environment, the temperature difference is lower. Enameling, anodizing, or coating metal objects such as radiators can increase the emission of infrared radiation and, consequently, the heat radiation. Thermal conductivity refers to the transfer of kinetic energy between adjacent atoms or molecules without any material transport in solids or liquids at rest. Heat flow occurs due to a temperature difference, i.e. heat flows from a hot environment to a cold environment. The heat flow through an object is primarily determined by the temperature difference between the hot surface (T_2) and the cold surface (T_1), the surface area A , the thickness d of the body, and the thermal conductivity λ . Thermal conductivity depends on the material and temperature and is given in $\text{W}/(\text{mK})$. Radiators are made of metals with high thermal conductivity, allowing the maximum amount of heat to be transferred from the warm water to the air in the room.

$$\dot{Q} = \frac{\lambda}{d} A (T_2 - T_1) \quad (5.4.3)$$

Thermal insulation

With sensible heat storage, thermal insulation is essential to reduce heat losses. Latent and thermochemical-thermal storage systems also require insulation, but it plays a less important role. This is because the temperature differences between the thermal storage and the environment are smaller or insulation is only required during charging and discharging processes. Equation 10.4 shows the relationship between temperature difference and heat losses in a storage system:

$$Q_{losses} = U \cdot A \cdot (T_S - T_U) \quad (5.4.4)$$

The main factors for thermal insulation are the surface area A , the storage temperature T_S , the ambient temperature T_U and the overall heat transfer coefficient U . This last coefficient determines the Kelvin temperature difference transmitted through one square meter of area, and is given in units of $W/(m^2K)$. For n layers it is defined as:

$$U = \frac{1}{\frac{1}{\alpha_i} + \frac{s_1}{\lambda_1} + \frac{s_2}{\lambda_2} + \dots + \frac{s_n}{\lambda_n} + \frac{1}{\alpha_a}} \quad (5.4.5)$$

Equation (5.4.5) shows that the heat transfer coefficient is determined by the thermal conductivity λ , the material layer thickness s_i , and the heat transfer coefficient α . This is a proportionality factor that describes the intensity of heat transfer at the boundary. The heat transfer coefficient is not a material constant. Instead, it is determined by the flow rate, the type of fluid, geometric relationships, and surface properties.

Table-5.4.1

Thermal conductivity of various materials at room temperature

Material	Issiqlik o'tkazuvchanligi λ W/(mK)
Water	0.562
Air (78% nitrogen, 21% oxygen)	0.026
Vacuum	0
Glass	0.76
Concrete	2.1
Iron	80
Copper	400
Aluminum	235
Diamond	1 000-2 500
Silver	430
Polyurethane (PUR)	0.021-0.035
Wool/cork	around 0.035

Tables 5.4.1 and 5.4.2 give material constants for a number of materials. A high degree of insulation can be achieved by replacing the insulating material with a vacuum (as in a double-walled thermos). At pressures below 0.001 mbar, thermal losses are almost exclusively due to thermal radiation.

Vacuum insulation is often used where the design cannot accommodate thick insulation.

Vacuum insulation, powder or fibrous insulation materials can also be used to reduce thermal radiation. This type of super-vacuum insulation reduces thermal conductivity to levels ranging from 0.01 to 0.003 W/mK. This means that their insulating capacity is five to ten times higher than that of conventional insulation materials.

Table-5.4.2

Thermal conductivity coefficients of various materials at room temperature

Material	Thickness, cm	Heat transfer coefficient U, W/(m ² K)
Concrete	25	3.3
Aerated concrete	40	0.16-0.21
Brick	24	1.5
Brick + composite thermal system (PUR)	30	0.3
Solid wood	20.5	0.5
Single-glazed window	-	5.9
Double-glazed window	-	3
Passive house standard window	-	0.5-0.8
Polystyrene	2	1.7
Polystyrene	10	0.35

5.5. Sensible heat storage system

Sensible heat is the absorption or release of heat that causes a significant change in temperature. The storage medium used is heated or cooled. The relationship between the amount of heat Q and the temperature is given by the following equation:

$$Q = m \cdot c_p \cdot (T_2 - T_1) \quad (5.5.1)$$

The amount of heat is determined by the temperature difference, the mass m , and the specific heat capacity c_p . These physical properties determine the amount of heat that must be added to one kilogram of mass to raise its temperature by one Kelvin. Table 5.5.1 shows the values of c_p for a number of storage materials. With thermal storage, a high specific heat capacity is preferable. This is because less material is then required to store a given amount of energy, and the thermal storage device is more

compact in design. Compared to latent or thermochemical storage systems, sensible-heat storage objects have a lower energy density. The energy density is limited by the specific heat capacity and the maximum permissible temperature difference.

The temperature difference is determined by the application boundary conditions or the temperature range of the phase change of the medium (for unpressurized water, from 0 to 100 °C). The advantage of liquid sensible storage media is the transfer and implementation of high charging and discharging capacities. The thermal capacity is determined by the pump capacity and in some cases the heat exchanger capacity. However, very high temperatures can be stored with solid storage media. Compared to the other two heat storage technologies, sensible heat storage systems have relatively low capital costs. They are also the most advanced and installed systems on the market (e.g., non-pressurized and pressurized water storage, thermal mass in buildings, regenerators in the process industry, and molten salt storage for concentrated solar power plants).

Storage materials

The choice of storage material is determined by:

- operating temperature of the system;
- suitability of the material for heat transfer;
- suitability of the material for transportation and storage;
- availability and cost of the material;
- environmental friendliness of the material.

The characteristics of materials used in sensible heat storage systems are given in Table 5.5.2. In addition to the specific heat capacity, the volumetric heat capacity is often also given for building applications. This is because weight is less important here than the required spatial volume. The volumetric heat capacity of non-porous solids is typically between 1.5 and 6 MJ/(m³ K). The most commonly used medium in sensible heat storage systems is water. Water has the following properties:

- available in large quantities almost everywhere;
- cheap;
- easy to transport;
- environmentally friendly, as it is non-toxic, non-aggressive, non-flammable and non-combustible;
- easy to work with (engineers have extensive experience in working with water, and workers are familiar with the technology);
- high specific heat capacity (4.219–4.185 kJ/(kg K));
- low thermal conductivity (0.562–0.560 W/(m K));
- prevents heat exchange between hot and cold water;

- stratification: warm water can be drawn directly from a higher storage location without requiring additional heating;
- different densities at different temperatures;
- stratification is possible;
- relatively low viscosity ($1.792\text{--}1.003 \times 10^{-6} \text{ m}^2/\text{s}$);
- an excellent heat carrier that provides high charging and discharging capabilities;
- excellent solvent properties;
- water acts as a heat carrier and heat storage medium for domestic water heating, as well as a product necessary for consumption.

The main application of water as a storage material is in buffer storage and heating systems. These systems balance short-term (few hours/day) differences between the heat produced and consumed. Other areas of water application are medium-term (weeks/months) and seasonal thermal storage (summer/winter) at temperatures between 0 and 100 °C. Seasonal storage is often used to store solar heat and heat buildings. In combination with district heating, these storage systems often provide enough solar heat to cover 50% of the total heat required for building heating and domestic hot water.

Table-5.5.1

Temperature ranges, heat capacities, densities, and applications for various materials

Middle	Temperature range	Specific heat capacity	Volumetric heat capacity	Density	Usage
Unity	°C	kJ/(kgK)	kJ/(m ³ K)	kg/m ³	
Water	0-100	4.19	4,175	998	Domestic water, heating appliances
Sand, gravel, stone	0-800	0.71	1,278-1,420	1,800-2,000	Keeping the soil warm
Granite	0-800	0.75	2,062	2,750	
Concrete	0-500	0.88	1,672-2,074	1,900-2,300	Thermally active components
Brick	0-1,000	0.84	1,176-1,596	1,400-1,900	
Iron	0-800	0.47	3,655	7,860	Storage materials for solar energy storage systems
Thermal oil	0-400	1.6-2.1	1,360-1,620	850-900	
Gravel-water filling	0-100	1.32	2,904	2,200	Soil thermal storage
Dissolved salts	150-450	1.3	1,725-1,970	2,243-2,561	Concentrated solar thermal power plant
Sodium	100-800	1.3	750-925	975-1,203	

Since the vapor pressure increases strongly at temperatures above 100 °C, this also increases the pressure requirements for the container. This in turn increases the cost of the storage system. As a result, thermal oil is usually used at temperatures above 100 °C. However, the thermal conductivity of thermal oil is significantly lower (about 0.1 W/(mK)) and has a lower heat capacity than water. By combining thermal oil with a high heat capacity rock filling, the storage capacity can be increased. This also reduces the amount of relatively expensive thermal oil required. Another alternative is molten salts. Molten nitrate salts can be used at temperatures up to 560 °C. At very high temperatures, solid storage systems made of ceramic (regenerator-type storage) or iron oxide rock (storage heaters) are used. Often, heat and cold are stored underground for long periods. The following types of storage are commonly used:

- aquifer storage facilities;
- cave storage facilities;
- geothermal probe storage facilities;
- geothermal collector storage facilities;
- gravel/water or soil/water storage facilities.

Various designs for a reasonable energy storage system are discussed in the following sections. For the purposes of this discussion, storage systems are classified according to the general nature of the storage medium. First, storage systems using solid storage media are discussed. This is followed by storage systems using liquid storage media. The discussion concludes with a description of storage systems using solid and liquid storage media.

Storage systems using solid storage media

This section describes six types of storage systems that have essentially similar designs and functions. In the case of geothermal probes, geothermal ground collectors, and foundation tanks, the solid storage medium (soil and concrete) is indirectly charged and discharged by a heat transfer fluid. Night storage heating also uses a solid storage medium (iron oxide rock), but here the heat is transferred by an electric heating element.

Geothermal-probe storage systems

Geothermal probe storage systems use soil or rock as the storage medium. Geothermal probes are usually used with water-saturated clay rocks, as these have a high heat capacity and low conductivity. If the groundwater flow rate is high, the heat is quickly removed. The heat is transferred through a U-tube or coaxial heat transfer pipe (often made of polyethylene) inserted 20-150 m into a vertical well. Drilling with a diameter of 10 to 15 cm and a depth of less than 100 m requires mining permits in Germany. Since polyethylene is a relatively poor thermal conductor (0.4 W/(mK)), this limits the thermal conductivity. The thermal conductivity can be increased by filling

the drill pipe with a bentonite-cement-sand mixture. A water-antifreeze mixture is used as the heat carrier, flowing through the pipe at a speed of about 0.5 to 1 m/s. This creates a turbulent flow, increasing heat transfer. Heat transfer is usually used to indirectly charge and discharge the system.

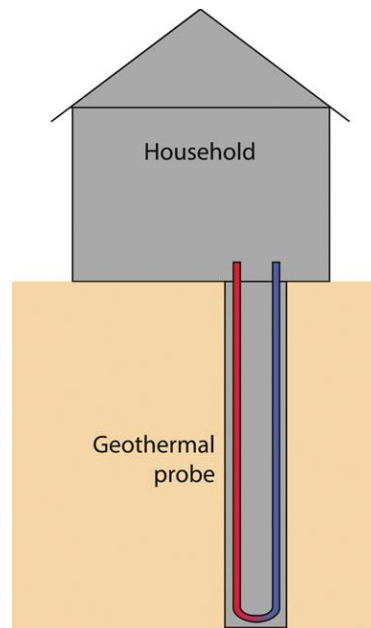


Figure 5.5.1. Geothermal probe.

By connecting several geothermal probes in a probe field, the soil can be used as a storage mass. The distance between the probes can be from 1.5 to 12 m. In general, there are no structures built on top of geothermal probe fields. Their storage volume can be more than 50,000 m³. The storage temperature is usually between 40 and 80 °C. Geothermal-probe storage systems can be used as both heat and cold storage facilities by installing a reverse heat pump. They are most effective in low temperature ranges and at low flow temperatures (e.g. 35 °C for underfloor heating) (see Figure 5.5.1).

Geothermal ground-collector storage systems

Geothermal ground collectors are installed near the surface in loose soils at a depth of 1 to 2 m. As a result, they are subject to seasonal temperature fluctuations. As in geothermal probes, geothermal ground collectors use the soil as a storage medium. The heat transfer fluid (a water-antifreeze mixture) flows through horizontal plastic pipes with a diameter of about 25 mm. The pipes are connected to a heat pump, so that geothermal energy can be used for heating. Excess heat is transferred from the building to the ground for cooling. The length of individual pipe sections is less than 100 m to limit the pumping capacity. Geothermal ground collectors require a large area for installation. Geothermal ground collectors for residential buildings usually require an area equal to twice the living area of the heated building. In addition, structures cannot be built on these objects, the coating on them cannot be very thick, and they have

limited applications in areas where deep roots penetrate the ground. Another disadvantage is that laying the pipe requires expensive excavation.

Table-5.5.2

Various types of ground extraction capacities using geothermal ground collectors

Soil properties	Heat output, W/m ²
Dry, sandy	10-15
Wet, qumli	15-20
Dry, muddy	20-25
Wet, muddy	25-30
Groundwater	30-35

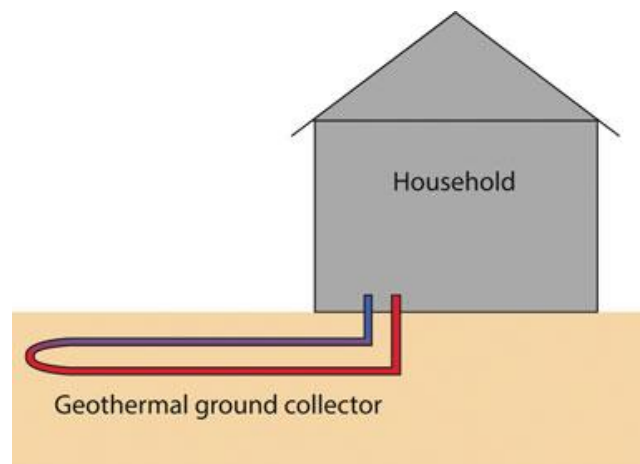


Figure-5.5.2. Geothermal ground collector.

Table 5.5.2 shows the heat output of different types of soil. The table shows that more heat can be extracted from moist soil types than from dry soil types. Since geothermal ground collectors are usually not in direct contact with groundwater, they can be built in Germany and other European countries without special permits (Figure-5.5.2).

Basic storage

The design of a foundation storage system is similar to that of a geothermal probe system. Here, the heat carrier flows through plastic pipes connected to the reinforcement of the building's foundation. The foundation structure acts as a heat exchanger. Being a thermally active component, it also acts as a storage mass together with the surrounding soil. This means that the system is indirectly charged and discharged. The storage system is usually insulated on all sides and is often used in conjunction with solar thermal devices. Here, the foundation storage system stores excess solar energy in the summer and releases it in the winter using a heat pump. The plastic pipes are usually connected to the reinforcement components with wire connectors. The pipe sections are individually controlled, so the system can continue to operate even if one component fails. Unlike geothermal probes or geothermal

ground-collector systems, structures can be built on foundation storage systems (Figure 5.5.3). Depending on the foundation, the system design may differ in details such as:

- energy piles: pile foundation used as a foundation structure (Figure 5.5.4);
- foundation absorber: base plate used as a foundation structure;
- diaphragm wall used as a foundation structure.

Theoretically, the system can provide heating supported by a heat pump, as well as cooling supported by a chiller. The temperature in the soil is in the range of 4-20 °C. Low temperatures should be avoided to prevent cracking of adjacent concrete structures.

Thermally active components

Thermally active components can be used to store thermal energy in the foundation of a building, as well as in the building itself. This use of thermally active components is often referred to as thermal component activation or concrete core activation.

Concrete parts of the building, such as ceilings, walls or floors, act as the storage mass here. The components are charged and discharged by water flowing through coiled pipes embedded in the concrete core. Underfloor heating is the most common form of this system.

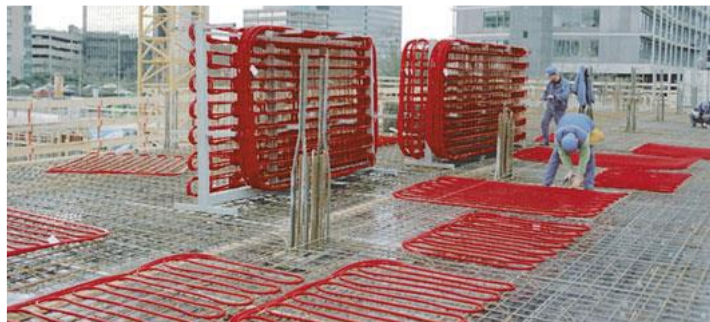


Figure 5.5.3. Basic storage system.

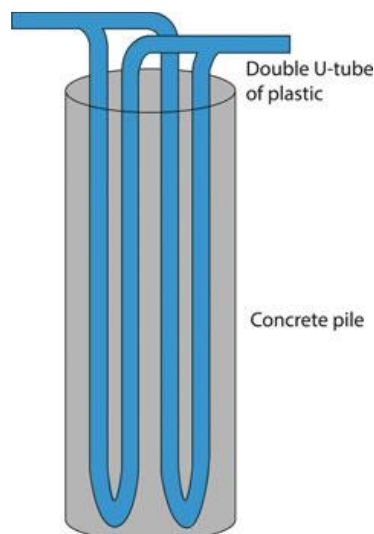


Figure 5.5.4. Energy package.

The temperature of the components is usually maintained between 18 and 35 °C. At temperatures below 18 °C, it may no longer be comfortable for people inside the building and condensation may form on the activated components. Large buildings can store energy for a long time. The carriers then react to temperature changes and keep the room temperature relatively stable. Thermally active components can be used for both heating and cooling purposes (see Figure 5.5.5).



Figure 5.5.5. Thermally active components.

Storage heaters

Storage heaters are also a form of energy storage in a sensible-heat storage system (Figure 5.5.6). Unlike all other sensible energy storage systems discussed here, storage heaters do not use a heat transfer medium for charging. Instead, heat is stored directly by converting electrical energy into heat using a heating element stored in an iron oxide matrix. Storage heaters absorb electrical energy, convert it into heat, and then store it in their core. The heater then releases the heat into the room over a long period of time. In the past, energy supply companies offered cheaper electricity at night to supply night-time heating units. In the future, the charging process may be related to other factors, such as the availability of excess renewable energy from wind power. Theoretically, any temperature range can be stored by converting electrical energy into heat. In practice, temperatures significantly higher than those required to heat a room (up to 600 °C) are maintained.

Storage heaters can be designed in such a way that the heat release cannot be controlled. With this design, the storage core is completely covered with weak thermal insulation. The heat is released by convection and radiation. However, in almost all storage heaters available in Germany, the heat release is automatically controlled. The heat release is controlled by a fan. The cold air drawn in by the fan flows upwards and then spreads out. Part of the air passes through the storage shaft and is heated as a result. This partial flow can exit the storage core at very high temperatures. As a result, the hot air is mixed with another partial flow that passes directly to the outlet through a rotary damper without being heated.



Figure-5.5.6. Storage heater section.

The most commonly used storage media are iron oxide and magnesite (also known by its brand name Feolite). The storage media must have a high storage capacity and also the ability to quickly release the stored heat when needed, i.e. the thermal conductivity of the medium must be as high as possible. Magnesite is a ceramic material made of iron oxide and magnesium. Its specific heat capacity is determined by temperature. At room temperature, its specific heat capacity is around $0.245 \text{ W}/(\text{kg K})$ and at 600°C it is $0.34 \text{ W}/(\text{kg K})$. Another advantage of magnesite is that its thermal conductivity is determined by temperature. At 200°C , the thermal conductivity is around $9.3 \text{ W}/(\text{mK})$ and at 600°C it is around $5.8 \text{ W}/(\text{mK})$. This doubles the thermal conductivity at low temperatures, allowing for high heat output even at low temperatures.

High temperature regenerator type storage

This type of storage avoids the problem of high vapor pressure of the liquid medium and allows for maximum storage temperatures of up to 1600°C . The maximum application temperature depends on the specific material.

Ceramics are commonly used. Furthermore, despite the relatively low specific heat, these typically inorganic solids can still achieve high storage densities where operating conditions exhibit large temperature differences. This is often the case with applications that use gases as heat transfer media in liquids.

Regenerators, also called regenerative heat exchangers, include a storage medium that is heated and cooled by intermittent heat exchange between hot and cold fluids. The storage operates in counter-flow: during charging, hot fluid enters the storage module at the “hot” end, and during discharging, the flow direction is reversed and cold fluid enters the “cold” end. Current industry examples include solutions in the steel industry, the glass industry, and industrial air purification systems.

The steel industry was one of the earliest applications. Even today, hot blast furnaces or Cowper furnaces, which preheat the blast furnace of a steelmaking furnace,

are used commercially. Two or three furnaces are used in alternating operation to provide heat for a long period of time (Figure-5.5.7).

Hexagonal sintered refractory bricks are a common inventory choice. The temperature at the top of the furnace can be as high as 1300-1600 °C. Heat rates are in the range of 100-300 MW. Regenerative thermal oxidizer (RTO) systems are used to clean air of volatile organic compounds by an exothermic process that occurs in a regenerator.

This process oxidizes organic solvents at elevated temperatures in the range of 800-1000 °C. Internal heat exchange between the hot purified gas and the incoming cold gas is achieved by intermittently storing heat in the ceramic inventory of the regenerator, which allows for the operation of a nearly autothermal plant with heat recovery efficiencies of up to 95% and higher. Commercial applications of a regenerative thermal oxidizer typically have volumes of up to 600 m³.



Figure 5.5.7. Regenerator (or Cowper) for storing sensible heat in solids.

In recent years, there has been increasing interest in regenerative storage for various power plants, including combined cycle power plants, concentrated solar power plants, and advanced adiabatic compressed air energy storage plants for utility-scale electricity storage. In this context, the technology has been further developed to improve efficiency and reduce costs. In particular, regenerative storage based on packed bed inventory has been considered. Although it is less common today, it can further reduce investment costs and avoid the scalability problems of previous designs.

Storage systems using liquid storage media

The following section discusses storage designs using a liquid storage medium. The main advantage of these systems is that in many applications the storage medium also acts as a heat transfer medium. This allows very high heat capacities to be achieved. The most commonly used material is water. Although cavern storage systems play a

minor role in thermal energy storage, there are many applications for storing heat and hot water in earthen basins and tank accumulators made of steel, concrete or plastic. Cold storage at temperatures between 0 and 20 °C is also discussed in this section.

Storing hot water in steel, concrete, or plastic containers

This section discusses the most commonly used types of storage for sensible heat energy. The storage medium is water. Temperatures up to 95 °C. These storage systems are typically used as buffer storage in small and medium-sized heating systems to balance supply over short periods. Larger models are used to connect local or central heating networks to balance supply over longer periods. Steel, concrete or plastic tanks are used to store hot or warm water. In larger storage systems, watertight, reinforced concrete is increasingly being used instead of steel tanks. Due to the low level of injected oxygen, domestic hot water is usually stored in corrosion-resistant materials such as steel, coated steel or copper. In sealed hot water or buffer systems with very low levels of injected oxygen, untreated and uncoated steel can be used. The use of glass-fibre reinforced plastics is a recent development.

These storage systems can be installed above ground, underground, in semi-pits or in buildings. A good surface-to-volume ratio is preferred to minimize thermal losses. As a result, larger storage systems have proportionally lower heat losses. This means that spherically shaped storage areas are the best, but for a number of reasons they are rarely built in practice. Spherical storage is difficult to design and construct. Due to the layers in the water, the contact area between hot and cold water must also be minimized to prevent thermal conductivity that disrupts the layer. For these reasons, cylindrical hot and cold water tanks are usually used, which can withstand extreme pressure. With these storage areas, a typical height-to-diameter ratio is between 2:1 and 5:1. Insulating materials include foamed materials, mineral wool, glass foam and glass wool.

Hot and hot water storage systems use the concepts of direct and indirect contact charging and discharging. With direct charging, the heat storage medium also acts as a heat carrier: there is no hydraulic separation. The produced warm water is fed to the storage tank above. It then remains above due to its high temperature and low density. If necessary, it can also be withdrawn from there.

These systems are called stratified storage systems. An example of such a system is a buffer storage tank filled with hot water. When heating is required, water is taken from the buffer tank and supplied to the heat removal points (radiators and heating batteries) via a pipe network. The advantages of direct charging and discharging are simple design and the lack of internal storage components. With indirect storage charging, the heat is transferred to the storage medium by a heat exchanger. This design is used in heat supply systems when different fluids are used for heat transport and heat storage. Here, the heat from the top of the storage device is transferred to the hot water

circuit by a heat exchanger. Tank-accumulator-in-tank-accumulator storage systems using three different heat carriers are an example of such systems (Figure 5.5.8). A water-alcohol mixture is used in solar collector circuits to protect against freezing and frost. The heat is transferred from the mixture to the hot water in the tank using a heat exchanger.

This tank contains a second tank filled with domestic hot water. The advantage of indirect charging and discharging is that it prevents corrosion, contamination and loss of storage material. This is because the storage components are sealed systems that do not come into contact with the environment or the storage material.

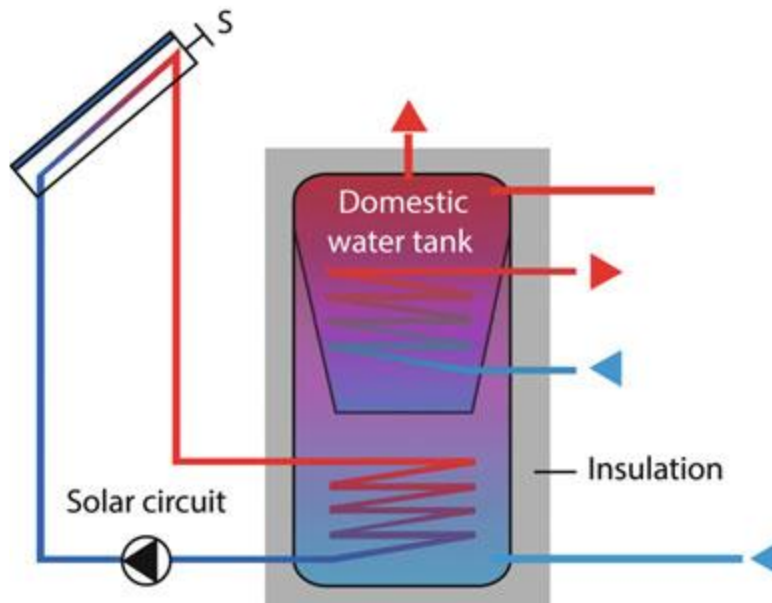


Figure 5.5.8 Tank-within-a-tank thermal storage system.

Small and medium-sized storage tanks (0.1-200 m³) often use cylindrical pressure vessels with two convex bottoms. Larger storage tanks (up to about 50,000 m³) usually use similar vessels with flat bottoms. The facilities are insulated by means of foam insulation between the steel wall and the cover panel or by inserting insulation panels.

Thermal insulators with high diffusion resistance should be used so that the insulator also acts as a moisture barrier. Soft foam molding made of synthetic rubber (EPDM) is often used. Polyurethane is usually used for joints and seams.

Vessels made of glass fiber reinforced plastics (GFRP) are smaller than steel vessels. Currently, there are several 1500 m³ GFRP tank accumulators. In the future, GFRP tank accumulators with capacities up to 6000 m³ will be available. GFRP tanks are made from various types of glass mixed with unsaturated polyester and vinyl resins. Rigid polyurethane foam (RPF) is often used as a thermal insulation material.

Concrete, reinforced concrete or prestressed concrete tanks are also used to store hot water and sometimes cold water. In new storage facilities, high-strength and high-

strength concrete can also be used to seal the system, eliminating the need for additional lining.

Cold water storage facilities

Cold water tanks store water at a temperature of 0 to 20 °C. Cold water tanks are connected to a central chiller of the cold water system. The design of this type of storage system is similar to that of hot and cold water storage systems. Cold water storage systems often use steel tanks due to their high mechanical load-bearing capacity and watertightness. In addition, they are easily formed and welded.

Cold water tanks can be constructed above ground, underground and in buildings. In general, these systems are charged and discharged directly. Since they are usually used for short-term storage, they have high charging and discharging capacities. As in hot water tanks, a water layer is formed in cold water tanks. However, since the differences in temperature and density are very small, diffusers are required to control the flow to maintain the layer. In cold water storage systems, typical feed and return temperatures are 6-12 °C.

Caves

Caves are underground caverns that are either artificially constructed (e.g. for ore mining or salt extraction) or naturally formed (e.g. from erosion) and filled with water for energy sources or sedimentation. These systems are directly charged and discharged. Caves cannot be artificially sealed or thermally insulated, meaning that the surrounding ground layers must be very stable and dense. Rock layers made of granite, gneiss, igneous rock or other highly stable sedimentary rocks are suitable for storage. Caves can be used for both short-term and long-term storage, but very few cave storage systems currently in operation worldwide use water as the storage medium. Most of these facilities are in Scandinavia. Due to their high ongoing costs, caves are best suited as large storage sites. This reduces the ratio of the one-time cost of developing and building a facility to the storage volume. Cave storage tanks are most commonly used for gas storage (natural gas and hydrogen).

Warm storage and hot water storage in ground basins

The main advantage of underground hot water storage is that the storage medium is water, a cheap material with high storage capacity and relatively easy to process. Thermal storage with reinforced or self-supporting covers can withstand high loads, such as vehicle or pedestrian traffic (Figure 5.5.9).

Floating covers cannot support these loads, but they are less expensive. Thermal storage with reinforced or self-supporting covers is also easier to access when maintenance is required (Figure 5.5.10).

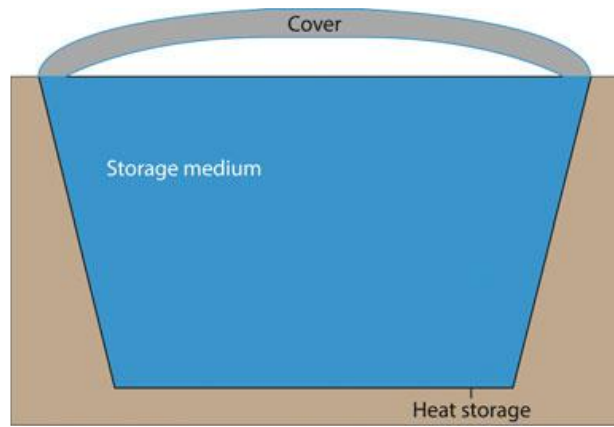


Figure 5.5.9 Design of a self-supporting hot water thermal storage device.

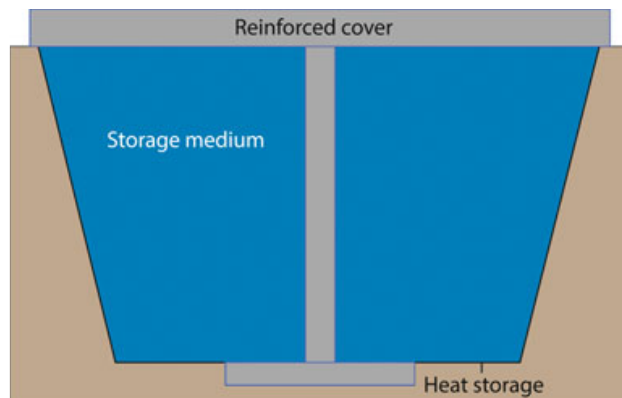


Figure 5.5.10. Design of a hot water thermal storage device with a reinforced cover.

Thermal storage with reinforced or self-supporting covers can withstand higher loads, such as vehicle or pedestrian traffic (Figure 5.5.9). Floating covers cannot support these loads, but they are less expensive. Thermal storage with reinforced or self-supporting covers is also easier to access when maintenance is required (Figure 5.5.10).

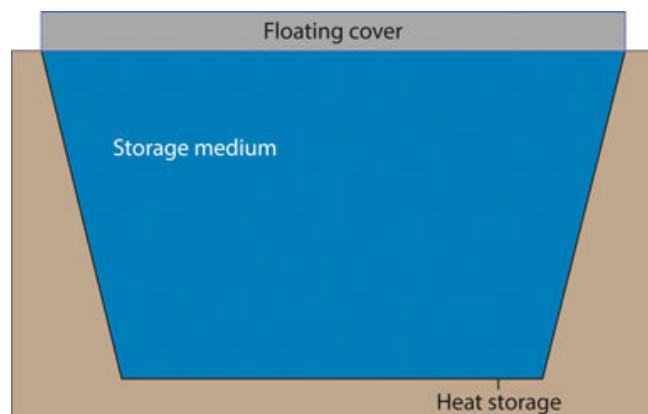


Figure 5.5.11. Design of a hot water thermal storage device with a floating cover.

Storage facilities with reinforced covers can be built in any size, but the necessary conduits in the floor and ceiling can lead to leaks and thermal bridges. Combined with the complex foundation, this can make the construction difficult overall. Floating covers are simple, inexpensive, and suitable for any storage volume, but they increase the difficulty of loading and unloading (Figure 5.5.11).

Currently, the planning and construction of self-supporting earthen basins with covers is very expensive and complex. In addition, the storage volumes and storage surfaces are limited.

Aquifer storage facilities

In aquifers, water is pumped into a cavity of loose or hard rock in the soil and then stored there. Aquifers are located either above or below impermeable aquifers. Types of aquifers include:

- Porous aquifers: sand and gravel;
- Fractured aquifers: sand, limestone, and basalt;
- Karst aquifers: limestone.

- The water-saturated soil can be accessed through one or more individual wells. The natural groundwater flow determines where these well systems can be installed. Hot or cold water is supplied to the “hot” or “cold” well for recharge. The storage capacity of the water and soil is used to store energy. Operation requires the extraction of water from one well. The water is then pumped through the well system through a heat exchanger to another well. The heat exchanger transfers heat to the secondary circuit or extracts heat from the secondary circuit. This allows the heat or cold to be stored and later re-supplied. This also means that the system is directly charged and discharged.

Filtration, chemical water treatment and maintenance of the system pressure are controlled above ground. This storage system can be charged with solar thermal energy in the summer, for example at operating temperatures above 50 °C. Aquifer storage systems are used for long-term storage. They store excess heat in the summer and deliver it back in the winter. In combination with heat pumps or sorption chillers, aquifer storage systems can also be used for heating and cooling buildings.

Aquifer storage systems are typically located at depths of 100-500 m. The spacing between wells is typically 50-300 m. Most of these projects are located in Scandinavia, the Netherlands or Belgium. Typical storage volumes are between 5,000 and 200,000 m³. In Germany, suitable geological conditions for these systems are found in the foothills of the Alps and in northern Germany.

For example, a building with an aquifer storage system was built in Rostock. The Reichstag building in Berlin also uses aquifers. Approximately 77% of the energy stored in the Reichstag building is recycled (or 67% including pumping energy). With cold storage, this ratio is approximately 93% (or 88%). In addition to heat losses, turbidity in heat exchangers and pumps can also cause problems. Depending on the type of soil, iron deposits in the pipes can also be a problem. Construction of an aquifer storage system (Figure 5.5.12) requires permission from local water authorities to recharge groundwater.

Earth basins

Groundwater thermal storage systems are categorized by storage material. Gravel-water thermal storage structures and soil thermal storage structures using a sand-water mixture can be constructed almost anywhere. After the groundwater basin is excavated, drainage mats are laid to prevent surface water from seeping into the underlying wall layers. Other components of the walls include a protective membrane, sealing layers, thermal insulation and a moisture barrier. The volumes of the facilities built to date range from 100 to 10,000 m³.

Groundwater basins can be charged indirectly using a heat carrier fed through plastic pipes. They can also be charged and discharged directly by pumping water through vertical and horizontal gravel paths. In both cases, the heat carrier (water) and the soil (gravel and sand) used serve to store thermal energy (Figure-5.5.13). Due to the thermal insulation materials used, the maximum storage temperature is limited to approximately 85 °C.

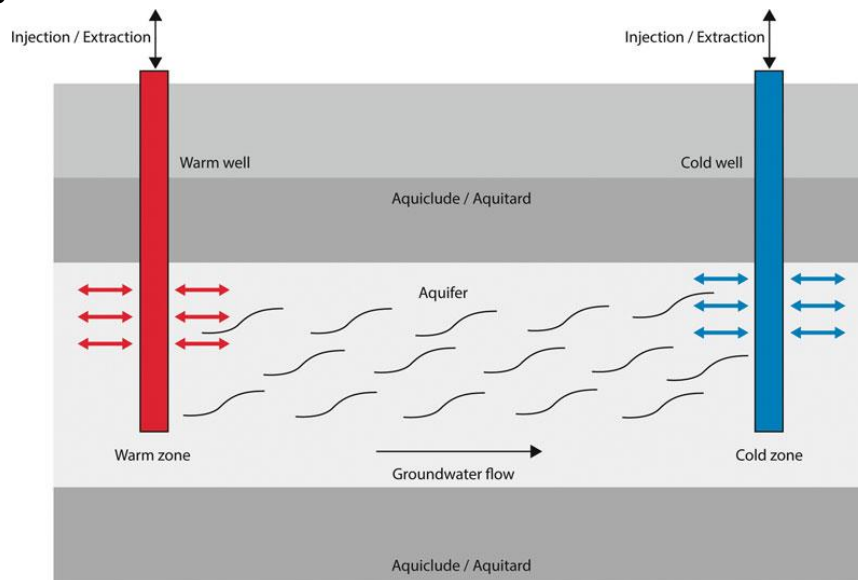


Figure-5.5.12. Aquifer storage system.

High temperature storage systems

High-temperature storage media include pressurized water, molten salts, mineral oil, and liquid sodium.

Steam accumulators and transportation. Storage for pressurized water

Steam accumulator technology (also known as “floating pressure water tank” or “Roots”) is the most modern technology in terms of heat storage used in process heat applications. Steam accumulators are charged by the condensation of steam fed into a liquid volume under pressure. During the discharge process, the pressure in the storage vessel decreases and saturated steam is released. Since water is used as both the storage

medium and the heat transfer fluid, high flow rates and short reaction times are possible (Figure 5.5.14). Thus, a buffer storage process with high thermal capacity is possible.

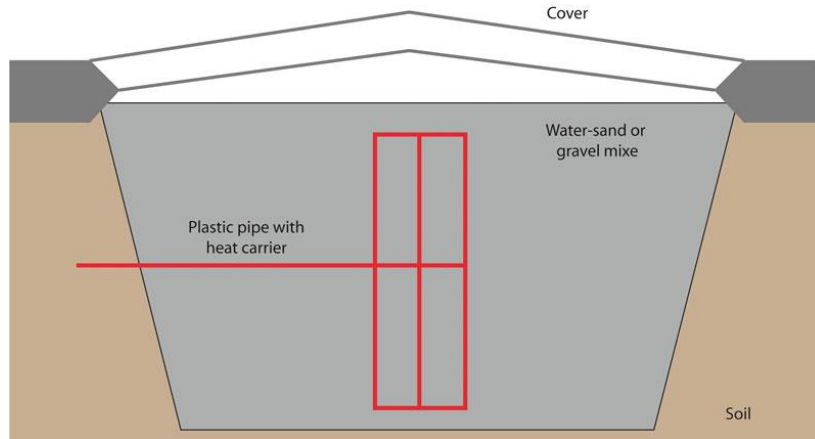


Figure 5.5.13 Groundwater heat storage facility.

Steam accumulators utilize sensible heat storage in pressurized saturated-liquid water, and the capacity is determined by the volume of the pressure vessel. The storage volume corresponds to the change in sensible energy of the liquid volume. While a larger pressure drop is advantageous over volumetric storage, the acceptable pressure change may be limited by efficiency and process requirements. The maximum operating temperature is limited to the critical point of saturated water (374 °C, 221 bar). The pressure vessel is advantageous in terms of cost. Thus, typical systems operate at a maximum temperature of 250 °C and an upper pressure limit of 40 bar.

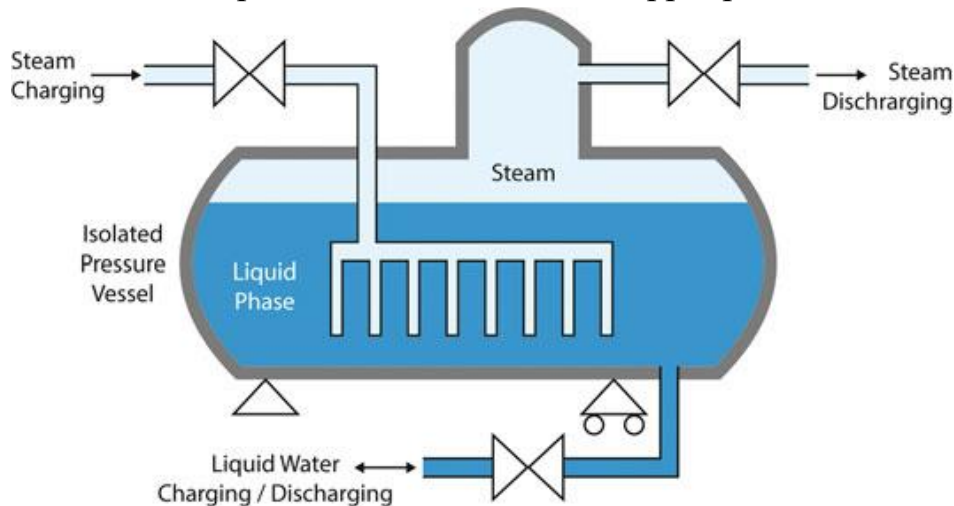


Figure 5.5.14 Schematic of a steam accumulator for sensible-heat storage in pressurized water (source DLR).

Ruths storage systems can equalize and compensate for variations in steam or pressurized water demand or supply. Steam accumulators can provide heat to industrial steam networks, cover peak power, supply batch processes or be used in conjunction with steam turbines or cogeneration. There are several processes that use steam accumulators, such as the mineral, pulp and paper, food, textile and chemical industries.



Figure-5.5.15. Two-tank molten salt system at the Andasol-3 solar thermal power plant (DLR has permission for this image).

For pressurized water, “fluctuating storage” or “thermocline” storage systems can also be used. Hot and cold water are in the same tank. The hot, less dense water is at the top, and the cold water is at the bottom. A temperature transition zone is formed between the cold and hot parts. This zone moves up and down during charging and discharging. The outlet temperature remains almost unchanged until the end of the discharge. A tall vertical cylindrical tank is used to achieve good stratification. Typical displacement tanks supply pressurized water up to 180 °C (about 30 bar). Large-scale variable storage systems are commercial products in the district heating sector.

Molten salt storage systems

Similar to domestic non-pressurized hot water storage tanks, high-temperature heat (170-560 °C) can be stored in molten salts by changing the temperature. Nitrate salts, known from their use as fertilizers, are used as storage media. These salts are solid at room temperature and liquid at higher temperatures. The high-temperature properties of molten salts are similar to those of water in terms of bulk density, viscosity, and transparency. The main advantages of molten salts are low cost, high thermal stability, and low vapor pressure. The low vapor pressure leads to pressureless storage designs (Figure-5.5.15). In general, molten salts have been used in a number of industrial applications involving heat treatment, electrochemical reactions, and heat transfer. The use of salts requires consideration of the lower temperature limit defined by the melting temperature. One aspect of molten salts is the undesirable freezing during operation. Freezing must be prevented in pipes, heat exchangers and storage tanks. Therefore, auxiliary heating systems are often installed. The thermal stability of the molten salts sets an upper temperature limit. Salt mixtures, rather than single salts, have the advantage of having a lower melting temperature. These mixtures can have thermal stability limits similar to the single salts of the mixtures. Thus, salt mixtures can have a larger temperature range than single salts. For solar thermal power plants (also called “concentrated solar power” - CSP plants), sodium nitrate salt mixtures are

the preferred fluids for storing thermal energy. Typically, a non-eutectic salt mixture of 60% sodium nitrate and 40% potassium nitrate by weight is used. This mixture is commonly called solar salt. The minimum operating temperature of solar salt is usually set at 290 °C (limited by the liquid temperature of about 250 °C). The maximum operating temperature is 560 °C, determined by thermal stability. The thermal energy storage system consists of a cold (e.g. 290 °C) and a hot (e.g. 560 °C) unpressurized tank-accumulator. For a temperature difference of 250 K, the storage density of the medium reaches a value of approximately 200 kW/m³. There are currently commercial solar thermal power plants with molten salt storage devices with a capacity of up to 5,000 MWh (Solana from Abengoa in the USA). In 2015, the installed capacity of molten salt storage in concentrated solar power plants was greater than 30,000 MWh, and the capacity of molten salt storage connected to the grid, expressed as electrical power, was 1,500 MWh. Typical storage capacities for concentrated solar power use range from 3 to 15 hours. At the time of writing, typical storage costs range from 20 to 75 euros/kWh, depending on parameters such as capacity and power, as well as the temperature difference between the hot and cold tanks. Research on storage components is focused on replacing two tanks with a single tank with a height stratification. There are three distinct zones in a tank: cold zone (at the bottom), mixing zone (in between) and hot zone (at the top). Some systems operate without packing material (thermocline concept), some systems use additional packing material within the fluid volume (thermocline-filler concept). This packing material can improve stratification and reduce the effects of free convection and conduction that can destroy the thermocline zone. In addition, the overall capital cost can be reduced by replacing the expensive fluid with a cheaper packing material.

Other high temperature fluids

Mineral, synthetic or silicone oils are suitable for short-term thermal storage. Mineral oil can be used at ambient pressures up to about 300 °C. Synthetic oils are thermally stable up to 400 °C, but at higher temperatures they must be kept under pressure, which is often not economical. Flammability and the cost of mineral oil are disadvantages of oil systems. Potentially, a cheap filler material such as cast iron or natural stone could store thermal energy and replace some of the heat-oil volume. Applications under consideration are conventional and solar-thermal power plants. Liquid sodium is rarely used as a storage medium. Although this material has excellent thermal conductivity and can be used over a wide temperature range, safety precautions are also required to prevent strong exothermic reactions, for example with water.

Compared to latent or thermochemical storage systems, sensible heat storage devices are the most advanced in development. This means that there is less scope for further development and cost reduction than the other two types of storage.

Nevertheless, as new application areas are discovered, improvements to sensible heat storage systems continue to be made (e.g. by reducing heat losses or by improving layers). High-temperature thermal energy storage is a cross-sector technology that is not only used in the industrial process heat sector, but also in other application areas such as solar thermal power plants. Molten salt storage has recently become fully commercial in large-scale solar thermal power plants. Thus, the latest high-temperature TES developments can be transferred to both industrial process heat and conventional power grids. The transfer of commercial thermal energy storage technologies and new high-temperature thermal energy storage to the research and development phase will also lead to reduced investment costs, simplified system integration, higher energy densities, and adaptation to special heat transfer fluids or new temperature ranges.

The latest combined heat and power (PtH) plants combined with hot water thermal energy storage demonstrate the potential for connecting the electricity and heat networks to further integrate volatile, renewable electricity generation. In the future, heat pump systems and high-temperature thermal energy storage will offer additional efficient solutions in this area. Due to economic constraints, seasonal solar thermal storage will only be possible in the near future using large reservoirs. This situation can be changed in two ways: by developing innovative insulation concepts that reduce the costs sufficiently for this application, or by using stratified charging devices that directly charge the facility by storing the heat transfer fluid at the appropriate temperature layer.

5.6. Latent storage of thermal energy

To determine the heat Q_{Lat} stored in the latent heat storage device, the enthalpy of fusion ΔH_{ls} with the phase change from solid to liquid must be added to equation 5.5.1:

$$\begin{aligned}
 Q_{Lat} &= \int_{T_1}^{T_2} mc_p dT + \Delta H_{ls} = \bar{T} \Delta S_{12} + T_{1s} \Delta S_{1s} \approx \bar{T} (\Delta S_{12} + \Delta S_{1s}) > Q_{Sens} \\
 &= \int_{T_1}^{T_2} mc_p dT.
 \end{aligned} \tag{5.6.1}$$

The phase change occurs at temperature T_{1s} , with a concomitant change in entropy ΔS_{ls} . The equation shows that, assuming a phase change between temperatures T_1 and T_2 , a latent thermal storage device can store more heat than a sensible heat storage device with a small change in temperature.

Under what circumstances does it make sense to use latent thermal energy storage? Water is a standard storage medium for sensible heat as well as latent cold. In

the past, for example, ice cubes were cut from frozen lakes to size and then stored in cellars until summer to cool food.

Water is abundant, cheap, environmentally friendly, and has a high specific heat capacity c_p , but it can only be used at ambient pressure at temperatures between about 5 and 95 °C. The following example shows that in some temperature ranges, phase change materials (PCMs) can have advantages over sensible-thermal storage.

As a general rule:

$$\Delta q^{PCM} = c_p^{PCM} \cdot \Delta T + \Delta h \quad (5.6.2)$$

Since the specific heat capacity of a phase change material in the solid phase is generally not the same as that in the liquid phase, three conditions determine the thermal energy stored in a latent heat storage reservoir:

$$\Delta q^{PCM} = c_{p,solid}^{PCM} \cdot \Delta T_{solid} + \Delta h + c_{p,liquid}^{PCM} \cdot \Delta T_{liquid} \quad (5.6.3)$$

The heat stored in a rational heat storage facility (in this case, a water storage facility) is defined as follows:

$$\Delta q^{water} = c_p^{water} \cdot \Delta T \quad (5.6.4)$$

with the following material information:

$$c_p^{water} \approx 4.2 \frac{J}{g \cdot K}$$

$$c_p^{water} \approx 2.1 \frac{J}{g \cdot K} \text{ and } \Delta h \approx 150 \frac{J}{g}$$

where c_p^{PCM} is the average specific heat capacity (average over solid and liquid phases) and ΔT total temperature difference

As a result:

$$\frac{\Delta q^{pcm}}{\Delta q^{water}} = \frac{c_p^{PCM}}{c_p^{water}} + \frac{\Delta h}{c_p^{water} \cdot \Delta T} \quad (5.6.5)$$

$$\frac{\Delta q^{pcm}}{\Delta q^{suv}} \approx 0.5 + \frac{35.7K}{\Delta T} \quad (5.6.6)$$

In this example, the storage capacity of a latent thermal storage device with a temperature difference of less than 70 K exceeds the storage capacity of a sensible hot water storage facility.

A phase change material with a melting enthalpy of 250 J/g (about the average melting enthalpy for salt hydrates) increases the potential temperature difference by 120 K. This means that the storage density of a phase change material-storage system relative to the material is superior to a hot water storage system. This difference is especially relevant when the application limits the usable temperature difference.

As a result, latent thermal storage systems are well suited for cold storage. Building cooling typically uses systems with temperature differences of around 6 K.

However, building heating and hot water supply typically use systems with temperature differences of 20-50 K or more. Water is the standard phase change

material for cold storage around 0 °C because of its very high enthalpy of fusion of 334 J/g (Table 5.6.1).

Figure 5.6.1 shows the basic relationship between temperature and the amount of heat stored. Phase change material-2, shown in green, has a lower enthalpy of fusion than phase change material-1. As a result, it should only be used with smaller temperature differences. With larger temperature differences, the heat capacity of water compensates for the enthalpy of fusion of the phase change material. In the diagram, the gradient of the line is equal to the heat capacity.

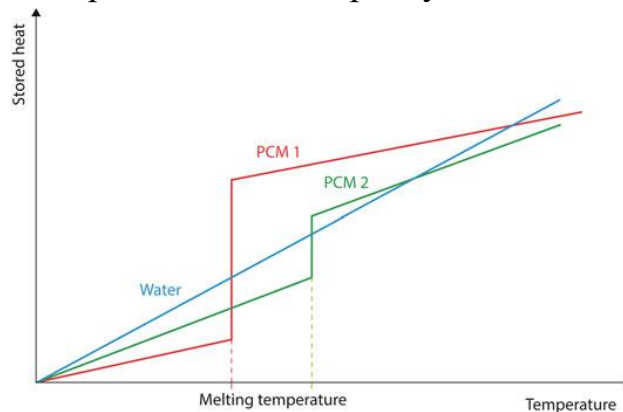


Figure 5.6.1. Amount of heat stored versus temperature.

Features

The main advantage of phase change material storage systems is that they can store large amounts of energy with small temperature differences. This means that the exergy losses of the storage are lower than in sensible storage concepts, since the addition of energy does not even increase the temperature difference with the environment. However, the implementation of the system is technically challenging. Depending on the material used, separation, undercooling, and uncontrolled volume changes can occur. Insufficient heat transfer, leading to low capacity, is also often a problem. Design solutions must be found here to ensure that the storage system meets the required parameters. As a result, phase change material storage systems are often more expensive than comparable sensible storage systems.

Characteristics of latent heat storage materials

The standard method for characterizing phase change materials is differential scanning calorimetry (DSC). Here, a small sample (5-10 mg) is placed in a crucible. This crucible is then placed in a symmetrical measuring chamber together with an empty crucible (control) and subjected to a defined temperature program. Two measurement methods are used. Typically, dynamic measurement with temperature ramps of 2 K/min or step measurement in which the temperature is gradually changed until thermodynamic equilibrium is reached. The heat flow is calculated from the temperature difference between the temperature sensors under the two crucibles

(Figures 5.6.2 and 5.6.3). An enthalpy-temperature diagram can then be developed using this data. Figure 5.6.4 shows a typical result of a dynamic differential scanning calorimetry measurement with a heating and cooling rate of 2 K/min. Here, the hysteresis is very small. This is because the substance being measured is pentadecane, an alkane. A linear heating curve at low temperatures and a cooling curve at high temperatures indicate a relatively constant specific heat capacity. However, with pentadecane, two-phase changes are not typical. The first phase change with a low enthalpy is equivalent to a solid-solid phase change. The second phase change at 5-15 °C indicates a solid-liquid phase change.

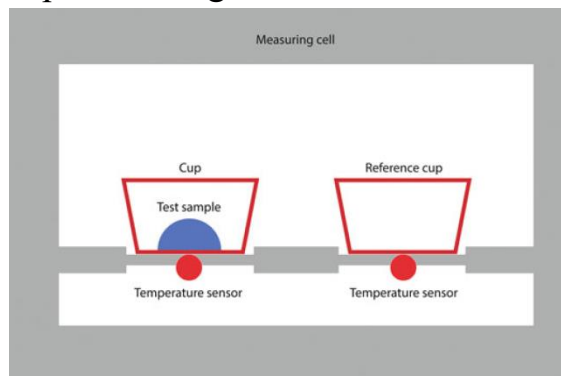


Figure-5.6.2. Symmetrical measuring chamber with test and control crucibles, as well as a temperature sensor under the crucible.

The T-history method can also be used to characterize a phase change material. In this procedure, two identical containers, one filled with the tested phase change material and the other with a control material, are subjected to rapid changes in ambient temperature. The responses of the samples are then used to produce a temperature-time diagram.



Figure 5.6.3. Differential scanning calorimetry Q 2000.

The control material is then required to determine the heat capacity and enthalpy of transition. Figure 5.6.5 shows the measurements resulting from using this procedure. The results show that the tested phase change material has a melting range rather than

a discrete phase change temperature. The solidification and melting temperatures differ. This effect is called hysteresis and can be as much as 10 K with some materials. In addition, there is evidence of undercooling, as mentioned above, with salt hydrates.

Latent-heat storage materials

There are a number of materials suitable for phase change heat storage. Figure 5.6.6 gives an overview of the different classes of materials, together with their melting temperatures and melting energies. In practice, the solid-to-liquid phase change is the most commonly used. Despite the high enthalpy of phase change, the transition from liquid to gas results in a large phase change that is difficult to control. As a result, it is rarely used. In some classes of materials, cooling of the liquid phase can lead to undercooling. Here, the solution cools below the melting range without crystallization: the sample is then in a metastable state.

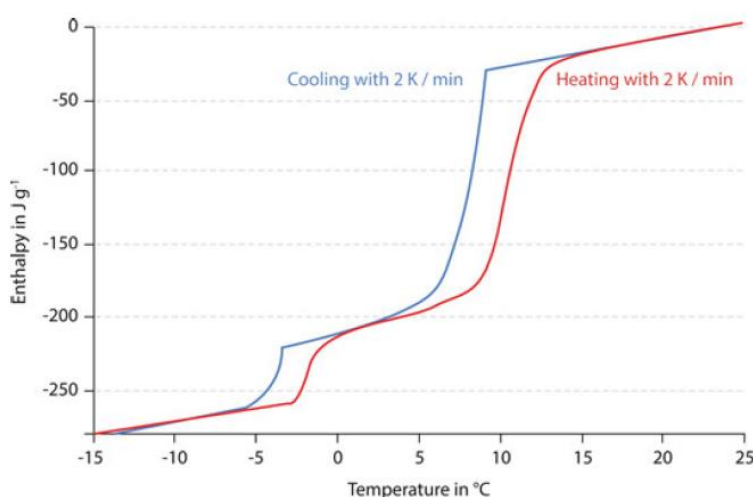


Figure-5.6.4. Enthalpy-temperature diagram obtained from dynamic differential scanning calorimetry measurements.

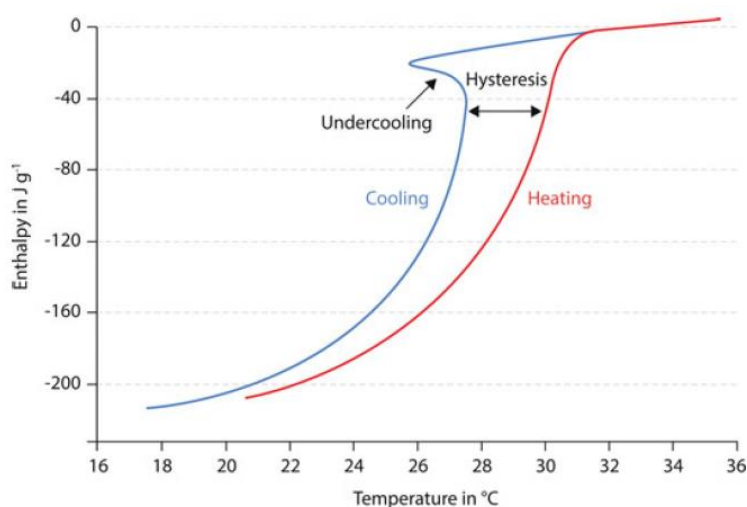


Figure-5.6.5. Typical T-history measurement of the phase transition of calcium chloride hexahydrate.

When crystallization begins, the material crystallizes and the enthalpy of the phase change is released. The material also heats up to or below the crystallization temperature, which (in a material without hysteresis) is equal to the melting temperature. The actual maximum temperature reached is determined by the low cooling rate at the beginning of crystallization and the heat losses to the surroundings during crystallization.

- Not all materials are suitable for use as phase change materials. The following are the criteria required for a PCM material.
- Physical criteria:
 - high phase change enthalpy and specific heat capacity;
 - appropriate melting temperature;
 - good thermal conductivity;
 - reversible phase change;
 - controlled low cooling;
- technical criteria:
 - physical and chemical stability;
 - low vapor pressure;
 - controlled volume change;
 - low corrosion with building materials;
 - appropriate melting and solidification properties; cycle stability;
- economic criteria:
 - low cost;
 - readily available;
 - environmentally friendly; low toxicity.

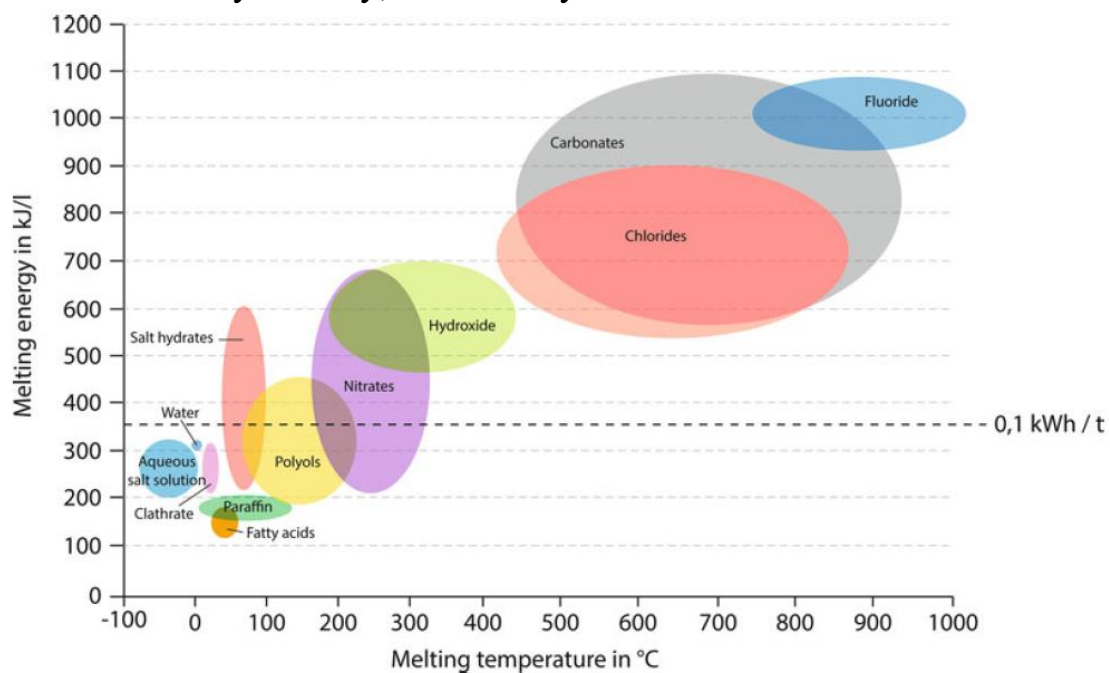


Figure 5.6.6. Classes of phase change materials.

For example, water meets these criteria. Due to its melting point, water is commonly used for air conditioning. Table 5.6.1 shows a selection of different materials, along with their melting enthalpies and melting points. In practice, the salt hydrate and paraffin material classes, other than water, are more commonly used in latent heat storage systems. As a result, they are described in more detail in the next section.

Salt-hydrates

Salt hydrates are salts that bind with water in strict stoichiometric ratios. Even small changes in the stoichiometric concentration change the enthalpy of transition of the salt hydrate. As a result, one criterion for using salt hydrates as PCMs is a constant water content throughout their service life.

Table-5.6.1

Properties of various phase change materials.

Material class	Material	Molecular formula	Melting temperature, °C	Melting enthalpy, kJ/kg	Thermal conductivity, W/m ² K	Density, kg/m ³
Eutectic salt water mixture	Water/sodium chloride	H ₂ O/NaCl	-21	222	-	1 165 (liquid 20 °C)
Water	Water	H ₂ O	0	334	0.597 (liquid 20 °C)	998 (liquid 20 °C) 917 (solid)
Salt hydrate	Calcium chloride hexahydrate	CaCl ₂ ·6H ₂ O	29	171	0.54 (liquid 39 °C)	1 562 (liquid 32 °C)
	Disodium hydrogen phosphate Dodecahydrate	Na ₂ HPO ₄ ·12H ₂ O	35-40	280	0.476 (liquid)	1 442 (liquid)
	Sodium sulfate pentahydrate	Na ₂ S ₂ O ₃ ·5H ₂ O	48	187	-	1 670 (liquid)
	Sodium acetate trihydrate	Na(CH ₃ COO)·3H ₂ O	58	226	-	1 280 (liquid)
	Magnesium nitrate hexahydrate	Mg(NO ₃) ₂ ·6H ₂ O	89	149	0.49 (liquid 95 °C)	(liquid)
	Magnesium chloride hexahydrate	MgCl ₂ ·6H ₂ O	117	165	0.57 (liquid 120 °C)	(liquid)
Salts	Sodium nitrate	NaNO ₃	307	172	0.51 (liquid)	1 900 (liquid)
	Potassium nitrate	KNO ₃	333	266	0.5 (liquid)	1 900 (liquid)
	Lithium nitrate	LiNO ₃	254	360	0.6 (liquid)	1 800 (liquid)
Salt mixture	Sodium nitrate + potassium nitrate	KNO ₃ /NaNO ₃	222	100	0.5 (liquid)	1 900 (liquid)
	Lithium nitrate + sodium nitrate	LiNO ₃ -NaNO ₃	194	265	0.5 (liquid)	1 900 (liquid)

Paraffin	Octadecane	$C_{18}H_{38}$	28	245	0.15 (solid)	777 (liquid)
Fatty acid	Lauric acid	$CH_3(CH_2)_{10}COOH$	43	178	0.15 (liquid 50 °C)	870 (liquid 50 °C)
	Myristic acid	$CH_3(CH_2)_{12}COOH$	58	186	-	861 (liquid 55 °C)
Sugar alcohol	Erythritol	$C_4H_5(OH)_6$	120	340	0.32 (liquid 140 °C)	1 300 (liquid 140 °C)
PEG	PEG 6000	$HO-[CH_2-CH_2-O]_n-H$	60	190		1 085 (liquid 70 °C)

Another problem with the use of salt hydrates is the inconsistent melting. During the melting process, several phases are formed, which are then separated due to their different densities. This means that the heat capacity decreases with increasing number of cycles. To counteract this effect and ensure high cyclic stability, salt hydrate containers can be mixed. Macro- or micro-encapsulation or borax injection can also be used to increase stability.

An additional difficulty with the use of many salt hydrates is their strong tendency to cool down. This means that the heat stored in practice is released at low temperatures or not at all. However, as in the case of pocket heaters, undercooling can also be desirable and beneficial. The nucleation can be accelerated by adding inorganic solids, which reduces the tendency to undercool. However, this also reduces the specific storage capacity.

These technical problems aside, salt hydrates have significantly higher bulk density, specific storage density, and better thermal conductivity than paraffins. They are also cheaper than paraffins.

Paraffins

Paraffins are organic compounds consisting of saturated hydrocarbons obtained by distillation of crude oil. They are mainly long-chain alkanes. The melting point can be adjusted as needed by changing the chain length.

Paraffins are used commercially for thermal storage at temperatures from -5 to 120 °C. Since paraffin mixtures consist only of paraffins of different chain lengths, they are uniformly soluble and have high cycle stability.

In addition, unlike metals, paraffins are environmentally friendly and non-corrosive. They do not pose any health risks and only show a tendency to freeze with micro-encapsulation. The disadvantage of paraffins is their flammability. This prevents their use in some environments.

Compared to salt hydrates, they are also expensive and have a very low thermal conductivity of around 0.2 W / (mK). In addition, phase changes can lead to volume changes of up to 20%.

The most important criterion for selecting a phase change material is its phase change temperature. This can be adjusted as needed by mixing alkane chains of different lengths.

However, mixing pure/commercial paraffins generally reduces the enthalpy of phase change and increases the melting range. The use of commercial paraffins has a similar effect. They are often used because of their low cost. For these reasons, the exact properties of these materials should be carefully considered. The RAL PCM certificate (RAL-GZ 896; <http://www.pcm-ral.de>) specifies useful quality and test characteristics in this regard.

Anhydrous salts

The cations are mainly alkali metals (e.g. Li, Na, K) and alkaline earth metals (e.g. Ca, Mg). The anions considered include nitrates, nitrites, hydroxides, bromides, carbonates, chlorides, sulfates, and fluorides. Many anhydrous salts are mixed, resulting in a variety of potential single salts and salt mixtures (binary and ternary systems). For temperatures up to 350 °C, in particular, alkali metal nitrates and nitrites and their mixtures are suitable phase change materials, if requirements such as processing and steel compatibility are taken into account. In the development of concentrated solar power technology, direct steam generation in absorber tubes has attracted interest in latent heat storage systems. The latent heat system acts as a steam storage system for the two-phase heat carrier water/steam. The melting point of candidate phase change materials is between 120 °C and 340 °C. Here, the application of storage concepts using sensible heat storage is generally not cost-effective. Sensible heat storage systems for two-phase heat carriers have low energy efficiency or high exergy due to large temperature gradients between the storage medium and the heat transfer fluid. In a latent heat storage system, both the phase change material and the heat carrier undergo a phase change at approximately the same temperature. In this way, larger temperature differences between water/steam and the phase change material can be avoided. In addition, steam storage solutions can be provided for the process-heating sector.

Composite materials

In their pure form, phase change materials often have low thermal conductivity or are characterized by an increased number of cycles. There are a number of potential solutions to this problem. This section discusses several concepts. To increase the thermal conductivity of the phase change material, for example, a graphite matrix can be impregnated with the phase change material. ZAE Bayern has used this method to increase the thermal conductivity by a factor of 8 to 100 by adding only 10% of the volume of graphite to the phase change material. To avoid poor thermal conductivity

and stability in large phase change material units, especially with phase change materials consisting of several components, the units are often packaged into smaller units. Plastic and metal containers can be used, as can bags made of plastic-metal composite materials. The advantage of this type of encapsulation is that it hermetically seals the phase change material, preventing material exchange with the environment. This is especially useful when using salt hydrates. This form of phase change material can also be used to conveniently retrofit existing thermal storage systems. This is because it is compact enough to easily fit into storage tanks. Air is often encapsulated with the phase change material to avoid volume problems (Figures 5.6.8 and 5.6.9).

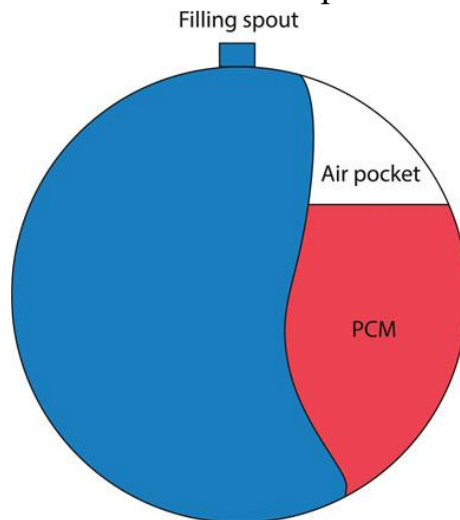


Figure 5.6.8. Macroencapsulated paraffin.



Figure 5.6.9. Microencapsulated paraffin. (Source: BASF SE).

In addition to macro-encapsulation, phase change materials can also be processed using micro-encapsulation. Regardless of the actual phase state, the microencapsulated phase change material behaves like a powder. The advantage of such encapsulation is that it makes the material easier to handle. For example, it can be conveniently incorporated into plasterboard or textiles. Since the encapsulation is small and contains little phase change material, mechanical damage to the encapsulation results in very little loss of phase change material. Among other companies, BASF commercially produces microencapsulated phase change materials. The phase change

material pellets are made of kerosene with a diameter of 2 to 20 μm and are coated with acrylic. The product is typically used in conjunction with building materials that are required for passive cooling of buildings. The microencapsulated phase change material can be mixed with paints, plasterboard, etc. Depending on the material, these building materials can contain 20-40% phase change material by weight.

The incorporation of phase change material into porous structures is another process used to increase thermal conductivity and stability. Wood fiberboard, for example, can be impregnated with phase change material. Paraffins can also be incorporated into plastic matrices. The advantage of these composite materials is that they can be used as a filler in storage areas, just like encapsulated phase change material. The Fraunhofer IFAM in Dresden is currently researching the binding phase change material in a metal matrix. Here, fiber or wire structures, as well as infiltrated metal foams, are used to improve thermal conductivity. With these processes, it is crucial to balance the ratio of phase change material to carrier material. This is because increasing the volume ratio of phase change material increases the storage capacity, but it reduces the thermal conductivity.

Heat transfer concepts

Heat transfer concepts for latent heat storage materials are primarily optimized to maximize the heat capacity. Various technical approaches can be used to improve heat transfer. This section discusses different heat transfer concepts. In general, heat transfer can be passive or active. With passive heat transfer, only free convection moves the phase change material and the heat transfer fluid does not come into contact with the storage medium. Heat transfer occurs at a defined surface (heat exchanger). This type of heat transfer can create molten fronts that negatively affect the heat transfer. Figure 5.6.10 shows examples of simple passive storage system designs. With active heat transfer, the storage material is also kept in motion by a stirrer. Here, a distinction is made between indirect and direct heat transfer. With indirect charging, the phase change material does not come into contact with the heat transfer fluid; With direct charging, the heat transfer fluid flows through the storage medium and no heat exchanger is required. Based on these heat transfer concepts, heat is transported across the boundaries of the system. The thermal capacity of this type of storage system can be increased by installing internal, heat-conducting structures. However, the potential is generally low. Such storage systems are generally inexpensive. Since no external devices are required, they have a high storage density. In addition, there is no material exchange with the environment with these sealed systems. As a result, the system is protected from contamination, corrosion, and loss of storage materials. Freezing packs, as well as macro- and microencapsulated phase change material systems, use this heat transfer concept.

The charging and discharging capacity of these storage systems can be increased by modular design (Figure 5.6.11). Here, a macro-encapsulated phase change material can be used as the storage system filling. The advantage of this approach is that the phase change material can be simply and flexibly integrated into systems with a range of tank geometries. The heat transfer fluid flows through the storage system and the phase change material filling. The total capacity of this type of storage system is increased by the large contact area.

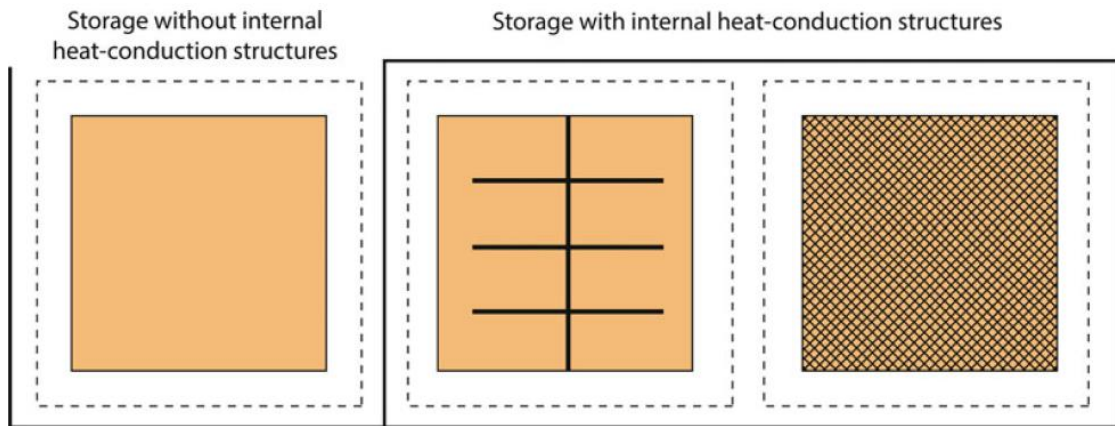


Figure-5.6.10. Simple storage systems with and without internal heat-conducting structures.

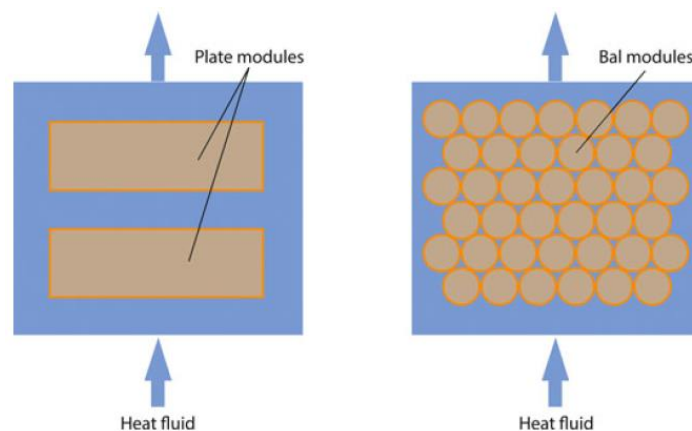


Figure 5.6.11. Module storage system; left: plate modules; right: ball modules (nodes).

However, the flow through the different modules is often uneven. This means that the heat transfer is not consistent across all individual modules. The additional encapsulation materials required also reduce the proportion of phase change material in the storage system, reducing the specific power and capacity. Figure 5.6.11 illustrates the basic design of this concept. Modular storage systems are common in Japan. There, due to the highest hourly electricity prices, the cold for air conditioning is often stored at night. In addition to filling, phase change material plates installed in ventilation ducts also use this heat transfer concept. The simplest form of passive charging is often used in sensitive storage systems. Here, a heat exchanger with a specially designed geometry

for the phase change material is integrated into the storage system. The heat transfer properties, as well as the required capacity and capacity, usually determine the geometry of the heat exchanger. The advantage of this concept is that a high proportion of storage material (over 90%) can be installed. The disadvantages are the earlier formation of the solution, increased heat transfer resistance, and the tendency to sequester with salt hydrates. Figure 5.6.12 shows the basic design of this type of storage system.

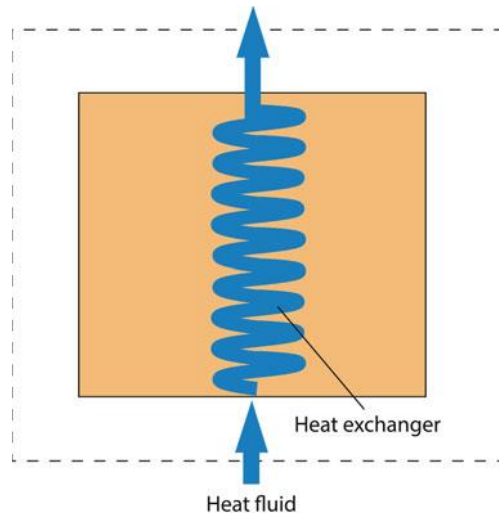


Figure-5.6.12. Storage system with internal heat exchanger.

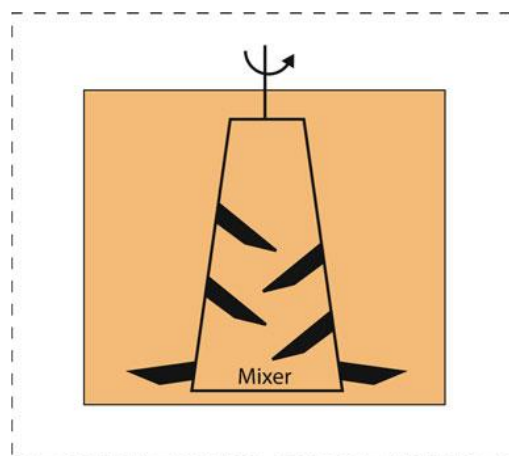


Figure 5.6.13. Active, indirect storage system with internal agitator.

Storage systems with an internal mixer and an integrated heat exchanger (Figure 5.6.13) use the concept of indirect, active heat transfer. The advantage here is, for example, that the separation with salt hydrates is reduced, thus increasing the cycle stability. The constant agitation prevents the formation of a solidification front with poor heat transfer coefficients; this ensures a stable power consumption and power output. However, the overall efficiency is reduced by the additional power consumption of the motor. Here, the solution is to use the mixer as a heat exchanger as well. A separate heat exchanger reduces the proportion of storage material, which reduces the thermal capacity of the storage system. Another option is to rotate the entire

tank-accumulator, so that an internal mixer is no longer needed. Direct contact storage systems use the concept of active, direct heat transfer. Here, the heat transfer fluid flows through the storage material, mixing it at the same time. This process prevents the separation of individual components, especially salt hydrates. The retention time of the heat transfer fluid can be adjusted to match the heat transfer coefficients of the phase change material by changing the geometry of the storage system, as well as the required density difference between the two materials. However, not all heat transfer fluids can be used here, since the materials are in direct contact and can mix with each other. Since no heat exchanger is required, direct contact storage systems are compact and have high capacity. However, these systems are difficult to regulate and it is impossible to allow the phase change material to completely solidify. This reduces capacity. The complete solidification of the phase change material, which allows for reasonable heat storage, is only possible with major technical changes (for example, by installing a bypass). However, this also reduces the possibilities, since the contact area is reduced. Figure 5.6.14 shows the basic design of a direct contact storage system.

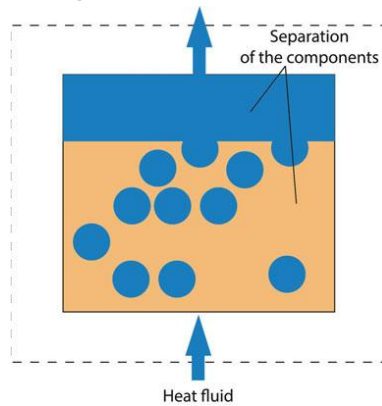


Figure 5.6.14. Direct-contact storage system.

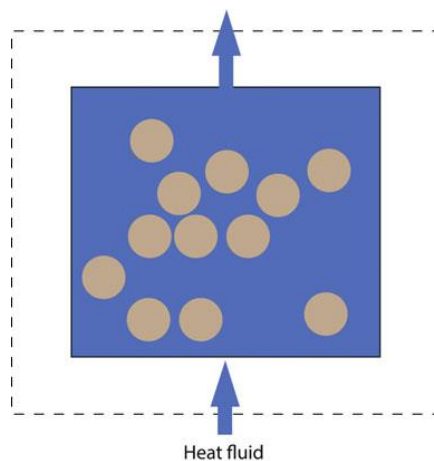


Figure-5.6.15. Slurry storage system.

Another storage concept that allows the heat transfer fluid and the phase change material to come into contact is mixtures. Even though the phase change material is solid, the mixtures can still be pumped. This means that the mixtures can be used to store thermal energy and simultaneously release heat or cold. In these applications,

they are commonly referred to as phase change mixtures. With ice mixtures, the heat transfer fluid and the phase change material are made from the same substance.

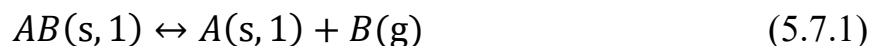
Emulsions containing substances of different hydrophilicity combined with emulsifiers can also be used. In addition, micro-encapsulated phase change material can be proportionally added to the heat transfer fluid to ensure that the mixture is still pumpable. However, care must be taken in this case to ensure that the pump does not damage the encapsulation. In applications, the proportion of phase change material is usually 20 to 40%, or in exceptional cases up to 60%. The lower the proportion of heat transfer fluid, the higher the viscosity. Higher viscosity requires more pumping power. This in turn increases costs. Since the substances are transported in a composite and the resulting temperature level is relatively constant, these systems have lower exergy losses than conventional concepts. This means that mixed storage systems can have very high capacities. Figure 5.6.15 shows the basic design of a sludge storage system.

5.7. Thermochemical-energy storage systems

Thermochemical energy storage systems use the reaction energy of reversible chemical processes or physical surface reactions. They have a particularly high energy density. The reversibility of the process is a key factor here. Furthermore, the energy is stored as reaction energy, not heat. As a result, there are no thermal losses during charging.

Technically at least, this allows for very long storage times. However, for economic reasons, thermochemical storage systems are rarely implemented. When the reaction partners are recombined during discharge, an exothermic reaction releases the stored energy. The following discussion divides thermochemical energy storage systems into three categories: chemically reversible processes, adsorption storage systems, and absorption storage systems.

In thermal energy storage systems, ideally, the endothermic and reversible dissociation of a condensed compound AB results in two reaction products: a condensed reaction product A and a condensed reaction product B in gaseous form. The abbreviations s, l, and g are used to indicate the states of aggregation of reactants, i.e., solid, liquid, and gaseous:



The reaction products can be easily separated because they are in different states of aggregation. This also allows the products to be stored without loss or risk of back reactions. However, the energy storage density is reduced by the volume required to store the gases.

This problem can usually be solved by condensing the gas phase or by using an “open” system that releases the gaseous components into the atmosphere and then reabsorbs them from the atmosphere when needed.

Reversible chemical processes

In principle, thermochemical energy storage systems can use any reversible chemical reaction. The decisive criterion here is the equilibrium temperature at which the products and by-products are in thermodynamic equilibrium. The discharge of the storage system (exothermic reaction) occurs at temperatures below the equilibrium temperature. However, the charge (endothermic reaction) must occur at higher temperatures.

Adsorption storage systems

Adsorption is the physical accumulation of molecules (adsorbates) on the surface of a typically porous adsorbent. Unlike chemical reactions, there is no fixed equilibrium temperature. The heat released is determined by the changing binding energy at the binding sites on the adsorbent and is therefore not constant. The storage density of an adsorption storage system is determined by the enthalpy of vaporization of the adsorbate, the binding forces between the adsorbate and the adsorbent, and the charge transfer between the charging and discharging temperatures.

Absorption storage systems

Unlike adsorption by aggregation, absorption involves the transfer of molecules to the absorbing phase. In this process, the absorbate is a soluble material dissolved in a solvent (the absorbent). Strictly speaking, absorption is not a chemical process, since no chemical reactions take place. Instead, it is a dilution process. Absorbents are liquids (such as salt solutions that can absorb water vapor), but adsorbents are solid, microporous materials.

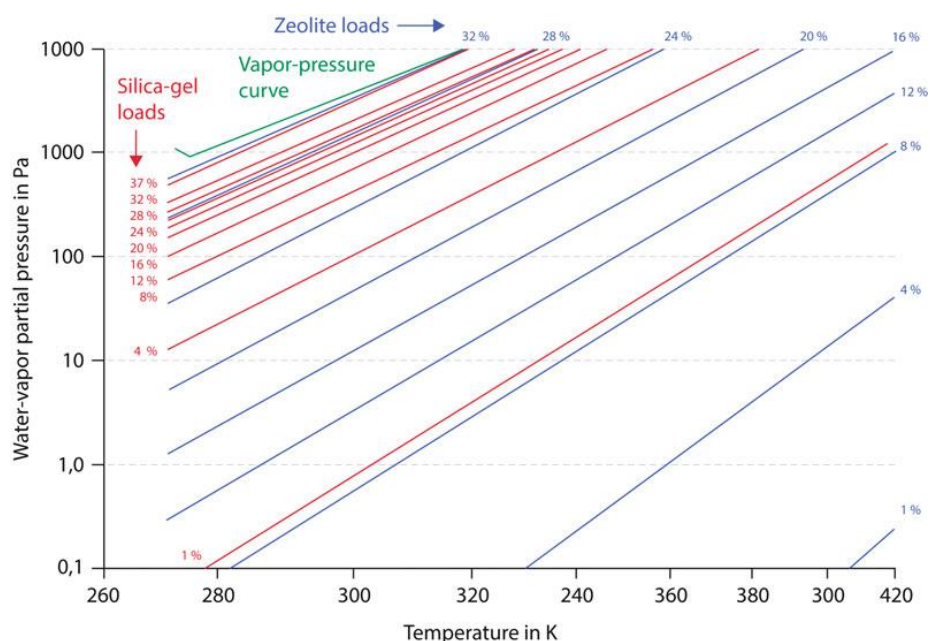


Figure-5.7.1. Isostere diagrams for silica gel/water and zeolite/water.

The energy storage density by absorption is determined by the molar mass of the absorbate, the affinity and aggregate state of the absorbent, and the absorbate.

Storage materials for thermochemical processes

Storage materials used for thermochemical processes are also divided into three categories: materials for chemically reversible reactions, adsorption materials, and absorption materials. The reactions that occur in thermochemical storage systems can be classified according to their reaction equilibria.

These equilibria are usually determined by pressure and temperature. With a certain combination of a substance, such as magnesium oxide and water, a certain reaction equilibrium is achieved at a certain temperature and partial pressure of vapor. These equilibria are called:

- isotherms (curves at the same temperature);
- isobars (curves at the same pressure);
- isosteres (curves with the same amount of water).

Table-5.7.1

Materials for reversible chemical reactions

Reaction type	Equation	Equilibrium temperature (1 bar) °C	Energy storage density kW·h/kg
Salt-hydrate	$\text{MgSO}_4 \cdot 7\text{H}_2\text{O} \leftrightarrow \text{MgSO}_4 + 7\text{H}_2\text{O}$	122	0.463
	$\text{CaCl}_2 \cdot 2\text{H}_2\text{O} \leftrightarrow \text{CaCl}_2 \cdot \text{H}_2\text{O} + \text{H}_2\text{O}$	174	0.091
Hydroxide	$\text{Mg}(\text{OH})_2 \leftrightarrow \text{MgO} + \text{H}_2\text{O}$	268	0.372
	$\text{Ca}(\text{OH})_2 \leftrightarrow \text{CaO} + \text{H}_2\text{O}$	521	0.373
Carbonate	$\text{CaCO}_3 \leftrightarrow \text{CaO} + \text{CO}_2$	896	0.463
	$\text{BaCO}_3 \leftrightarrow \text{BaO} + \text{CO}_2$	1,497	0.298
Metal hydride	$\text{MgH}_2 \leftrightarrow \text{Mg} + \text{H}_2$	293	0.834
Catalytic reactions	$\text{SO}_3 \leftrightarrow \text{SO}_2 + 0.5\text{O}_2$	767	0.340
Steam change	$\text{CH}_4 + \text{H}_2\text{O} \leftrightarrow \text{CO} + 3\text{H}_2$	687	1.672

Table-5.7.2

Adsorption materials

Adsorbent	Maximum content $\text{kg}_{\text{H}_2\text{O}}/\text{kg}_{\text{adsorbent}}$	Adsorption enthalpy $\text{kJ}\cdot\text{t}\cdot\text{s}/\text{kg}_{\text{H}_2\text{O}}$	Energy density $\text{kJ}\cdot\text{t}\cdot\text{s}/\text{kg}_{\text{adsorbent}}$
Zeolite 13X	0.32	0.964	0.308
Silica gel desiccant balls	0.37	0.758	0.280

Figure 5.7.1 shows the curves for the same water content. The maximum content is obtained when the isosteres approach the vapor curves. The air is then saturated with vapor. These conditions maximize the water content. It is 37% for silica gel and 32%

for zeolite. With this type of adsorption system, it is important to note that the increase in isosteres varies with water content. This is not usually due to chemical reactions.

The diagram shows that obtaining a lower content at constant pressure (equivalent to the charging process when used for storage) requires significantly higher temperatures with zeolite than with silica gel. However, lowering the zeolite leads to an increase in temperature.

Reversible chemical reactions

Storage materials for reversible chemical reactions differ in their equilibrium temperature at ambient temperature and the amount of energy converted in the reaction. These parameters are given in Table 5.7.1. The energy storage densities listed here are based on the mass of the reactants and indicate the maximum levels that can be achieved. The reaction of barium carbonate to barium oxide leads to a maximum temperature of 1497 °C. Theoretically, the highest energy storage density is produced by the steam reforming of methane and water to carbon monoxide and hydrogen. Ideally, this density is 1,672 kWh/kg. However, the practical application of such reactions in thermal energy storage systems still requires extensive development and research.

Adsorption

Adsorption processes are classified as a type of thermochemical storage reaction. The most common adsorbents are zeolite and silica gel. Table 5.7.2 shows the maximum water content and energy density for a number of materials. Also, the energy storage densities listed here indicate the maximum achievable levels.

The main criterion for adsorbents in adsorption processes is the specific surface area of the material. The more water that can be stored on the surface, the greater the amount of energy that can be stored. Table 5.7.3 shows the specific surface areas of various adsorbent materials in m² per gram.

Table-5.7.3

Materials for adsorption and their surface areas

Adsorbent	Chemical molecular formula	Specific surface area m ² /g
Aluminum oxide	Al ₂ O ₃	100-400
Adsorber sludge	Masalan, stirol	100-800
Silica gel	SiO ₂	500-800
Carbon molecular sieve	C (antrasit)	500-1,000
Zeolite	Al ₂ O ₃ , SiO ₂	800-1,200
Activated carbon	C	800-2,000

However, the achievable discharge temperature is determined by the specific adsorption enthalpy or binding energy (adsorption enthalpy minus vaporization

enthalpy). This means that under certain conditions, certain adsorbents can adsorb very large amounts of water. Since the water is very loosely bound to the adsorbent, the amount of useful heat released by the storage system is relatively small.

Table-5.7.4

Absorbent materials

Adsorbent	Absorption	Absorption enthalpy kJ/mol _{absorbate}	Molar mass g/mol _{absorbate}	Energy storage density kWh/kg _{absorbate}
H ₂ O(l)	LiBr(s)	49.04	86.85	0.157
H ₂ O(l)	NH ₃ (g)	34.18	17.03	0.558
H ₂ O(l)	H ₂ SO ₄ (l)	95.28	98.08	0.270

Absorption

Table 5.7.4 lists various adsorbents along with their molar masses, enthalpies of absorption, and theoretical energy storage densities. Absorbents generally have very low binding energies. This results in a smaller temperature rise during discharge than solid adsorbents.

Designs

Compared to latent heat storage systems, thermochemical storage designs are significantly more complex. In addition to heat exchange, materials must also be exchanged. In addition, materials must be separated or exchanged simultaneously as heat is added or removed. The challenge here is to design compact storage facilities that maximize the efficiency of heat and material transfer. Broadly speaking, there are two types of thermochemical energy storage design: sealed and open systems.

Sealed systems are typically airtight systems with a vacuum seal that allows the use of any type of gaseous component. The reaction requires a very low pressure in the vacuum. However, the gaseous components must be stored in the system after the charging process. The B components can be compressed to reduce the volume required for storage. Figure 5.7.2 shows the basic design of this type of condenser.

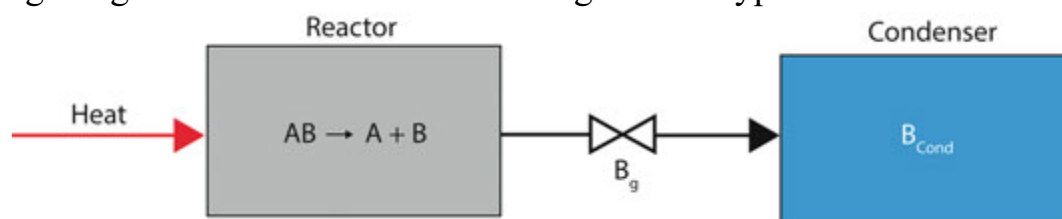


Figure-5.7.2. Gas component condenser design.

For the reverse reaction, the B components must be vaporized again. This evaporation process requires ambient heat and largely determines the useful energy or capacity of the storage device. If they are thermally charged, sealed systems can provide heat by releasing the enthalpy of reaction in the reactor or provide cold by

absorbing the heat of vaporization in the condenser. The zeolite/water material combination, for example, can lead to the formation of cold ice produced by the condenser.

In open systems, these components must be present in the atmosphere. They are released during charging and then reabsorbed during discharging. Various designs can be used with open systems. Since they are the simplest and least expensive, fixed bed designs are the most common. The adsorbent here is in the form of a packing. The adsorbate (usually water vapor) flows through it at ambient pressure (Figure 5.7.3). Other reactor designs replace the mass flow of solid or liquid sorbent with the flow of adsorbate. The advantage of these reactors is that the capacity and storage capacity are decoupled. The capacity is determined by the size of the reactor, but the capacity is determined by the size of the sorbent storage vessel. In practice, these types of reactors are usually used only with liquid reaction mixtures. In adsorption storage systems using solid zeolite, for example, the mechanical stress generated poses a major problem in transporting the zeolite to the reactor. With open adsorption systems, no storage is required for the gaseous components B inside the storage system. Ideally, they are atmospheric components that can be temporarily stored in the ambient air. This increases the energy storage density by a fraction of the volume removed (which was used to store the gaseous components). However, it should be remembered that there must be a sufficient concentration of gaseous components B in the atmosphere to discharge the open adsorption storage system. (Figure 5.7.4).

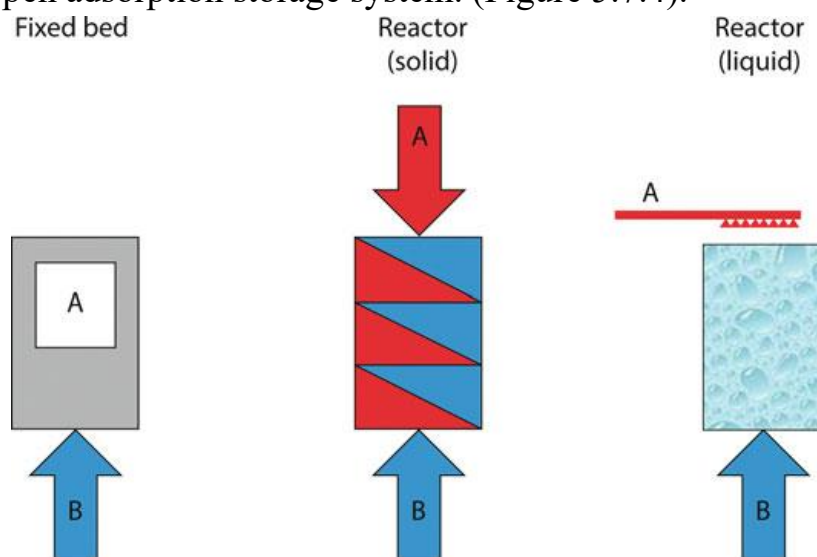


Figure-5.7.3. Various open, thermochemical storage facility designs.

The zeolite/water system is the most common type of open adsorption storage design. It is more of a physical process than a chemical reaction. The operating principle of this storage system is based on the adsorption of water vapor by the action of electrostatic forces on a microporous zeolite (Figure 5.7.5). This process acts as a

discharge process in the storage system. Here, the adsorption enthalpy is released in the form of heat.

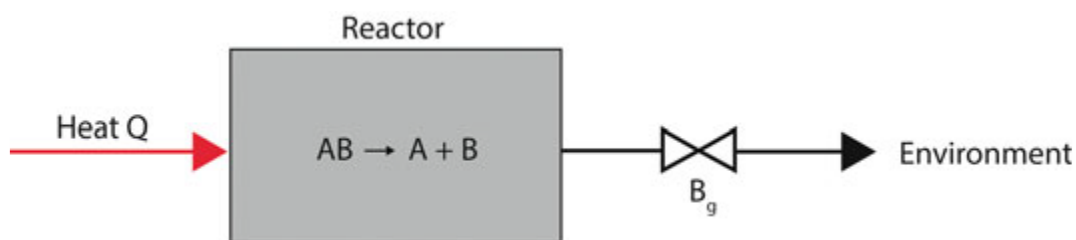


Figure-5.7.4. Design of atmospheric-gas components.

Additional heat is required to charge the storage system. This heat releases the water stored in the zeolite and then evaporates. The vapor produced can be released into the atmosphere using air as a carrier, eliminating the need for temporary storage. However, there must be a sufficiently humid air flow to empty the storage system. As a result, this thermal storage system is well suited for drying processes. This process is already used in many modern dishwashers. Research into the design of efficient thermochemical storage systems is currently focused on developing new storage materials. The goal here is to maximize the enthalpy of reaction (or enthalpy of sorption) at the required temperatures.

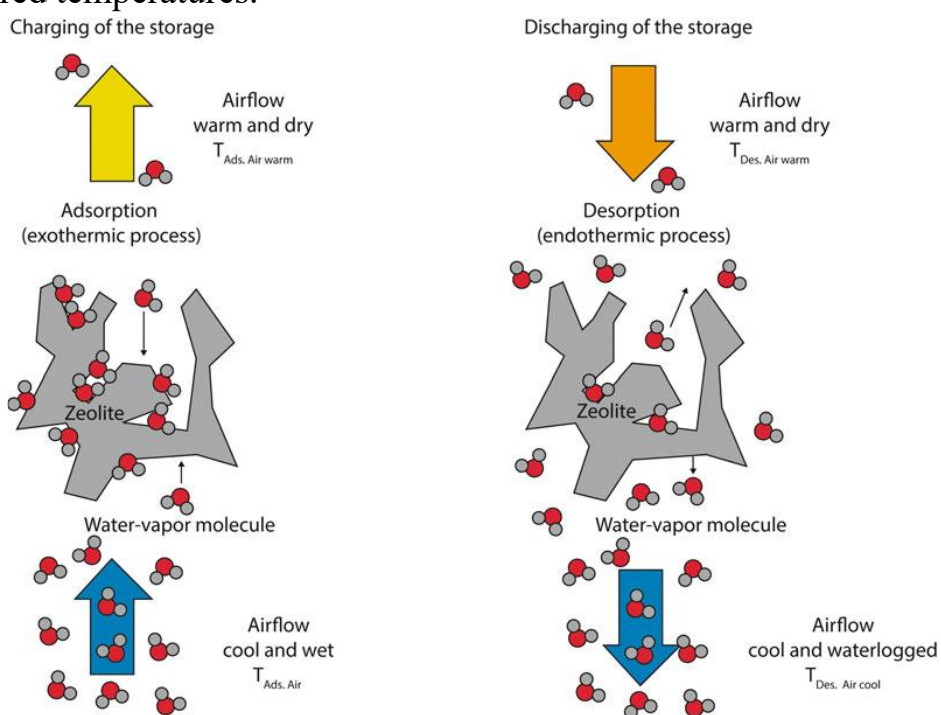


Figure-5.7.5 Zeolite/water storage system.

Research on sorption storage systems is focused on optimizing currently used adsorbents, as well as developing new classes of adsorbents, such as AlPO, SAPO, and MOF. Scientists in Germany and elsewhere are also investigating liquid and solid adsorbent combinations (so-called composite materials). In addition to sorption processes, researchers are also studying other chemical reactions that can be used for high-temperature applications and seasonal heat storage.

Relevant material combinations are also being tested to determine their stability and aging mechanisms under application conditions.

In addition to storage materials, another important area of research is the design of the thermochemical storage system reactor. Here, the goal is to optimize the thermal capacity and, ideally, reduce costs, for example, by minimizing the auxiliary energy required. Modeling tools are being developed for different types of storage reactors to assess their potential advantages.

5.8. Heat accumulators

It is known that the reserves of energy sources on earth are limited. If the annual fuel consumption rate is taken as planned for the year 2000 (25 billion tons) and used at the current level, their reserves can be enough for consumption for 140-150 years. This is, of course, a very short period in historical terms.

Therefore, improving the structure of the fuel and energy balance is an important task. The widespread use of gas and coal instead of oil products as fuel, and the widespread use of nuclear energy, including fast neutron reactors, in places are necessary requirements for the implementation of this task.

For this purpose, great importance is attached to the construction of nuclear and hydroelectric power plants, thermal and solar power plants. The rapid construction of the nuclear power plant in southern Ukraine, the Talimarjon combined heat and power plant and other energy facilities, which are giant structures of our time, is a clear proof of this.

As is known, one of the largest and most inexhaustible energy resources is solar energy. The power of the share of sunlight falling on our planet alone is $2 \cdot 10^{14}$ kilowatts.

The UN also pays great attention to the effective use of solar energy in the national economy. At conferences held at the initiative of UNESCO, many materials being developed in this area are summarized and future problems are discussed.

In our country, in recent years, more than 20 solar installations have been developed based on a comprehensive program of devices using solar energy. In particular, heliotechnical scientists have launched solar installations with an energy production rate of 500 MWh in Crimea. A high-temperature solar furnace created by the Institute of Physics and Technology of the Academy of Sciences of Uzbekistan has been tested and recommended for production.

A number of solar devices have been built and tested in the use of solar energy in the national economy, especially in agriculture. In particular, solar coolers operating at the expense of solar energy, devices for providing rooms with hot water, solar cells that directly convert solar energy into electricity, chlorella cultivation devices based on the photosynthesis effect of sunlight, Sterling engine and autonomous solar energy

devices, solar water purifiers, seed irradiation concentrators, solar dryers, solar greenhouses and hotbeds, and similar devices can be included.

In Central Asia, in particular, in the southern regions of our Republic, there is an opportunity to heat greenhouses and hotbeds using sunlight and thereby save some fuel. For example, in areas with a geographical latitude of 39° , heated greenhouses consume 2,500-3,500 kg of conventional fuel (coal) or 110-120 m³ of gas per year per square meter of cultivated area.

So, using solar energy rationally, it is possible to save the same amount of fuel and reduce costs. If we take into account the amount of fuel consumed in ordinary dryers for drying fruit, the amount of fuel that can be saved increases even more. For example, in the PSK-20 dryer for drying fruits using steam, 200 kilograms of fuel are used to obtain one ton of dried product.

Therefore, here we will dwell in more detail on devices with heat accumulators that accumulate daytime solar energy.

Accumulation of solar energy is necessary not only for heliohomes, heliogreenhouses, but also for solar dryers.

As is known, solar dryers operate mainly during the day, and in order for solar dryers to dry day and night, it is necessary to place a heat accumulator in them.

Since the angle of inclination of the sun changes, the amount of energy falling on the earth's surface also changes. This, in turn, negatively affects the heating of solar cells and solar greenhouses. Secondly, since the day is long in the summer months, the period of energy falling from the sun is also long (the earth's surface is illuminated for 14 hours), and in the winter months it is short (the earth's surface is illuminated for 9 hours). Since the angle of inclination of the sun takes negative values in the winter months, its height also decreases. Therefore, in order to collect more energy, it is necessary to change the slope of the surfaces receiving it relative to the horizon, that is, to make it perpendicular to the sun's rays.

If it is not possible to change the inclination of the transparent surface of the solar device relative to the horizon for technical reasons, the inclination angles of the transparent surfaces of solar devices intended for operation in winter, for example, solar houses and solar greenhouses, are taken as $\alpha_1 = \phi + \delta$ in winter and $\alpha_2 = \phi - \delta$ in summer, depending on the geographical latitude of the location.

For example, at the geographical latitude of Karshi district $\phi=39$, $\alpha_1=32$ should be equal to $\alpha_2=32$ in the winter months and $\alpha_2=16$ in the summer months.

Devices designed to operate in the summer months, including water heaters, fruit dryers, and saltwater desalination plants, usually have small tilt angles.

A third disadvantage of using solar energy is its dependence on the weather. First of all, the amount of solar energy is affected by the clarity of the atmosphere.

Taking into account the absorption, scattering, and reflection of sunlight by the atmosphere, we take the fraction reaching the Earth's surface to be equal to Q_m . If we call the solar constant (the amount at the boundary of the atmosphere) Q_0 , the relationship between them is as follows:

$$Q_m = K^m Q_0$$

K - is the atmospheric transparency coefficient, m - is the mass of the atmosphere through which sunlight passes.

It is known that the mass of the atmosphere depends on the height of the sun above the horizon. For example, when $\alpha = 0$ (the sun is on the horizon), $m = 1$. When $\alpha = 30^\circ$, $m = 2$, and when $\alpha = 90^\circ$ (the sun is at the zenith), $m = 35.4$.

Depending on the geographical latitude, climate, location, etc., the Earth's surface is divided into cloudy and cloudless days in places.

From the point of view of solar technology, solar installations are most effective if they are installed in places where the number of sunny and semi-sunny days is 320 or more.

In the Central Asian Republics, including the southern regions of Uzbekistan, the number of such days is 315-330 days a year.

The fourth factor that negatively affects solar installations in the summer months is dusty winds.

First, when dense dust rises, the penetration of solar energy onto transparent surfaces decreases, and secondly, dust settles on the surface of the device. The dust layer prevents the passage of solar energy into the device. As a result, most of the time of the device's employees is spent cleaning the glass.

The fifth difficulty in using solar energy is its variability throughout the day, that is, although solar energy passes into solar installations during the day, the passage of energy stops towards evening.

The main ways to overcome the disadvantages associated with the change in solar energy throughout the year and day are:

- a) during the day, accumulating part of the solar energy in certain devices and using it at night;
- b) reducing the heat loss of the device to a minimum.

The period of solar energy accumulation is divided into 4: annual accumulation, seasonal accumulation, daily accumulation, and periodic accumulation.

Annual storage consists of collecting solar energy in hot weather and seasonal storage in cold weather, collecting solar energy in autumn (for 3 months) and using it in winter (for 3 months).

In daily storage, part of the solar energy that enters the device during the day is collected in the winter and spring months and used at night after sunset. In periodic storage, solar energy is collected for 3-4 days and used at night and on cloudy days,

and when the outside temperature rises, the battery must be disconnected and added again if necessary.

In solar heating systems, two types of batteries are mainly used: 1) heat-absorbing - heat-capable batteries, 2) chemical (isothermal) batteries.

In heat accumulators, soil, gravel, clay, water, concrete, sand, etc. are used as heat-accumulating materials.

Table 5.8.1 lists some constants (quantities) of materials that can be used in heat accumulators.

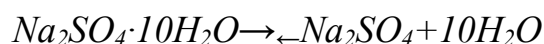
Table-5.8.1

Substances	λ , W/t °C	C, Kj/Kg °C	α , kg/m ³	α 10 ² m ³ /s
Water	1,2	1	2500	0,2
Gravel	0,361	0,22	1840	0,00136
Concrete	1,280	1,20	2000	0,00494
Baked brick	0,36	0,26	1200	0,00149
Soil	2,6	0,21	1900	0,46
Basalt	1,8	0,20	3850	0,00492
Sand	1,130	2,09	1650	0,00492

As can be seen from the table, the most suitable material for heat-storage batteries is water, since the heat capacity of water is greater than that of other substances.

According to scientists, the diameter of the stones used in stone batteries should be 2.5-3.5 cm. Otherwise, the heat-receiving surface decreases, the hydraulic resistance in the batteries increases, and the power of the fans that drive the air will have to be increased.

In chemical batteries, the accumulation and release of heat is based on a change in the aggregate state of the accumulating substances. As is known, since the change in the aggregate state of substances occurs at a constant temperature, such batteries are also called “isothermal batteries”. For example, sodium sulfate crystal hydrate accumulates heat as a result of a chemical reaction:



In this case, its temperature is maintained at 32.3 °C and slowly heats up, losing the water content. Table 5.8.2 presents the latent heat of some inorganic salt hydrates that can be used as storage (accumulating) substances. A comparison of heat-storage batteries with chemical batteries showed that in them it is necessary to take different volumes of material to store the same amount of heat, and vice versa, 1 m³ of material stores different amounts of energy. For example: 1 m³ of water at 303 K stores 126,000 kJ/m³; 357,000 kJ is stored in sodium sulfate decahydrate. This is 62.8 times more energy than is stored in water. It should be noted that in batteries built for heating solar

cells and solar greenhouses, hot, humid air is mainly used as a heat carrier, and in some cases water.

Table-5.8.2

Mixture	Melting temperature, T, K	Heat achieved, kJ/kg
$CaCl \cdot 6H_2O$	302-303	214,3
$Na_2CO_3 \cdot 10H_2O$	305-319	260,8
$Na_2CO_3 \cdot 12H_2O$	309	266,5
$Ca(NO_3)_2 \cdot 4H_2O$	313-315	200
$Na_2SO_4 \cdot 10H_2O$	305	243,5
$Na_2SO_4 \cdot 5H_2O$	322-325	200

Thermal accumulators designed for heating houses can also be widely used in helioparniks and heliogreenhouses. Solar energy can also be stored in electric accumulators. Electric accumulators are placed on the territory of solar power plants. The capacity of electric accumulators should be such that the energy received from the sun during the day can fully satisfy the operation of the consumer of electric energy at night. To do this, firstly, about half of the electricity generated by the solar power plant during the day should be used to charge the accumulators. Secondly, electric accumulators can be used in vehicles, for example, heliocars, helioplanes, helioboats, etc. In these cases, part of the electric energy received from photovoltaic cells during the day is used to charge the accumulator, and at night the energy received from this accumulator is used to drive the engines.

In the future, another convenient way to accumulate solar energy is to collect water in special basins located at a certain height. For this, water is pumped into the upper basin using special pumps at the expense of part of the energy generated by the solar power plant, and in the evening the generator starts generating electricity.

The next problem is related to the storage of energy accumulated in solar installations during the day until the morning. This measure is mainly carried out by reducing the heat loss of the device. The calculation of heat loss in solar installations has been thoroughly studied by A.M. Shklover, K.F. Fokin and others. For this, it is necessary to carry out heat-technical calculations for solar houses, solar greenhouses, solar generators and other devices, and to ensure low heat flow losses for all cladding walls.

Experiments have shown that if the transparent surfaces of a solar greenhouse and a solar dryer facing south are made of a single layer of glass or polyethylene film, the device will cool down quickly. Therefore, the north side of such solar devices is made of several layers of heat-insulating layer, and the transparent surfaces facing

south are made of two layers of glass or polyethylene film with an air gap of 5-10 cm between them.

The total amount of heat lost from solar greenhouses-dryers is expressed as follows:

$$\sum Q_y = Q_m + Q_n + Q_\varphi + Q_t$$

where, $\sum Q_y$ - total heat loss; Q_m - amount of heat lost through the glass surface; Q_n - the amount of heat lost through the opaque surface of a greenhouse; Q_φ - amount of heat lost through the base; Q_t - amount of heat lost through cracks.

In such cases, the heat transfer coefficient of the total surfaces $K=1.2-1.9 \text{ W/m}^2\text{K}$ will achieve a minimum amount of heat loss.

Experiments have shown that when the layer between the transparent surfaces of solar greenhouses is a polyethylene film with a thickness of 4 cm and 6 cm, normal humidity regimes are achieved, thus, heat accumulators play an important role for low-temperature solar structures, in particular for solar houses, solar greenhouses, and solar greenhouses. Calculation of the energy accumulated in a solar greenhouse per day:

$$\sum Q_{ak} + Q_{tup} + Q_{ekin} \quad (5.8.1)$$

where, Q_{ac} - The amount of energy stored in a rock-solid heat accumulator. Q_{earth} - the amount of energy accumulated in the earth (soil). Q_{plant} - the amount of energy accumulated in crops (plants).

We consider the amount of heat Q received by the accumulator during forced air circulation. Heat exchange between the air and the accumulator occurs in its channels. During the day, the air heated to a temperature of t , moves along the accumulator channels by natural gravity and gives off its heat to their walls, gradually cooling to a temperature of t_2 , and at night the reverse process occurs. The amount of heat of the air heated by the accumulator is equal to:

$$Q_{ak} = XF_k(t_1 - t) \quad (5.8.2)$$

where, X - Heat transfer coefficient by period of shaking, t_1 - The average temperature of the heated air in the accumulator channel during the period t_n , F_k - internal surface area of the canal; t' - average temperature of the channel walls.

Full battery life

$$T = \tau_H - \tau_{ox} \quad (5.8.3)$$

where, τ_H - battery charging cycle, τ_{ox} - battery discharge period. To determine the coefficient of X , we use Rummel's formula: ζ - distance between adjacent battery channels.

Ψ - The Gerber function is expressed as follows::

$$\Psi(F_o) = \sqrt{\frac{F_o}{2\pi}} \sqrt{\frac{ch \sqrt{\frac{4\pi}{F_o}} \cos \sqrt{\frac{4\pi}{F_o}}}{ch \sqrt{\frac{4\pi}{F_o}} + \cos \sqrt{\frac{4\pi}{F_o}}}} \quad (5.8.4)$$

where, F_o - Fourier and Rovna criterion. $F_o = \frac{\alpha \tau_0}{x^2}$ is calculated from the coefficient of utilization of the function walls and 0,92-0,24 appears on the border (around): For soil and concrete it is 0.68. ζ - Dimensionless coefficient calculated using Rummel's formula.

$$Q_{max} - Q_{min} = \zeta(Q_H - Q_{ox}) \quad (5.8.5)$$

where, $Q_H - Q_{ox}$ and τ_H - The average temperature of the surface of the channel walls in the accumulator during period τ_{ox} . $Q_{max} - Q_{min}$ - maximum and minimum temperatures of the channel walls at the end of the heating and cooling periods.

The total amount of heat involved in heat exchange between the battery walls and the air during the charging and discharging periods through the total heating density is expressed by the following formula:

$$a) \text{ during charging } Q = \alpha \tau_H F_e (t - Q) \quad (5.8.6)$$

$$b) \text{ during the blackout } Q = \alpha \tau_{ox} (Q - t) \quad (5.8.7)$$

where, F_e - surface of the channel walls. The heat transfer coefficients α and α' included in this formula can be determined using the following theorem (theory).

To calculate the heat transfer by air to the surface of the heat accumulator, we use the Bio $B_i = \frac{\alpha d}{\lambda}$ and Pekle $P_e = \frac{\sigma d}{a}$ criteria. Here, d is the channel diameter, α - v is the air velocity in ms, and α is the temperature transfer coefficient.

The thermal conductivity coefficient is determined by the following formula:

$$\alpha = \frac{\lambda}{C_p \gamma_e} 9 \quad (5.8.8)$$

where, C_p - heat capacity of air at constant pressure. γ_e - specific gravity of air.

We obtain a functional relationship connecting the two aforementioned criteria.

$$B_i = f(P_e) \quad (5.8.9)$$

In this,

$$\gamma = \frac{\lambda}{d} f\left(\frac{\sigma d}{Q}\right) \quad (5.8.10)$$

According to Nusselt research:

$$\gamma = c\lambda^{0,4}d^{-0,21}(\sigma c\gamma)^{0,79}$$

So,

$$f = \left(\frac{\sigma d}{\rho}\right) = C \left(\frac{\sigma d}{\rho}\right)^{0,79} \quad (5.8.11)$$

where, C – the coefficient is about 18.

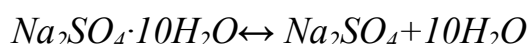
The pressure and moist air temperature of X via C_l , γ magnitudes were brought to the following state by heating the Shaku.

$$\gamma = \frac{3\sigma^{0,8}}{4\sqrt{d_m}} (kcal/m^2 \text{ soat } ^\circ K) \quad (5.8.12)$$

where, d_m – equivalent diameter of the channel.

5.9. Chemical heat accumulator

In chemical batteries, the accumulation and release of heat is based on the change in the aggregate state of the accumulating substances. As is known, since the change in the aggregate state of substances occurs at a constant temperature, such batteries are called isothermal batteries. For example, sodium sulfate crystals accumulate heat as a result of a chemical reaction.



In this case, its temperature is maintained at 32.3 °C and it slowly heats up, losing the water content. Table 5.9.1 lists the latent heat of hydrates of some inorganic salts that can be used as storage materials.

Table-5.9.1

Mixture	Melting temperature T, K	Heat achieved, q_k kJ/kg
$CaCl_2 \cdot 6H_2O$	302-303	174,3
$NaCO_4 \cdot 10H_2O$	305-319	260,8
$NaPO_4 \cdot 12H_2O$	309	266,5
$Ca(NO_3)_2 \cdot 4H_2O$	313-315	200
$Na_2SO_4 \cdot 10H_2O$	305	243,5
$Na_2S_2O_3 \cdot 5H_2O$	322-325	200

A comparison of heat capacity accumulators with chemical accumulators showed that in them it is necessary to take different volumes of material to store the same amount of heat, and vice versa, 1 m^3 of material stores different amounts of energy. For example, 1 m^3 of water at 303 K stores 120,000 kJ/m^3 , which is 2.8 times more heat than sodium sulfate decahydrate.

It should be noted that in batteries built for heating solar houses and solar greenhouses, mainly hot dry air is used as a heat carrier, in some solar greenhouses and solar houses, hot humid air, and in some cases water.

Solar energy can also be stored in an electric battery. The electric battery is placed in the territory of solar power plants. The capacity of the electric battery should be such that the electricity received from the sun during the day can fully satisfy the consumer at night. To do this, firstly, about half of the electricity generated by the solar power plant during the day should be used to charge the batteries. In recent years, two-component composite chemical batteries made from salt hydrates have been widely used.

Table 5.9.2 below presents the compositional structure and thermal physical characteristics of some of the composite salt hydrates used.

Table-5.9.2

Inorganic compounds and compositions for use as heat accumulators

Composition, relative mass %	T_m °C	ΔH_m /J/g	ρ , g/sm ³
<i>CaBr₂·6H₂O (55) CaCl₂·6H₂O</i>	14,7	140	-
<i>Ca(NO₃)₂·6H₂O(51ch55)/ LiNO₃·3H₂O</i>	16,5	250	-
<i>LiNO₂·3H₂O (52ch55)/ LiNO₂·3H₂O</i>	17,2	220	-
<i>CO(NO₃)₂·6H₂O (41,5ch52,5)/4H₂O(5,8ch9,7)/ LiNO₃·3N₂O</i>	22,3	265	-
<i>LiNO₂·3H₂O (55ch55)/ Ni(NO₃)₂·6H₂O</i>	24,2	230	-
<i>Ca(NO₃)₂·4H₂O(45)Zn/(NO₃)₂·H₂O</i>	25	130	1,93
<i>Ca(NO₃)₂·4H₂O(55ch55)/CO(NO₃)₂·6H₂O</i>	29,2	130	-
<i>Ca(NO₃)₂·4H₂O(55ch45)CO(NO₃)₂·6H₂O</i>	30	135	1,67

However, it is possible to select scientific directions for the use of such complex inorganic compounds and compositions in heliocomplexes and to determine the physicochemical nature of the composition specified therein and recommend their use. Determining the optimal variant of a heat accumulator made of such chemical compounds, their nonlinear dynamic processes and their functional characteristics, and achieving effective use as a heat accumulator are problematic issues.

When using materials, bodies, chemical element compounds and composite materials used as heat accumulators in heliogreenhouses, heliohomes and other low-potential solar devices, it is necessary to determine their heat storage capacity, heat capacity, optimal placement dimensions, mass and temperature change processes in spatial transitions. For solar greenhouses, it is important to prepare accumulators of optimal sizes and capacities from wet soil filled with polyethylene films, stones treated with black fuel oil and bitumen, a mixture of sand and metal shavings, barrels of various sizes and plastic bitumen filled with water, and used transformer oils, and

calculate the amount of heat accumulation in them. For solar houses, various building materials are currently used as heat accumulators. For solar houses, it is economically effective to choose building materials from heat-accumulating bricks, compounds of inorganic chemical elements, as well as building materials that are compatible with the thermal physical characteristics, heat capacity, and spatial transitions of objects that can be used as heat accumulators. Japanese scientists have put forward the idea that they can use the super technology (heat generation consisting of a combination of chemical elements) that they invented at any time and anywhere. This technology, based on the production and accumulation of heat from chemical elements, can be used for heating houses and other purposes. The results of studies conducted by many authors have shown that in addition to the use of daily heat storage batteries and chemical batteries in solar installations, it is advisable to use an electric heater to provide additional heat on chronically cloudy and cold days. The use of a chemical battery is divided into two stages. In the first stage, the part of the volume of excess heat of chemical liquid salts between the deposition temperature t_{dep} and the solidification temperature t_{solid} is used. As a result, the volume chemical heat is used until the amount of heat is equalized over the entire cross section. In this process, the shape and dimensions of the chemically crystallized salt do not affect the heat transfer during accumulation, only the volume V and the cooling surface of the chemical body depend on F and $d\tau$ on time, and the amount of heat released during the spatial transition is determined by the following formula:

$$dQ = JF(t - t_c)dt \quad (5.9.1)$$

where, J - coefficient of heat transfer from the surface of a chemical element to a solar device, $KKal/m^2 \cdot \text{hour} \cdot K$; $\Delta t = t - t_c$ - temperature difference in a chemical ice solution, $^{\circ}C$. Therefore, during the melting process of a salt composed of a combination of chemical elements, its temperature decreases, and as a result, the change in the volumetric heat storage content of the system over the time interval $d\tau$ is equal to:

$$dq = -V\gamma c dt \quad (5.9.2)$$

and the amount of heat transferred from the chemical heat accumulator to the solar devices during a time interval τ is calculated by the formula:

$$\tau = \frac{V}{F} \frac{\gamma c}{L} \ln \frac{t}{t_{kuy}} \quad (5.9.3)$$

In the second stage, the chemical element compounds transfer the heat accumulated in them during solidification to the solar devices. As a result, the air temperature inside the solar device is moderated due to the solar energy absorbed and accumulated by the chemical element compounds and salts.

CHAPTER VI. SOLAR THERMAL DEVICES

6.1. Solar water purifiers. Types and various designs of solar water purifiers

As a result of scientific and technological progress, the problem of drinking water is being put forward as the most urgent problem among the global problems facing humanity, such as energy, food and environmental problems. In the future, organic fuels (primarily oil and gas) can be replaced by nuclear energy or non-traditional, renewable sources of solar, hydraulic, wind, geothermal, wave energy. All this allows experts to look optimistically at the future and distant future of energy production.

Desalination of brackish water is carried out by removing various salts contained in water or, conversely, perfectly pure water molecules from the composition of salt water. Water desalination devices are divided into two groups. In group I devices, the aggregate state of the water being desalinated is changed, while in group II devices, it is not changed. Group I includes thermal and freezing desalination devices, and group II includes chemical, membrane, extraction, adsorption and biological desalination devices. Since the number of salt molecules in brine is much smaller (30–100 times) than the number of water molecules, the method of extracting salts from water seems theoretically economical, but in the devices created and tested so far, this method has not been technologically and economically justified.

The Arab scientist Mas'udi (late 9th century–957 AD) confirmed Aristotle's idea of the water cycle in the atmosphere through an experiment. He conducted a very simple and reliable experiment: he evaporated sea water and obtained pure condensate from this vapor. This experiment was the first experiment on desalination of salt water.

The distillation method is considered the most effective method for desalination of salt water. Of the total amount of water that is desalinated in the world, 96% is produced by distillation, 2.9% by electrodialysis, 1% by hyperfiltration, and 0.1% by other types of devices (freezing, ion exchange, etc.).

The contribution of solar water desalination to this is currently 0.05% (approximately 7,000 m³/day). Distillation is associated with a change in the aggregate state of water. When water boils, its free molecules, due to heat and vibrational motion, gain enough energy to overcome the forces of molecular bonds and fly off the surface of the water. The evaporation surface of the water acts as a filter through which water molecules can pass and salt ions cannot.

The resulting steam contains practically no salt, so when it condenses, pure water is formed. This method of distillation under the influence of heat is called distillation. The optional distillation device consists of an evaporator and a condenser, which are directly connected to each other.

Currently, several large distillation plants operating on fuel and nuclear energy have been built in different regions of the world. Among them, dozens of complexes can be cited, such as Hong Kong's Xiangnan ($180,000 \text{ m}^3/\text{day}$), Shevchenko ($130,000 \text{ m}^3/\text{day}$), Japan's Oita ($10,000 \text{ m}^3/\text{day}$), Sardinia ($36,000 \text{ m}^3/\text{day}$), built by Italian specialists.

Salt water can also be desalinated by distillation using solar energy. This method is especially useful in deserts and deserts where electricity is not available, and where the population is sparse and scattered. Solar water desalination plants can also be used individually in more densely populated areas, depending on the need.

Solar water heaters operating on the “hot box” principle are characterized by their simplicity of construction and ease of maintenance. Simply put, devices of this type are created by installing a container with salt water instead of a light absorber inside the “hot box”. Although the solar water heaters created in the past and modern ones operate on the same principle, they differ significantly in their design solutions and efficiency. Sunlight passing through a transparent coating (glass or film) installed obliquely at the top of the device heats the salt water or wetted fabric in the container installed at the bottom. The evaporation of the heated water is accelerated. Due to the difference in temperatures at the bottom and top of the device, water vapor rises with the convective flow and hits the lower surface of the transparent coating, where it condenses, since the temperature on this surface is lower than the temperature of the saturated vapor-air mixture inside the chamber. The water formed by the condensation of vapors slides along the surface of the coating and falls into a special collecting trough, from which it is directed to the water collection tank. The first solar water treatment device operating on the “hot box” principle was built in 1872 in Las Salinas, Chile, by Charles Wilson and Harding. The device has the form of a black wooden box covered with asphalt at the bottom, on which a glass is installed at an angle. It is equipped with devices for pumping salt water and extracting fresh water. According to available data, the maximum productivity of this device, which occupies a surface area of 4760 m^2 , was 22,700 liters of fresh water per day ($4.8 \text{ l per } 1 \text{ m}^2$ of surface area) and has served for 36 years. Since then, much research and practical work has been carried out to improve this type of construction. In this field, the contributions of V.B. Weinberg, V.V. Weinberg, K.T. Trofimov, M. Telkes, V.A. Baum, G.Yo. Umarov, A.N. Tekuchev, R.B. Bayramov, B.M. Ochilov and other scientists are particularly significant.

The solar water heater with a wettable surface, designed by K.T. Trofimov, is oriented south and inclined at an angle of $20\text{--}250^\circ$ to the horizon. A black cloth is laid on the bottom of the device, which is continuously wetted with salt water flowing from the tank above (Figure 6.1.1). The efficiency of inclined solar water heaters with a wettable surface is much higher than that of similar devices with a horizontal location (up to 6 l/m^2 in summer). This is made possible by the maximum absorption of solar

radiation on the wettable surface, the small distance between the evaporation and condensation surfaces, and the low heat loss through the side walls, due to the device being positioned perpendicular to the sun's rays.

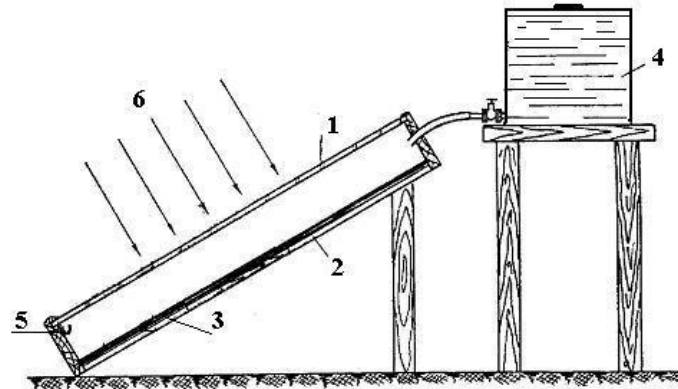


Figure-6.1.1. Scheme of K.T. Trofimov's solar water purifier: 1–glass; 2–tin; 3–wetting surface covered with black cloth; 4–container with salt water; 5–condensate collection device; 6–sunlight.

Less material and labor are spent on the manufacture of the device, which increases the economic efficiency of the device. Such devices are especially convenient for individual consumers. At the same time, it has a number of disadvantages related to the rapid wear of the fabric, the difficulty of controlling the amount of salt water flowing from the tank in accordance with the operation of the device and ensuring uniform wetting of the fabric, and the removal of residual salts from the fabric. It is important that the wetting fabric always remains black. Synthetic fabrics containing 80–90 percent carbon retain their black color for a long time even when exposed to sunlight.

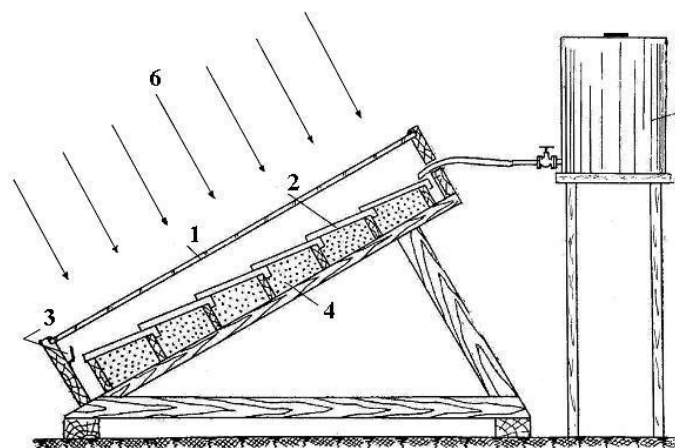


Figure-6.1.2. Scheme of A.N. Tekuchev's solar water purifier: 1–glass; 2–ceramic plate; 3–condensate collector; 4–thermal insulation; 5–salt water tank; 6–sunlight.

A.N. Tekuchev's solar water purifier is also placed at an angle to the horizon, facing south (Figure 6.1.2). At the bottom of the device, ceramic plates with an insulated bottom are stacked, tilted from top to bottom. When salt water is pumped

from the upper tank, the ceramic plate is wetted, and the water contained in it evaporates due to the energy of sunlight and condenses on the lower surface of the glass. Experiments conducted in Samarkand in 1933 showed that the average efficiency of the device is about 3 l/m^2 . A similar solar water purifier was created and put into operation in Florida in 1960 by M. Telkes. The device cost \$ 13 per sq. m, the highest efficiency was 7 l/m^2 , and the service life was not less than 10 years.

In the 1950s, a special program was developed in California to desalinate salt water. Based on the requirements of this program, M. Telkes designed a solar water desalination plant similar to the C. Wilson plant created in Chile (1872). M. Telkes' solar water desalination plant consisted of 5 wooden troughs connected by pipes, each with an evaporation surface of 18.2 m^2 ($15.2 \text{ m} \times 1.2 \text{ m}$). This plant differed from Wilson's single-slope plant, which was placed directly on the ground, in that it had two slopes and was installed above the ground using special supports. The troughs of this plant, with a total area of 91 m^2 , were filled with 10–30 mm of salt water. The highest daily efficiency of the plant was 375 l or 4.1 l/m^2 .

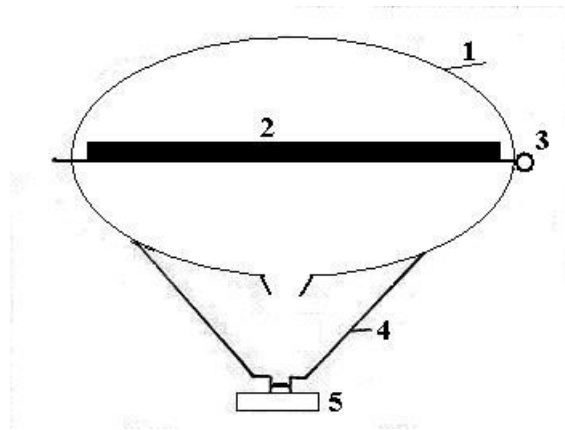


Figure-6.1.3. M. Telkes' film-type solar water heater: 1–transparent cover; 2–black porous evaporator; 3–connection; 4–container for fresh water; 5–device for pouring fresh water.

Another interesting design is the closed basin type solar water desalination plant, the first of which was created in 1958 at the research station established by the US Department of the Interior's Salt Water Use Administration in Florida, where various water desalination devices were tested (Figure 6.1.4). Concrete, glass, asphalt, and cement were mainly used to construct the plant. Pre-prepared asphalt was laid directly on the ground, and then construction work was carried out on top. This plant, with an evaporation surface of 230 m^2 , consists of six sections. Seawater is poured into the basin with a thickness of 250–300 mm. The upper part of the plant is covered with double-sloped glass covers using special columns. The resulting distillate is collected through plastic pipes. When the daily amount of solar radiation is around 5000 kcal/m^2 , the relative efficiency of the device is on average 3 l/m^2 , f.i.k. 26–35%. The service life of the device is set at 50 years. A distinctive feature of the multi-water solar water

purifier is that it is an inertial device, in which water distillation continues both at night and on cloudy days following sunny days.

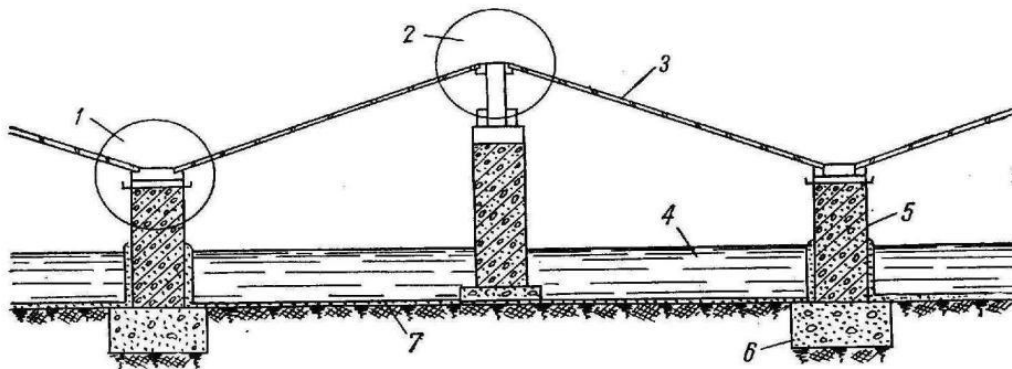


Figure-6.1.4. Schematic of a closed basin-type solar water heater: 1–devices for collecting distilled water; 2–device with the upper part of the glass installed; 3–glass cover; 4–salt water; 5–concrete block; 6–concrete support; 7–pre-prepared asphalt layer.

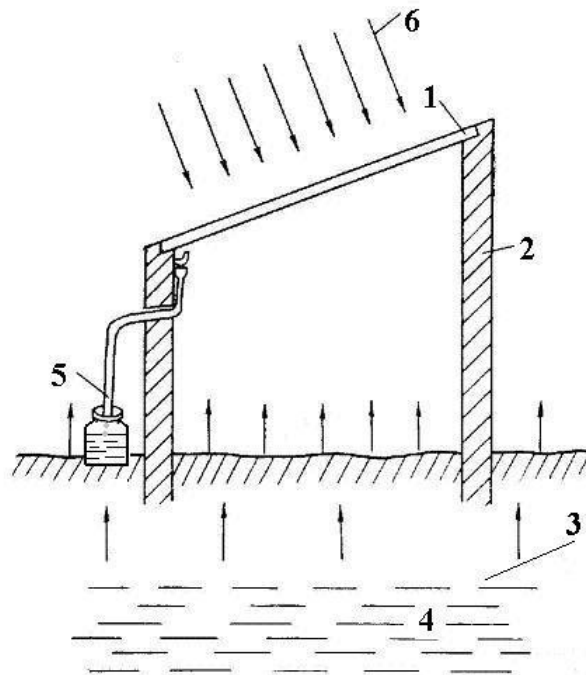


Figure-6.1.5. Heliodevice for extracting fresh water from soil: 1–transparent coating; 2–casing; 3–capillaries; 4–moist soil; 5–device for collecting distilled water; 6–sunlight.

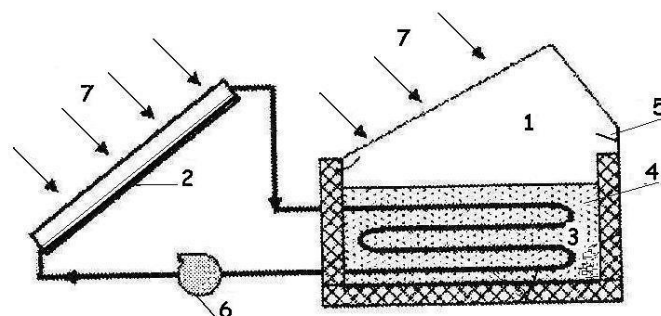


Figure-6.1.6. Active solar water chiller device: 1–greenhouse-type solar water chiller; 2–solar collector; 3–heat exchanger; 4–brine; 5–condensate; 6–pump.

The device proposed by M. Kobasi allows you to get fresh water from the soil (Fig. 6.1.5). Experiments near Tokyo and in Pakistan have shown that the efficiency of the device is $1.3 \text{ l} / \text{m}^2 \text{ day}$. Such a device is especially useful in dry deserts and deserts. If a box with a tilted glass (film) on top is placed slightly above the ground, the salt water contained in the soil rises up and evaporates, filling the closed space and condensing on the lower surface of the glass (film). On extremely hot days, the temperature of the glass remains higher than the temperature of the earth's surface, and the device works only in the evening. The fresh water produced in this way is extremely clean and is 10 times less radioactive than water from open reservoirs.

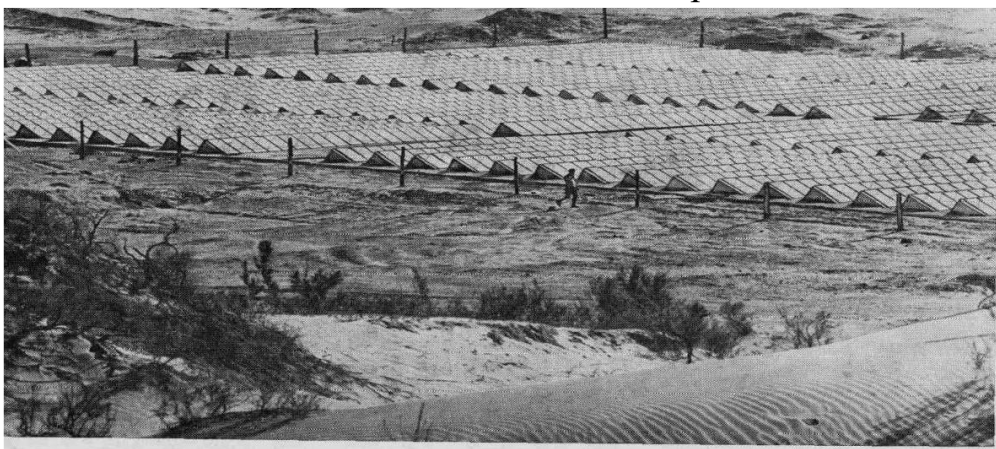


Figure-6.1.7. Greenhouse-type solar water treatment plant (Karaqum, “Baharden”).

Based on research conducted at the Institute of Physics and Technology of the Academy of Sciences of Turkmenistan under the leadership of R.B. Bayramov, the Giprovodkhoz Design Institute developed a project for a greenhouse-type solar water desalination plant with a useful surface area of $2,400 \text{ m}^2$ in 1962. The first stage of the plant with an area of 600 m^2 was built and tested in 1968 at the Bakharden state farm in the Karakum desert.

6.2. Heat and mass transfer processes in solar water heaters

The main part of the inclined-step solar water heater is a base with a glass top and steps at the bottom for storing salt water (Figure-6.2.1). The base can be made of stainless steel, polymer materials, or reinforced concrete, depending on the conditions in which the device will be used. The height of the steps is selected based on the condition that the water evaporation surface is large. Depending on the length of the base (l), the number of steps (n), the angle of inclination of the device (α) and the thickness of the step (δ), the height of the steps (h) is determined by the following expression:

$$h = \left(\frac{l}{n} - \delta \right) \operatorname{tg} \alpha$$

Figure 6.2.1 shows a scheme of a stationary solar water heater with a reinforced concrete base. Regardless of the material from which the devices are made, they all work on the same principle: the transparent cover of the device - the glass, most of the sunlight falling on it, passes through it and is absorbed by the steps and the salt water in them, where it is converted into thermal energy. In turn, infrared rays emanating from the heated surface of the device are directed towards the glass, but since the glass is opaque to these rays, they fall back to the bottom of the device, contributing to the formation of a “greenhouse effect” in the device. The vapors emerging from the surface of the heated water rise due to convection and hit the lower surface of the glass, condense, collect in a vat as distilled water and are removed from the device through a special pipe. In our country, the device is installed at an angle of 30-35° degrees to the horizon, facing south. This will allow the device to receive the maximum amount of solar energy throughout the year. The direct solar radiation energy entering the device through the unit area of the transparent coating of the water purifier device is determined from the following expression:

$$Q_{kir} = B_{kir} \cdot I \quad (6.2.1)$$

where, $B_{kir} = B_{sh} B_r B_{ch}$ - solar radiation penetration coefficient; V_{sh} – solar radiation transmittance of glass; V_r – coefficient that takes into account the reduction in solar radiation due to the presence of frames; V_{ch} – The coefficient of transmission of solar radiation through dust and debris in glass. The average values of these coefficients are given in Table 6.2.1.

The main part of the radiation flux falling on the working surface of the device is absorbed by the bottom of the device and water, depending on the optical properties of the materials, and is converted into thermal energy, the amount of which can be determined from the following equation:

$$Q_1 = I_0 (1 - \rho)^2 \cdot (1 - A_{sh})^2 \cdot (1 - R_s) \cdot A_{tub} \quad (6.1.2)$$

where, $I_0 = I_{to'g'ri} + k I_{tarq.}$; $I_{to'g'ri}$ - direct solar radiation intensity, W/m²·hour; $I_{tarq.}$ – diffuse solar radiation intensity, W/m² ·hour; ρ and A_{sh} – the reflection and absorption coefficients of glass, respectively; R_s – water's light reflectance; A_{tub} – light absorption coefficient of the bottom of the device; k – coefficient of penetration of scattered radiation into the device;

Table-6.2.1

Average values of the coefficients B_{sh} , B_r and B_{ch}

Coefficients	0°	20°	30°	40°	50°	60°	70°	80°
B_{sh}	0,931	0,928	0,913	0,891	0,860	0,822	0,720	0,512
B_r	0,890	0,835	0,820	0,800	0,760	0,673	0,585	0,443
B_{ch}	0,940 – 0,960							

Since the temperature of the bottom of the device and the brine is higher than that of the rest of the parts, heat is transferred to the environment in different ways, depending on the properties of the materials and the temperature difference in different parts.

The thermal energy generated by the absorption of sunlight in the bottom of the device and in the water is mainly spent on water evaporation and heat loss.

The amount of heat transferred to the environment through the bottom of the device is determined as follows, depending on the temperature difference between the water and the environment ($t_{suv}-t_{tashq}$), the heat transfer coefficient of the bottom (λ), the thickness (δ_{bottom}) and the heat transfer coefficient to the air (α):

$$Q_2 = \frac{1}{\frac{1}{\alpha} + \frac{\delta_{bottom}}{\lambda}} (t_{suv} - t_{tashq}) \quad (6.1.3)$$

The amount of heat transferred from the water to the bottle is:

$$Q_3 = [C] \left[\left(\frac{t_{suv} + 273}{100} \right)^4 - \left(\frac{t_{1sh} + 273}{100} \right)^4 \right] \quad (6.1.4)$$

where,

$$[C] = \frac{1}{\frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3}}$$

is the radiation coefficient of the water-glass system; C_1 , C_2 and C_3 – radiation coefficients of water, glass, and a black body; t_{1sh} – The temperature of the bottom surface of the glass.

Due to the evaporation of water, circulating flows of the steam-air mixture occur in the chamber of the device, depending on the temperature difference between the water and the glass (the pressures in the layers near the water and the glass). During the movement of the steam-air mixture in the chamber, some heat is transferred to the glass. This amount of heat depends on the temperature difference between the evaporating water and the glass and the heat transfer coefficient of the mixture to the glass ($\alpha_{kond.}$) and is determined as follows:

$$Q_4 = \alpha_{kond.} (t_{suv} - t_{1sh}) \quad (6.1.5)$$

The amount of heat passing through the glass is:

$$Q_5 = (t_{1sh} - t_{2sh}) \frac{\lambda_{sh}}{\delta_{sh}} \quad (6.1.6)$$

where, t_{2sh} – temperature of the outer surface of the glass; δ_{sh} , λ_{sh} – glass thickness and heat transfer coefficient.

The heat transferred from the glass to the environment by radiation and convection is calculated as follows:

$$Q_6 = C_{sh} \left[\left(\frac{t_{2sh} + 273}{100} \right)^4 - \left(\frac{t_{tashq} + 273}{100} \right)^4 \right] + \alpha(t_{2sh} - t_{tashq}) \quad (6.1.7)$$

Thus, the amount of heat absorbed by the bottom of the device and the water (Q_1) can be written as the sum of the heat losses through its bottom (Q_2), the heat radiated from the glass (Q_3), and the heat released during condensation of the vapor (Q_4):

$$Q_1 = Q_2 + Q_3 + Q_4 \quad (6.1.8)$$

On the other hand,

$$Q_6 = Q_3 + Q_4 + I_0(1 - \rho)A_{sh} \quad (6.1.9)$$

The useful working coefficient and efficiency of the device are determined as follows:

$$\eta = \frac{Q_1 - Q_2 - Q_3 - Q_{konv}}{I_0} \quad (6.1.10)$$

or,

$$\eta = \frac{Q_f}{I_0} = \frac{rG}{I_0} \quad (6.1.11)$$

$$G = \int_{\tau_1}^{\tau_2} \frac{1}{r} dQ d\tau \quad (6.1.12)$$

where, r – heat of vaporization, kcal/kg, J/kg; τ – device lifetime; Q_{konv} . – The heat released by the steam-air mixture as it cools from t_{suv} to t_{lsh} due to convection.

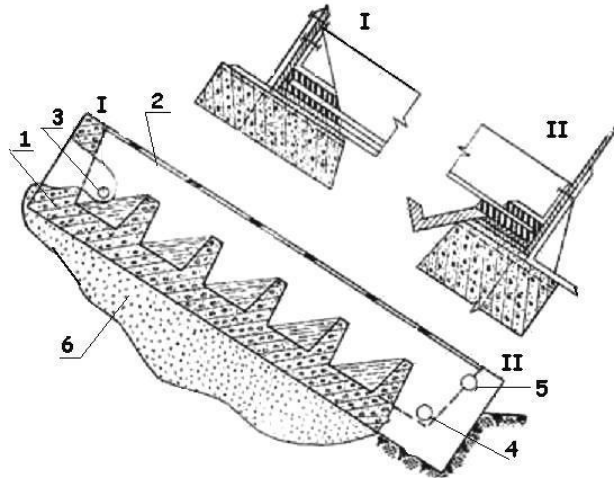


Figure-6.2.1. Scheme of a sloped solar water desalination device: 1–reinforced concrete base of the device; 2–transparent cover (glass); 3–pipe for pouring brine; 4– pipe for draining excess brine; 5–pipe for distilled water; 6–dry sand (thermal insulation); I, II–installation of a glass and a distilled water collection device on top of the device.

A comprehensive study and analysis of the heat and mass transfer processes occurring in the device, as well as practical research, leads to the conclusion that on average 45–50 percent of the solar energy received by the device is spent on

evaporating water (useful work), and the rest is spent on heating the device and various heat losses.

6.3. The working process of greenhouse-type solar water chillers. Evaluation of the thermal technical characteristics of a solar water chiller

The simplest type of solar chillers is a single-pitched greenhouse-like chiller of the “hot box” type, the structure of which is shown in (Figure 6.3.1). Salt water is poured into a special flat container-boiler with the inside of the “hot box” facing up through a funnel 1) 5). In a single-pitched chiller, the glass surface 3) should be directed to the south and poured at a certain angle to the horizon, taking into account the maximum incidence of solar radiation throughout the season. For the territory of Central Asia, the angle relative to the horizon is 25-30 degrees, and for the city of Karshi, 38 degrees.

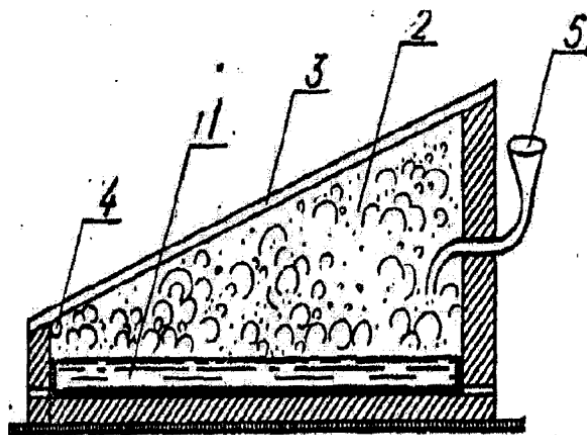


Figure-6.3.1. Structural diagram of a greenhouse-type humidifier.

In 1970, based on research conducted at the Institute of Physics and Technology of the Academy of Sciences of Uzbekistan and the Bukhara State Pedagogical Institute, and taking into account the advantages and disadvantages of the “Baxarden” device, a greenhouse-type solar water desalination complex with an evaporation surface of 608 m² and mainly made of reinforced concrete and glass was created by B.M. Ochilov in the “Boymurodkuduk” settlement located in the central part of the Kyzylkum desert (Figure-6.3.3). The complex consists of 39 separate desalination devices, each with an evaporation surface of 15.6 m² (12 m x 1.3 m), arranged in two rows on an area surrounded by a metal mesh wall with a corridor of 0.3 m between them. The distance between the two rows is 1.85 m.

The concrete-covered area where the devices are located is prepared with a slope suitable for collecting rainwater in a separate pond. The device has a double-slope glass cover (Figure-6.3.2), with the front glass set at an angle of 30° and the rear glass at 60° to the horizon. All devices are filled with 7 cm of salt water (salinity 14 g/liter) through pipes connected to a reservoir that is pumped from a 29 m deep well using a belt pump.

The distilled water formed in the front and rear glasses of the device is collected in inclined channels and flows through pipes into the distilled water pool. The devices are periodically (1-2 times a year) cleaned of residual salts by draining water. Each device operates independently of the others, and malfunctions in one do not negatively affect the operation of the other. However, these malfunctions lead to a decrease in the overall efficiency of the complex.

Test work was carried out in the complex in 1970-1972. It was found that the device can produce up to 900-1000 liters of clean water per year per square meter. The average daily productivity of the device during the year was 2.5 l/m^2 , increasing to an average of 3.5 l/m^2 in summer and dropping to an average of $0.3\text{-}0.5 \text{ l/m}^2$ in winter. The test results show that this type of device is an effective method for providing clean drinking water to settlements located 10-15 km from clean water in deserts and deserts, but they also show that there is a need for structural improvements: the large size of the device requires a lot of material and manual labor to build it at the installation site; the horizontal location of the device and the double-slope design of the transparent coating lead to an unprofitable increase in the condensation level relative to the level of saline water (evaporation level) where solar radiation falls, and to an increase in heat loss; the large-surface glazed part located above the ground is subject to great mechanical stress under the influence of winds and storms that occur in the desert and often requires repair. All this negatively affects the high efficiency and effectiveness of the device.

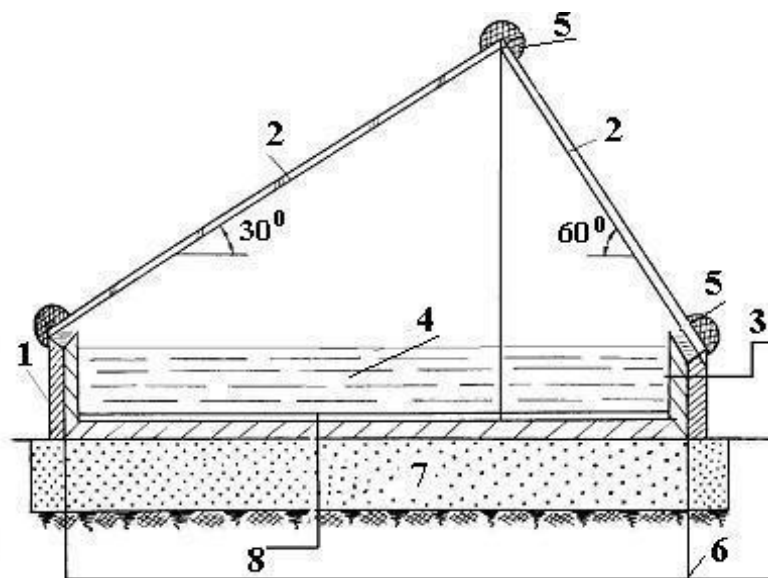


Figure-6.3.2. Schematic of a greenhouse-type double-slope solar water desalination plant: 1—reinforced concrete pool; 2—glass; 3—pipe for pouring brine; 4—brine; 5—seal; 6—pipe for distillate water outlet; 7—dry sand.

It should be noted that, along with greenhouse-type devices, a 100 m^2 slope-step solar water heater (note the upper part of Figure 6.3.3 b) was also built as part of the complex and comparative experiments were conducted. It is noteworthy that the

disadvantages inherent in greenhouse-type two-slope devices were practically not observed in slope-step devices. This became the basis for research on the structural improvement of slope-step solar water heaters and their recommendation for large-scale production.

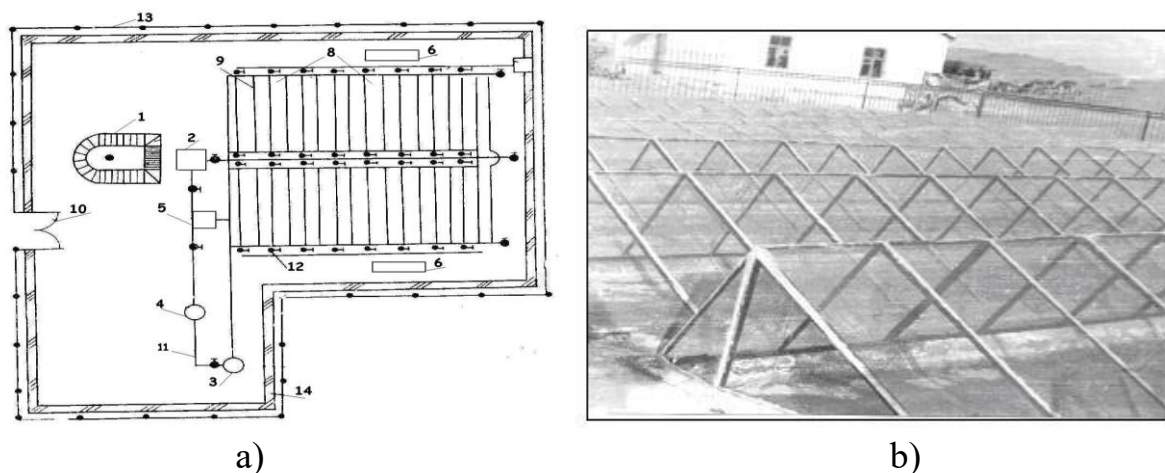


Figure-6.3.3. Layout (a) and general view (b) of a greenhouse-type solar water heater (Kyzylkum, Boymuradkuduk village). a): 1–well; 2–pool for salt water; 3 and 5–pools for distilled water; 4–pool for drinking water; 6–pool for collecting rain and snow water; 7–pool for excess water; 8–greenhouse-type solar water heaters; 9–slope-step solar water heaters; 10–gate; 11–pipes; 12–taps; 13–metal mesh wall; 14–corridor.

6.4. Solar dryers. Their working process and various designs

Drying is one of the most important stages of many technological processes in various sectors of the national economy. In various fields of mechanical engineering and technology, various materials and products, including agricultural products, are dried. Drying is the initial, intermediate or final stage of a particular production. A sharp increase in energy demand leads to the use of alternative energy sources. The relatively low efficiency of power plants operating on such energy sources complicates their use. The increase in the world's population also leads to an increase in the demand for dried food products. The role of solar dryers in the continuous supply of dried food products to the population is incomparable. The drying process has come a long way in its development - from simple to high-tech. It has developed in parallel with the development of fundamental and applied sciences and technologies. Currently, both theoretical and practical aspects of drying have been thoroughly studied, and a whole series of fundamental works have been written on this subject. Scientists who made a great contribution to the theory and practice of drying are A.S. Ginzburg, P.D. Lebedev, A.V. Lykov, B.S. Sajin, O. Krisher and others. To date, various methods, devices and various auxiliary equipment have been developed in various fields of engineering and technology for drying various materials and products, including industrial ones. This

is a whole industry with a serious scientific basis. Natural drying of agricultural products has been used everywhere and since ancient times, and at the same time the products are dried by spreading them on the ground, hanging them under a canopy or picking them up on pallets.



Figure-6.4.1. Solar dryer with polyethylene coating.

Recently, rapid technologies have been used to dry fruits and vegetables. It is good that such technologies are increasingly entering the food processing industry, but, despite the high efficiency of these devices, they consume a large amount of heat or electricity (for example, it is estimated that an average of 2.25 kWh of electricity or 0.675 kg of conventional fuel is consumed to produce 1 kg of ready-made raisins), have a negative impact on the environment, increase the cost of the product, and reduce its naturalness.

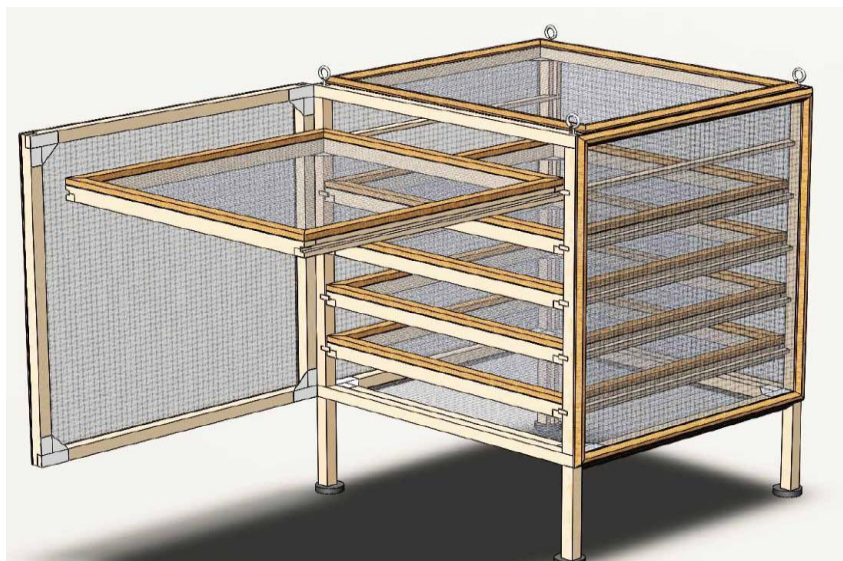


Figure-6.4.2. Natural fruit dryer of mesh box type.

The use of solar-powered fruit and vegetable drying devices leads to a sharp reduction in the above-mentioned shortcomings. Most importantly, these devices

operate only at the expense of solar energy, do not harm the environment, and serve to obtain an environmentally friendly product.

In 1932, A.B. Barzunov made a solar cooker using a transparent organic film. In order to reduce heat loss, he covered the bottom of the device with aluminum sheet. To increase heat exchange and air flow turbulence, the base of the device was made in the form of a staircase. The solar device was equipped with a branch pipe and draft pipes for air intake.

In 1936, a greenhouse-type solar cooker was tested by K.G. Trofimov. To make this device, wooden frames were placed on brick supports at an angle of 200 to the horizon and covered with glass. The fruits that have undergone preliminary processing are spread in circles at the rate of 10 kg per sq. m. and fed into the device. The steam-air mixture is removed by natural gravity. This device requires a lot of labor to manufacture and maintain.

In the 1950s, A. Ismoilova conducted a comparative study of radiation, convective, and convective-radiation solar dryers, and each method had its advantages and disadvantages.

There is a lot of interesting information about the use of solar dryers in foreign countries, including the USA, Germany, Switzerland, Italy, France, and Russia, for drying grains, grass, and alfalfa, in addition to fruits and vegetables. For example, comparative experiments were conducted at the National Laboratory of Civil Engineering in Lisbon on the drying of black cherries and grapes in the open air and in a solar dryer. The cross-section of the solar dryer was constructed in the form of a rectangular elongated chamber and covered with a film. At the beginning and end of the chamber, devices were placed that heated the air under the influence of solar radiation, and in the middle part, mesh trays on which the fruit was spread. The humid air from the device was drawn in by an electric fan and discharged through a suction pipe. The black cherries dried in 5 days, while it took 22 days to dry in the open field. The grapes treated in the helio device dried in 6 days, and the untreated ones in 9 days, while in the open field it took 14 and 19 days, respectively.

In 1972, a tunnel-type solar dryer with a surface area of 448 m² was put into operation in Australia with a helio-air heater. The heated air was blown into a tunnel with a thin black birch base on which fruit was spread using a ventilator. The device allowed to obtain an average of 770 kg of products with a moisture content reduced by 18–30 percent per day. M. Akkurt and M.K. Selkun conducted experiments by installing a gas-fired heater as an addition to the solar-powered dryer. To speed up the drying process or when the sun was not shining, a gas burner was turned on. A high-quality product was obtained and a certain amount of energy was saved. In 1978, a solar dryer with a surface area of 1951 m² was built in Freeno, California. 700 tons of

limestone placed in the chamber of the device acted as a heat accumulator, returning the accumulated energy to the device for up to 24 hours.

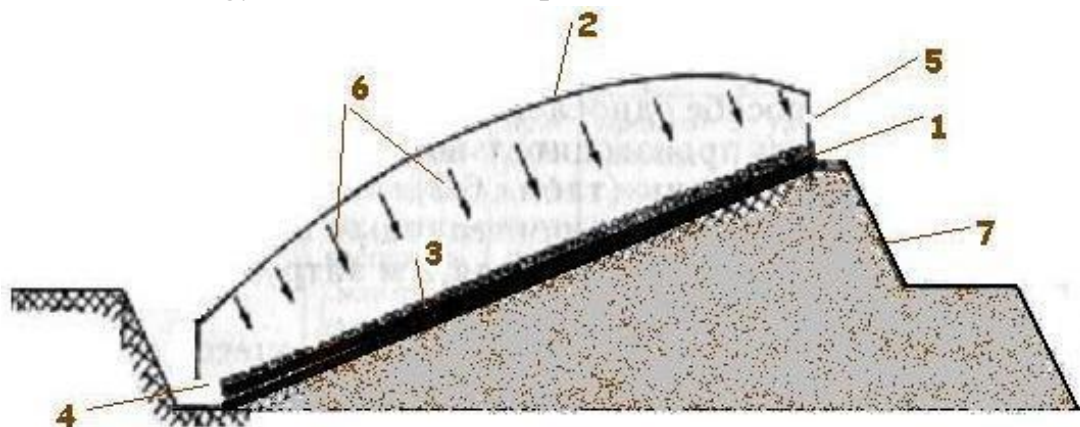


Figure-6.4.3. Schematic of a film inclined solar radiation dryer: 1–frame; 2–film; 3–product to be dried; 4–air inlet; 5–moist air outlet; 6–sunlight; 7–base soil.

In the 1980s, a film inclined solar radiation drying device (QRQQ) was created and tested at the Institute of Physics and Technology of the Academy of Sciences of Uzbekistan and the Tashkent Institute of Irrigation and Agricultural Mechanization (Fig. 6.7.3). The advantages of this device are the low cost of the light-receiving surface unit, low metal consumption, partial accumulation of daytime energy, and the fact that due to the slope of the area, solar radiation falls by 20–25% more than on a horizontal surface. Sunlight passing through the film is absorbed by the product and soil. The temperature of the air inside the device is about 300 degrees higher than the ambient temperature. The slope of the device and the gradient of the temperatures inside and outside the device create conditions for the formation of natural air flow.

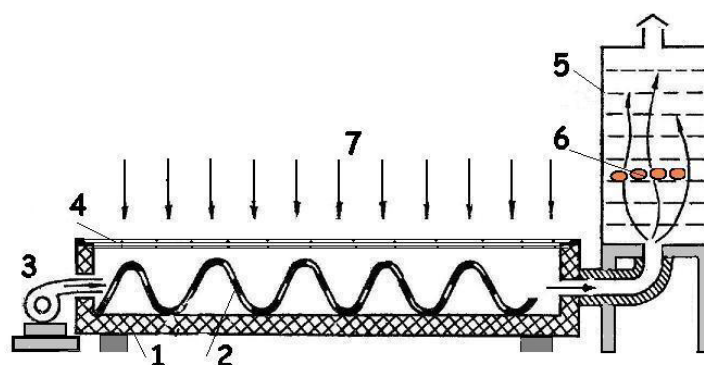


Figure-6.4.4. Scheme of a convective solar dryer: 1–wooden box; 2–black-painted multi-hole corrugated tunic base; 3–fan; 4–double-layer glass; 5–drying chamber; 6–product to be dried; 7–sunlight.

The complex consists of 20 devices with an area of 6 m² each. The devices are installed at an angle of 25° from the horizon and facing south. The drying chamber in the complex measuring 1.5 x 1.5 x 2.0 m is designed for 500–600 kg of product. In continuous drying mode, 0.5–0.6 kg of finished product is obtained per day from each sq m of glazed surface. To increase the productivity of the device, a continuous drying

mode recommended by the employees of the Institute of Mechanics and Structures Seismology of the Academy of Sciences of Uzbekistan was used. The product is dried in the drying chamber until its moisture content drops to 45–50%, then it is removed from the chamber and placed in a shaded place, and after a certain time it is dried again in the chamber to the end. During the initial period of drying, weakly bound moisture in the product is removed. In this case, 3350–4200 kJ of heat is consumed to remove each kg of steam from the product, and the efficiency of the device reaches 55–60%. Then, as the binding energy of moisture with the substance increases, the energy consumption for removing each kg of moisture increases and at the end of the process reaches 92100–109000 kJ, while the useful work coefficient drops to 5–7%.

The scheme of the solar generator proposed by V.N. Danilin and B.D. Babayev (Russia) in 2001 is shown in Figure 6.7.5. The device can consist of several different compartments. Each compartment consists of 1 front and 2 rear walls and 3 covers attached to a metal frame 4. The front wall and cover can be made of wood or other materials in the form of a frame with a transparent film. The back wall consists of a board covered with galvanized tin on its surface. 5 trays are densely placed in the compartments of the frame 4. The trays are installed at an angle equal to the geographical latitude of the location to the horizon and facing south. A container with a heat accumulator (for example, crystalline hydrate salts) is placed under the trays in direct contact with them. The trays are equipped with 7 barriers that prevent the dried product from sliding along the slope and make the surface rough, which creates conditions for better air contact with the product surface. The dryer is connected to the draft pipe 8 through a neck 9, which is equipped with a draft control valve 10. Several holes are made in the front wall for air intake. The dryer works as follows: the lid 3 is opened, the tray 5 is filled with wet product, and an opaque filter cloth is placed on top and the lid is closed.

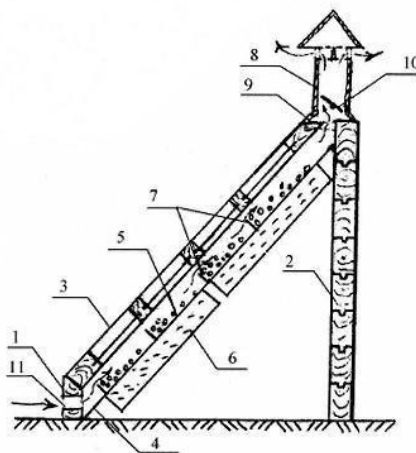


Figure-6.4.5. Scheme of an inclined solar heater with a heat accumulator: 1–front wall; 2–rear wall; 3–cover with a transparent coating; 4–metal frame; 5–trays; 6–

containers with a substance acting as a heat accumulator; 7–barriers; 8–drawer; 9–neck; 10–slat; 11–holes.

During the day, sunlight heats the fabric, product, tray, container and air inside the device. Air draft is created inside the device. Natural convection of air and heating together ensure the efficient drying process. During the day, the heat accumulator material heats up and melts, and in the evening it crystallizes and releases its heat into the device. Thus, the dryer can operate both day and night.

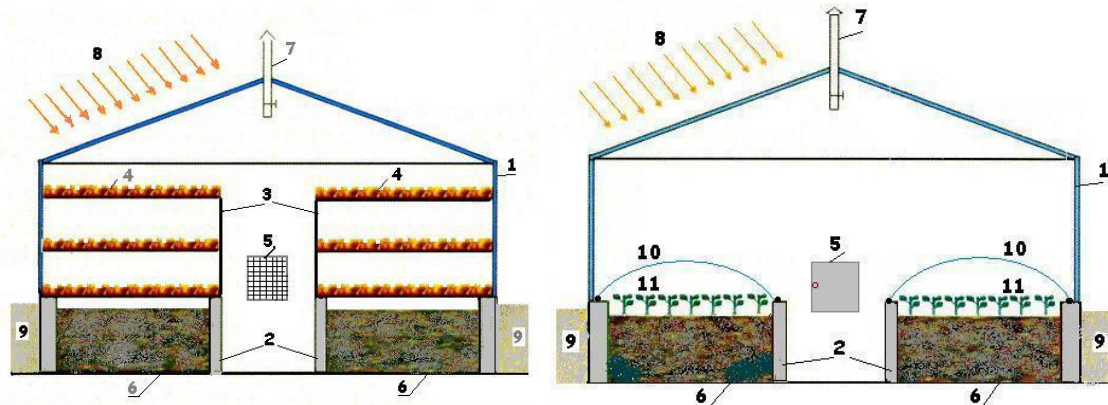


Figure-6.4.6. Scheme of the “GQI-40” solar dryer-greenhouse (summer and winter operating mode): 1–glass-covered building mounted on metal frames; 2–reinforced concrete or brick walls; 3–mesh trays; 4–product to be dried; 5–mesh window; 6–fertile soil; 7–draft pipe; 8–sunlight; 9–dry sand; 10–film coating; 11–greenery and seedlings.

The experimental model of the “GQI-40” solar device consists of a metal structure installed in the north-south direction, covered with 3 mm thick window glass, 7.40 m long, 2.62 m wide, 2.30 m high in the central part, with a transparent cover at the top inclined at 30° to the horizon (Figure 6.4.6). These holes, with a total surface area of 13.5 m², are filled with fertile soil and serve to grow seedlings and greens in winter, and in summer a mesh tray is placed on top and used as a drying area. There is an entrance door in the front of the device, and a mesh window measuring 0.60 m x 0.40 m in size is installed in the back to allow air exchange if necessary.

Inside the device, fruits and vegetables to be dried are washed and separated from the weeds on 3-layer mesh trays installed longitudinally on both sides, and are spread evenly at a rate of 8–12 kg per sq m, depending on the type of product. The device works on the principle of a “hot box”. There are two draft pipes with shutters in the upper part of the device, through which natural ventilation is carried out. Experiments have shown that in the summer-autumn season, the air temperature in the device chamber is 20–250 °C higher than the outside air, and this is a favorable condition for drying fruits and vegetables. Therefore, forced ventilation using a

ventilator is not required. In winter, the soil inside the boxes is prepared, seeds of edible greens are sown, and a film covering is placed on top.

Such conditions are sufficient for growing greens, and additional heating is not required (except in harsh winters). In early spring, in addition to greens, seedlings can be grown in the device. By increasing the number of devices (the length of the device can also be increased) and placing them parallel to each other, it is possible to create an efficient solar dryer-greenhouse with an optional drying area.

The “GQI-3” solar dryer-greenhouse device is included in the category of mini-technologies, which also serves to grow fruits and vegetables in the summer-autumn season and greens and seedlings in winter and early spring using solar energy. The mini solar fruit and vegetable dryer-greenhouse device consists of a “hot box” made of wood with a length of 1500 mm, a width of 720 mm, a front part of 600 mm, a rear wall height of 900 mm, and a front wall and top covered with clear glass. On both sides of the device there are glass-framed doors and mesh windows providing natural ventilation. The device chamber has three rows of sliding mesh trays on which the products to be dried are spread. The surface of each row of mesh trays is 1 m² (two of 0.5 m²). The total drying area of the device is 3 m². To provide forced ventilation of the products in the drying chamber, a mini-fan powered by a 4–5 Watt solar battery is installed at the bottom.

After the product to be dried is washed, the device is placed on the mesh trays, the doors and windows are closed. The main part of the sunlight falling on the glass surfaces of the device enters the device chamber, is absorbed by its parts and fruits and vegetables, and is converted into thermal energy. When the temperature inside the device starts to exceed 550 °C, the mesh window is opened. The air mixed with the steam released from the products is removed outside by convective currents using natural ventilation. Experiments show that there is almost no need for forced ventilation. However, if a solar battery is installed and forced ventilation is carried out using a mini-fan, the drying time of the product is reduced by up to 20 percent. To ensure uniform drying of the products on all mesh trays, their positions can be changed from time to time. Usually, mechanically bound moisture is removed from the product. At the end of the drying process, the moisture content of the product is approximately 18–25 percent. In fruits and vegetables with a high sugar content, evaporation occurs much more slowly than on the free surface of water. When the product dries, it is removed from the mesh trays and the device is prepared for the next operating mode. When drying colorless grapes, as well as other varieties of grapes, in the shade, a black multi-hole metal screen is installed above the upper mesh tray of the device and in the front part. In this case, the sun's rays fall on the screen, not directly on the product. Infrared rays emanating from the heated screen play an important role in drying the

product. Although the drying process is delayed by 1–1.5 days compared to without a screen, the quality of the product increases further.

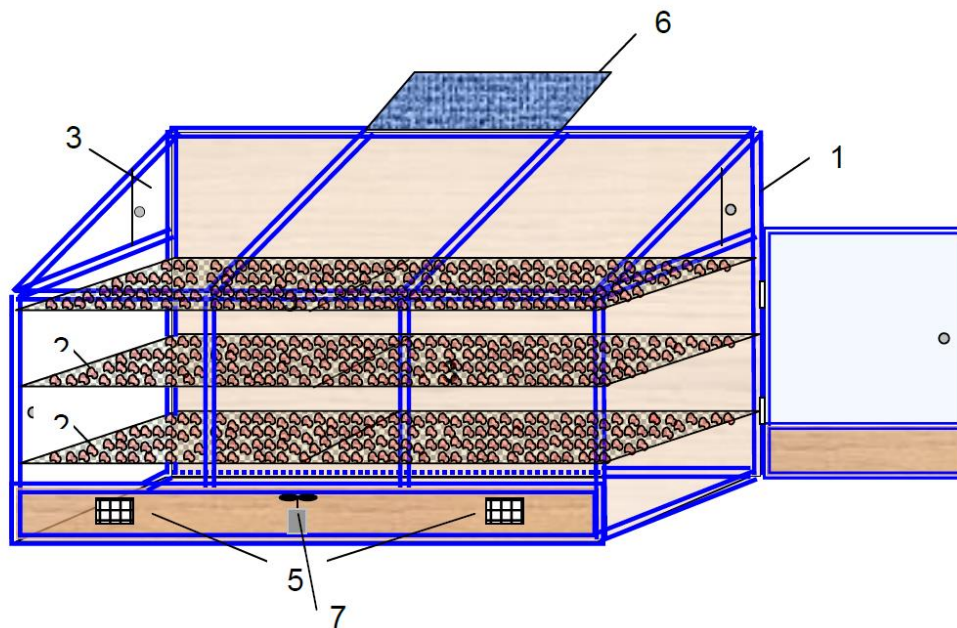


Figure-6.4.7. Working scheme of the “GQI-3” solar-heater-greenhouse-greenhouse device: 1–wooden frame covered with glass; 2–mesh trays for spreading dried fruits and vegetables; 3, 5–windows providing natural air exchange; 4–glass door of the device; 6–solar battery; 7–fan (connected to the solar battery).

6.5. Solar fruit dryers

In summer and autumn, wet fruits (apples, apricots, grapes) are dried in high quality. Fruit and vegetable products can be dried in the following ways:

1. Drying products directly in open areas under concentrated solar radiation.
2. Drying products in special devices for artificial drying.
3. Drying products in special solar devices.

Studies show that drying fruit products in special solar devices has a number of advantages. These advantages include the following:

1. Less time is spent on drying in solar devices.

2. The quality of products dried in solar devices is better than that of products dried in open areas, that is, the content of dried products is higher in vitamin C, and the dried products are cleaner. There are various types of solar dryers, which differ from each other in the heat transfer coefficients of the material and the volume of the dried product. The dryer walls are mainly made of materials with very low heat transfer coefficients. The views of solar dryers are shown below. Agricultural products dried without protection from the air suffer significant losses, as a result of incomplete drying, rotting, mold, bird droppings, insect damage, and wind exposure, which increase product waste.

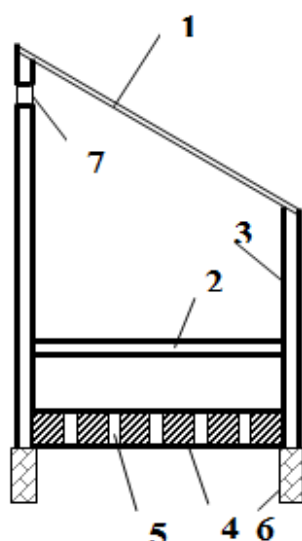


Figure-6.5.1. Solar dryer with direct irradiation of wet material: 1-transparent insulation; 2-platform for material; 3-wall; 4-thermal insulation; 5,7-slot; 6-base.

The use of a “hot box” type solar device increases the efficiency of the dryer and reduces product losses. It significantly reduces the drying time and improves the quality of the product, while preserving vitamins. However, the utilization rate of solar dryers in agriculture is very small. In some cases, it may only be used for a few weeks a year, and this, of course, does not ensure high economic efficiency of the solar dryer.

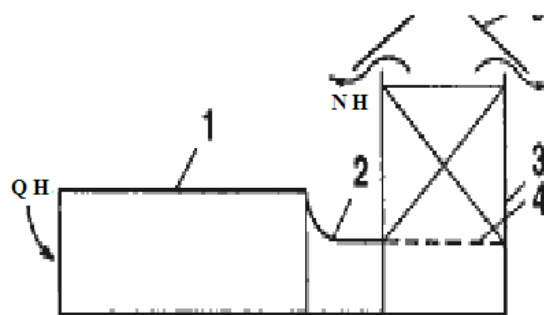


Figure-6.5.2. Solar-air film dryer with heater: 1 - solar-air film heater; 2 - air supply pipe; 3 - dryer chamber; 4 - grille; 5 - umbrella, QH-fresh air, NH-humid air.

Solar dryers are distinguished by the direct and oblique movement of solar energy. In the first type of installation, the drying energy is absorbed directly by the material itself, and the material to be dried is placed on a platform with the inner walls of the chamber painted black. This type of solar dryer is shown in Figure 6.5.1. It consists of a transparent solar insulation at the top, perforated platform side walls for placing the dried material (the southern wall is made of transparent material), a perforated thermal insulation base for air intake. There are holes in the upper part of the northern wall to remove moist air from the solar dryer. The second type of dryer device consists of a solar air heater and a chamber or tunnel dryer. In a chamber solar dryer, the material being dried moves along a layer, placed from bottom to top on a

mesh tray, while in a tunnel dryer, the material moves along a conveyor belt, but in the opposite direction to the direction of the air flow.

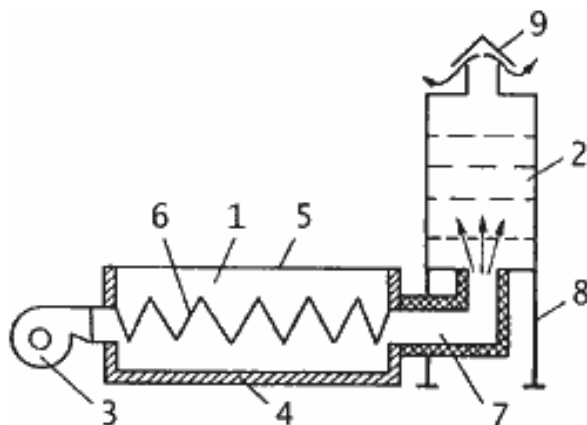


Figure-6.5.3. Solar dryer with a fan-heated chamber: 1 - air heater; 2 - drying chamber; 3 - fan; 4 - heat-insulated housing; 5 - transparent insulation; 6 - absorber; 7 - air supply pipe; 8 - support; 9 - canopy.

Let's look at some examples of a chamber solar dryer. Simple dryers using a polymer film can be designed as shown in Figure 6.5.2. It works on the basis of natural convection. Air is heated in a film solar air heater and enters the lower part of the dryer chamber from the air outlet, where the wet material is placed on a perforated base (mesh, grille). As can be seen in Figure 6.5.3, the solar chamber dryer is implemented with a forced spray method. It includes the following: an air heating device; a drying chamber; a fan. The transparent insulation in the heat insulation shell of the air heater is made of light-absorbing, blackened corrugated metal. From the heat-insulated air guide, hot air enters the drying chamber onto a perforated tray for the material being dried, which is covered with a canopy from above and mounted on a base.

The presented solar dryer has high efficiency. The collector efficiency reaches 75% at a high air flow rate ($0.5 \text{ m}^3/\text{s}$ or $0.13 \text{ kg/s}\cdot\text{m}^2$), the pressure drop reaches 250 Pa. The payback period is about 5 years.

A simple and inexpensive solar dryer is made of transparent and black polymer films, stabilized against ultraviolet rays. (Fig. 6.5.4). A 0.1 mm thick polyethylene film was laid on the wooden frame, the bottom was made of black polyethylene film (0.1 mm), this bottom had a thickness of 75 mm and a row of corks was placed on it, which served as thermal insulation. The lower part of the side walls was sprinkled with soil, and the length and thickness of the collector were 30 m and 4.6 m. The heated air enters a cylindrical chamber with a diameter of 1.5 m and a height of 1.8 m, where 1.75 tons of household products are placed in several layers with a thickness of 150 mm and dried. The panels of the solar air collector are attached to the fan using an air duct. The light-absorbing surface of the air collector is a matrix, which consists of a contact surface that absorbs solar radiation and heats the superheated air. The two side and rear

walls of the housing are made of iron covered with a copper layer, which is thermally insulated.

The transparent insulation is made of a special durable polymer material that is highly transparent to solar radiation and reflects ultraviolet radiation.

Typical modules, 4.2 m wide and 2.5 m long, are usually placed in a row. Two panels, 14.5 m long, are welded to a single fan, through which air is drawn from this solar collector. Thus, for a solar dryer with a flat solar collector surface of 120 m², one fan with a power of 3.5 kW is sufficient. The daily operating capacity of the dryer is 19 MJ/m², which means that 400 kg of dried product can be obtained (Figure-6.5.5).

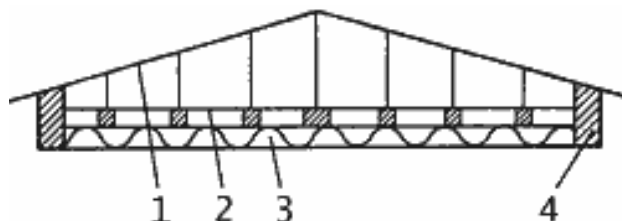


Figure-6.5.4. Film solar dryer: 1 - transparent polymer film; 2 - black film for spreading fruits; 3 - thermal insulation; 4 - side walls.

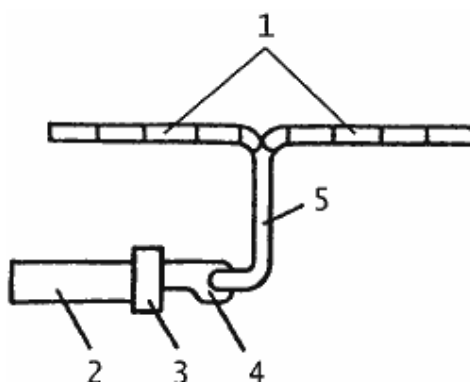


Figure-6.5.5. Modular solar dryer: 1 – air heater module; 2 – drying chamber; 3 – fan; 4 – air distributor; 5 – air duct.

The grain is placed in a horizontal hopper, which is driven by heated air. Similar devices are used to dry corn and other grain products, as well as tobacco leaves. For drying grain, heated air from a horizontal hopper or a vertical hopper can be used. The air temperature (layer height up to 4 m) should be 2...30 °C or 5...15 °C (layer height up to 1.5 m) higher than the ambient temperature. The limitation of the layer thickness is dangerous due to the formation of condensation of water vapor in the upper part, which is especially observed on rainy days and days with high air humidity. The following methods can be used to dry fodder and hay: Drying with hot air (30 °C) or warm air (40...80 °C); Blowing with slightly heated air (0...10 °C);

Ventilation with unheated air outside and drying the hay on the ground under natural conditions. Drying hay with solar energy consumes less energy than drying with a liquid fuel-powered device and is approximately equal to the energy consumption during drying with unheated air. The system uses a solar air collector,

which increases the air temperature to 20 °C on a sunny day and to 10 °C on a cloudy day. In this case, the moisture content of the hay decreases by 5% on a cloudy day. A solar collector with ordinary glass or the roof of the building itself can be used as a solar energy collector. The bottom of the solar collector is located under it and air is circulated using a fan.

Construction timber can be dried in a heat-insulated chamber with a volume of 65 m³. 10 m³ of material is placed in it; air circulation is carried out using a fan; the air is heated in a collector with a surface area of 75 m².

The solar air dryer attracts with its simplicity, it is not dependent on weather conditions, since drying occurs only at high temperatures and low humidity. If there is a lack of heat, the resulting product will be of poor quality. Usually, the prepared products are transferred to a single or multiple-layered plain thread and hung in the sun, or simply placed in a single row on a plywood sheet and dried in the sun.

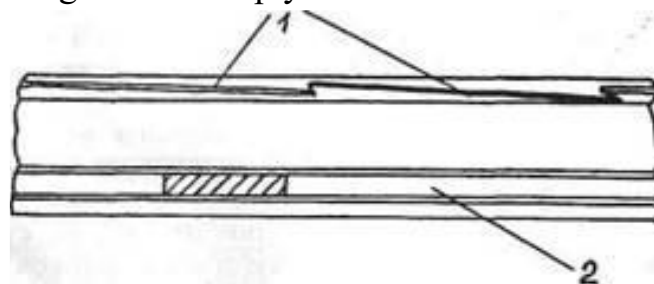


Figure-6.5.6. Drawing of a solar-wind dryer device: 1 - glass; 2 - two-layer base.

This type of drying takes a long time (4-15 days). During drying, it is necessary to ensure that the product being prepared does not get wet. In the evening, it should be taken indoors to protect it from possible precipitation, and if necessary, covered with gauze to protect it from mosquitoes and other insects.

A solar-wind dryer, with this method, large quantities of grapes, apricots and other fruits are dried. Such dryers can be used in some regions of Russia in the summer months and early autumn in dry and sunny areas, as well as in the Caucasus, Moldova, and southern Ukraine (Figure-6.5.6).

The use of solar-wind dryers is limited in northern regions due to the risk of product spoilage. The convenience of the solar-wind dryer is that it can be used as a specially designed solar dryer. The device is simple and convenient for everyone.

The size and structure of a solar dryer vary depending on the material or the volume of the product being dried. The dryer's working principle is based on drying fruits with the help of sunlight, while protecting them from dust, insects, rain, and moisture (dew). That is why the device must be such that it can transmit as much sunlight as possible and simultaneously remove the moisture that has formed.

The temperature-humidity regime in solar dryers depends mainly on solar radiation, external temperature, air humidity, type of dryer, and other factors. These factors change throughout the day. Therefore, controlling the temperature-humidity

regime in solar dryers is one of the important problems. The quality of dried fruits and vegetables depends on the temperature-humidity regime in the drying chamber, preliminary processing of fruits, and other influences. To solve this problem, a continuously operating automated solar fruit dryer device was created (Figure-6.5.7). The processes occurring inside the device are controlled automatically. The maximum temperature inside the chamber is 62 °C. In recirculation mode, the temperature difference along the height of the chamber is 4-6 °C.

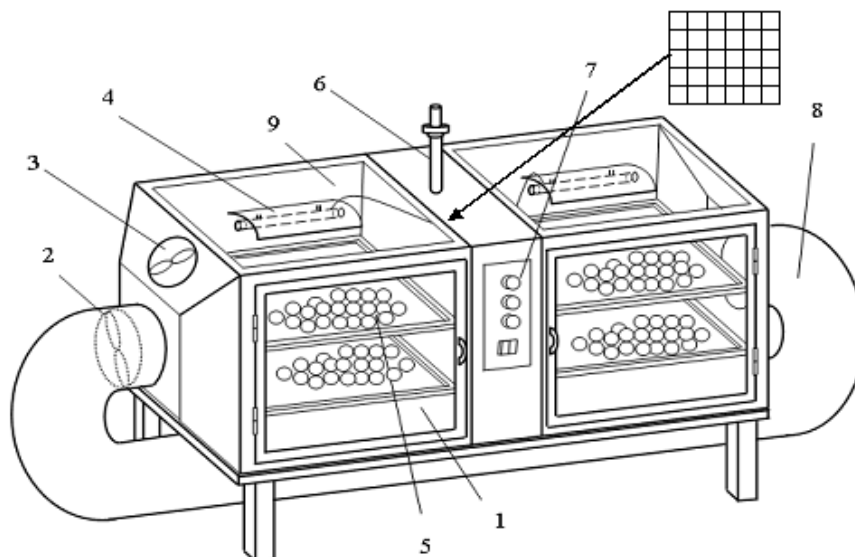


Figure-6.5.7. Schematic diagram of the drying device. 1 – drying chamber, 2,3 – fans, 4 – IR-reflector lamp, 5 – product tray, 6 – contact thermometer, 7 – control panel, 8 – air flow circulation pipe, 9 – glass layer, 10 – photocell.

In this case, the speed of the heat carrier is determined to be 1-1.5 meters per second, and the relative humidity of the air is 35%. From 14⁰⁰ to 16⁰⁰ hours, the temperature decreases to 30-34 °C. Naturally, at low temperatures, the drying rate of fruits decreases. To maintain the drying rate of fruits at a certain rate, an additional heat source (IR lamps) is used in the chamber. When the temperature drops below 35-40 °C, the thermoregulator connects the IR lamps. If necessary, the IR lamps can be connected to the network using a regulator at any temperature not exceeding 70 °C. IR lamps are used for continuous operation of the device. The energy required to operate the device is provided by converting solar energy into electrical energy using a photocell. The high quality of fruits dried in this device distinguishes it from products dried in other devices. Depending on the ripeness and season of the product, each square meter of the device can be used to dry 25-30 kg of apricots, 10-15 kg of cherries, 25-30 kg of tomatoes, 25-30 kg of raisins, 5-10 kg of melons, etc.

Fruits and grapes contain vitamins necessary for the body, which should be consumed in a certain amount. According to the recommendations of the world's authoritative academies of medical sciences and folk medicine, for the normal

development of the human body, it is recommended to consume 7 kg of dried fruits or 5-7 raisins per day.

Today, in world civilization, it can be seen that the drying of fruits and grapes is conditionally divided into 3 directions:

- 1) natural drying in the open air under the influence of solar radiation;
- 2) drying using gas, oil or electricity using specially designed machines;

3) drying using special solar devices using solar energy. Based on the studies conducted, it can be said that natural drying takes a long time, and if the weather is unfavorable, the product is lost. For artificial drying, 0.5 kg of fuel is consumed per kg of raisins. Solar devices do not require energy. However, they require a larger area than artificial drying.

Currently, our scientists have created devices that effectively use solar energy to dry agricultural products, and work is underway to create efficient structures.

Table-6.5.1

Dryers	Outdoors	Radiation dryer	Chamber dryer	Combined
Product per square meter	15-20	15-20	15-20	20-25
Drying time (days)	17-20	7	9	6-8
Productivity kg/m ²	0,25	0,65	0,5	0,9

Solar dryers can be conditionally divided into 3 groups. 1. Radiation dryer. In this case, the product to be dried is placed in a hot box. The fruits are dried directly under the influence of sunlight. 2. Convective dryer. In this case, the fruits to be dried are placed in a special chamber with trays, and air heated by solar-air heaters is directed into the chamber using a fan. 3. Combined dryer. The following table 6.5.1 presents the results of comparative tests of grape drying in various devices. When calculating any dryer, the following are taken as initial data.

1. The efficiency of the dryer relative to wet or dry product.

$$\Pi_i \frac{kg}{day} \text{ or } \frac{kg}{hour}$$

2. Initial moisture content of the dried product W_H %;
3. Final moisture content of the dried product %;
4. Ambient temperature °C;
5. Temperature inside the drying chamber °C;
6. Drying time τ days.

The dimensions of the solar dryer are calculated in accordance with the following formulas.

Initially, the amount of moisture evaporated in the drying chamber is determined from the following formula.

$$m = \frac{G_1 W_1 - G_2 W_1}{100} \quad (6.5.1)$$

where: G is the mass of the material before drying and the mass of the dried (fruit) product, respectively; $W_1 W_2$ - moisture content of the material before and after drying, %.

The amount of heat required to evaporate this amount of moisture in the material (fruit) is found from the following expression.

$$Q = \frac{1000m}{d_2 - d_1} i_0 \quad (6.5.2)$$

where, i_a – is enthalpy of drying agent,

$$i_a = ct_0 + 0,001d(2500 + 1,81t_0) \quad (6.5.3)$$

In this, d_1, d_2, d_0 are the absolute humidity of the drying agent in the drying chamber, the air leaving the chamber, and the dry air. C - is the heat capacity of the dry air;

Solar radiation - the solar energy falling on the transparent surface of a convective dryer - is found from the following relationship.

$$ADJS \cos i = Q_{nur} + Q_{kon} + Q_{IB} + Q_B = C \left[\left(\frac{T}{100} \right)^u - \left(\frac{T_0}{100} \right)^u \right] + \alpha(t_6 - t_0) + Q_{IB} + Q_B \quad (6.5.4)$$

where, J – is the intensity of solar radiation (energy) falling on a transparent surface of a dryer W/m^2 . D – is the transmittance of glass or film. A - absorption coefficient. T - temperature in the drying chamber. T_0 - the temperature of the glass; the amount of heat removed from the drying chamber and with the exhaust air. t_0 - average air temperature. t_k - camera temperature. α - heat transfer coefficient, $\alpha=5,7+3,8v$ where, v – air speed.

The thermal characteristics of glass or film are taken from the data sheets. For example, the thermal characteristics of PTFE film are as follows:

$$D=0,88; A=0,94; \lambda_\delta=0,44 \text{ W/MK}; \alpha=8 \text{ W/m}^2\text{k}$$

film thickness $d=0.001m$; the size of the transparent surface of the chamber is determined to obtain the necessary heat from the material (fruit). The selection of a fan that moves the drying agent in the dryer is determined by its air consumption as follows. The daily evaporation rate from the fruit (product) to the medium is found as follows.

$m_{orr} = \frac{m}{\tau}$; where, τ – is the fruits drying time.

The dryer air consumption (L) is found in the following formula:

$$L = \frac{1000m}{d_2 - d_1}$$

The value of d_1, d_2 is dependent on the air parameters.

The average building of fruits can be found to build all the fruits, knowing the duration of the duration of τ - knowingly.

$$L = L_{orr} \tau \quad (6.5.5)$$

L - The volume of air, knowingly, can be found in the following expression.

$$L = \frac{L}{\rho} \quad (6.5.6)$$

As it is not mentioned above, m is found in the heat balance sheet the amount of heat spent drying m mass fruit.

$$Q = Q_1 + Q_2 + Q_3 + Q_4$$

where, Q_1 and Q_2 are appropriate amount of heat necessary to heat and evaporate. Q_3 - The amount of hovered heat due to convection; Q_4 - The amount of heat-cutting air with the air used in the dryer. This heat amounts can be found in the following world-class.

$Q_1 = C_m(t_2 - t_1)$ - The heat amount necessary to heat the product, C - is a heat capacity of the fruit.

$Q_2 = rm$ - The amount of heat necessary to evaporate the fruit.

$Q_3 = \alpha(t_2 - t_1)$ - The amount of heat lost due to convection.

$Q_4 = L(i_2 - i_1)$ - The amount of heat that is administered out with the air used.

The solar radiation entering the dryer through a shake is found in the following expression.

$$Q = BJS \cos i \quad (6.5.7)$$

where, i - the corner of falling into the surface of the sun. Using this relationship, the useful work factor of dryer can be found.

$$\eta = \frac{rm}{Q_1 + Q_2 + Q_3} \quad (6.5.8)$$

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Co-funded by the
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