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DEVELOPMENT OF RESOURCE-SAVING TECHNICAL AND TECHNOLOGICAL SUPPORT FOR PRODUCTION AND ANIMAL HUSBANDRY

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Kaletnik G., Yaropud V., Babyn I.

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Authors:

Kaletnik Grygorii – Doctor of economic sciences, professor, academician of the National Academy of Sciences of Ukraine, Vinnytsia National Agrarian University, Ukraine.

Yaropud Vitalii – Candidate of technical sciences, Associate Professor, Vinnytsia National Agrarian University, Ukraine.

Babyn Ihor – Candidate of technical sciences, Associate Professor, Vinnytsia National Agrarian University, Ukraine.

Editor:

Kaletnik Grygorii – Doctor of economic sciences, professor, academician of the National Academy of Sciences of Ukraine, Vinnytsia National Agrarian University, Ukraine.

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ABSTRACT

Animal husbandry, as an integral component of the agro-industrial complex, plays a decisive role in ensuring Ukraine's food security, stimulating the development of rural areas and promoting the country's economic growth. However, the industry is significantly affected by both internal and external factors, which requires a comprehensive analysis and development of effective development strategies.

The relevance of the monograph lies in the need for a comprehensive analysis of the current state and prospects for the development of animal husbandry in Ukraine in the context of global changes occurring in the agro-industrial sector. The growth of the world population, changing consumer preferences, climate change and increased competition in the global food market pose new challenges to the industry. At the same time, animal husbandry has significant potential for development and can become one of the drivers of economic growth in Ukraine.

The main method of increasing the productivity of animals remains the creation and improvement of breeds with a high level of genetic potential through purposeful selection and breeding work. However, for the use of livestock, it is important to provide technological conditions that allow the animals to show their potential as effectively as possible.

Animal health and productivity largely depend on the microclimate of livestock premises. If it does not comply with the optimal zoohygienic parameters, the hope of cows decreases by 10...20%, the weight gain of animals - by 20...30%, the departure of young animals reaches 30%.

The creation of a favorable microclimate in livestock premises also affects the working conditions of service personnel, the service life of buildings, and the improvement of operating conditions of technological equipment.

Making mistakes in the projects of farms, complexes and various deviations in order to save money during the construction and reconstruction of buildings leads to a violation of the parameters of the microclimate in the premises, that is, a change in the conditions of maintenance. A suboptimal microclimate can lead to disruption of the

interaction between the animal organism and the air environment. According to a number of researchers, environmental factors affect the body in a complex manner, therefore, for an effective assessment of the microclimate, it is necessary to consider its indicators in combination with each other. Thus, a low temperature with optimal humidity leads to an increase in metabolism and an increase in heat production, while high temperature and air humidity make it difficult for moisture to evaporate from the skin and, as a result, the body overheats - the condition worsens.

The microclimate in cowsheds is determined by many factors, including climatic conditions in the region, the time of year, structural features of livestock premises, and the efficiency of the main technological processes. The natural and climatic conditions of the Udmurt Republic, the climate has a moderately continental character, are of great importance in this regard. The average long-term air temperature in January is -14°C (in cold periods the air temperature reaches -45.0°C), in July it is +19.0°C (in the period of extreme heat it reaches +38°C). The average annual precipitation is from 450 to 600 mm. The wind load is 27 kg/m with the predominance of the northwest wind throughout the year.

Inconsistency of the actual constructive solutions with the typical causes significant heat losses of buildings, creation of unsatisfactory conditions for keeping animals, deterioration of the air environment. Nevertheless, livestock premises are often built in farms without the necessary calculations, documents and without compliance with regulations.

The monograph was written as part of the implementation of the state theme on the basis of the Vinnytsia National Agrarian University Development of a complex of technical and technological support for energy- and resource-saving production of animal husbandry products within the framework of EGD (State registration number: 0123U101794).

CONTENT

1 ANALYSIS OF MODERN SECURITY SYSTEMS MICROCLIMATE OF PIG PREMISES

1.1 Influence of microclimate parameters on animal productivity

For a significant part of the year, and according to some technologies, the whole year, most farm animals are indoors [1]. In this regard, it is necessary to create a microclimate in livestock premises that will correspond to the physiology of animals and poultry and have a favorable effect on their condition, health, productivity and product quality [1].

The microclimate of the livestock premises is the state of the environment, which is formed as a result of the vital activity of animals under the conditions of a certain technology. The main microclimate indicators include: temperature T; relative air humidity W, %; chemical composition of air (carbon dioxide content CO_2, ammonia NH_3, hydrogen sulfide H_2S); the presence of dust (mechanical pollution) and microorganisms (biological pollution) in the air; speed v, m/s, and direction of air movement; lighting. The air regime is disturbed when animals breathe (emission of heat, moisture, carbon dioxide, etc.), as well as as a result of evaporation from manure. Gases (carbon dioxide, ammonia, hydrogen sulfide), as well as factors such as moisture and heat, are among the main pollution factors that most affect the development of animals.

Standardized generalized values of the main indicators of the microclimate according to the requirements of departmental norms of technological design [2, 3, 4] are given in the table 1.1.

Deviation of microclimate parameters from physiologically determined norms weakens the resistance of animals to diseases, causes the departure of up to 40% of young animals, a decrease of 10-20% of milk yield, a decrease of up to 30% of weight gain at fattening, wool shearing of up to 20%; requires additional costs of feed and funds for treatment. Deterioration of the microclimate also shortens the service life of livestock premises and their technological equipment [5].

Table 1.1.

Normalized (averaged) values of microclimate indicators

Rooms	The value of indicators					
	T, C^{o}	W, %	v, m/s	content in the air		
				CO_2, %	NH_3, mg/l	H_2S, mg/l
cowsheds:						
- restraint	10	75	0,5	0,25	0,02	0,01
- unattached maintenance	5-8	70-75	0,5	0,25	0,02	0,01
Maternity ward	15-17	75	0,3	0,2	0,01	0,005
Sow farms	18-22	70	0,15			0,005
Pig pens for rearing	18-22	70	0,2			0,01
Fattening pig farms	14-20	75	0,3	0,2	0,02	0,01
Sheep farms with:						
- young man	16	70-75	0,2	0,25	0,02	0,01
- adult livestock	10		0,3			

In addition to species and age characteristics and the density of animal housing, the microclimate in the livestock premises is influenced by other factors: climatic conditions; structural features of the building and materials from which its elements are made; ways of keeping animals; means of mechanization [6, 7].

Zootechnical and sanitary-hygienic requirements for creating a microclimate are reduced to the fact that all its indicators are maintained within the limits defined by the norms of technological design of premises for keeping animals [12, 13]. It should be emphasized the importance of maintaining the stability of the level of microclimate indicators. A sharp violation of regimes is especially harmful. If the deviation from the optimal norms for one or another indicator is accompanied mainly by a decrease in animal productivity, then a sharp fluctuation in regimes (for example, temperature) is often the cause of disease and death of animals, primarily young animals [8, 9].

The first significant factor after feeding, which has a significant impact on the animal's body, is the temperature of the environment. Air temperature is the main physical stimulus of the body that affects its heat exchange. Any decrease in air temperature below the critical temperature leads to an increase in metabolism and heat production in the animal's body, to overconsumption of feed. If compensation of losses is impossible or untimely, then there will be a decrease in productivity. When cattle are kept in rooms with an air temperature below 5 °C, milk production decreases by 1-2 liters from each cow, weight gain of calves decreases by 15-20%. Young animals are most sensitive to low temperatures. Thus, newborn piglets have almost no subcutaneous fat and poorly developed physical thermoregulation. Therefore, they are practically unable to store the heat generated in the body as a result of the metabolic process. In addition, they have a large surface area per unit mass of heat, and their heat output is much higher than that of adult animals. The mechanism of physical thermoregulation in piglets and calves begins to function from 6-10 days after birth, and is actively included in the process only after 10-12 days in calves and after 30 days in piglets. Therefore, in the first 10 days of life, up to 80% of sick young animals die, and about 26% of pathologies are caused by non-infectious diseases of a cold nature [10].

In studies [11], a graph of the dependence of the productivity of meat breeds of cattle and their feed consumption on the air temperature in the area of their keeping is given (Fig. 1.1). From the graph in Fig. 1.1, it follows that for meat breeds of cattle, high productivity of animals in terms of weight gain is observed in the air temperature range of 10-26 °C.

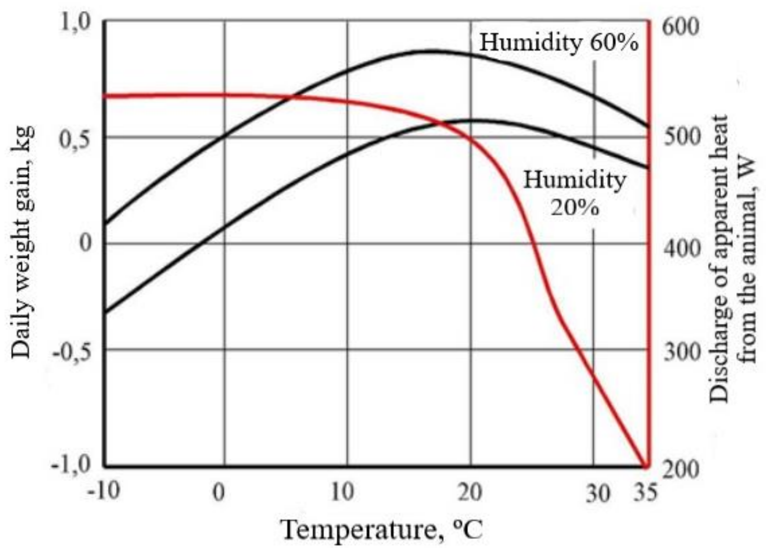


Fig. 1.1. Dependence of productivity and return of apparent heat meat breeds of cattle from temperature and relative humidity of the air in the area they are kept

Studies [12, 13] established that lowering the ambient temperature below the optimum increases the exchangeable energy requirement of pigs by 17 kJ/(kg·ºC) of piglets from 20 to 45 kg of live weight on average, and of fattening pigs from 45 to 85 kg - by 15 kJ/(kg·ºC), boars and sows - from 85 to 120 kg - by 13 kJ/(kg·ºC). When kept with a temperature in the room below the optimum, the average daily gains of fattening pigs decrease by an average of 22 g for each degree of temperature that is below the optimum. In other words, when the temperature drops by 3 degrees below the norm, the overconsumption of feed is about 9% (Fig. 1.2).

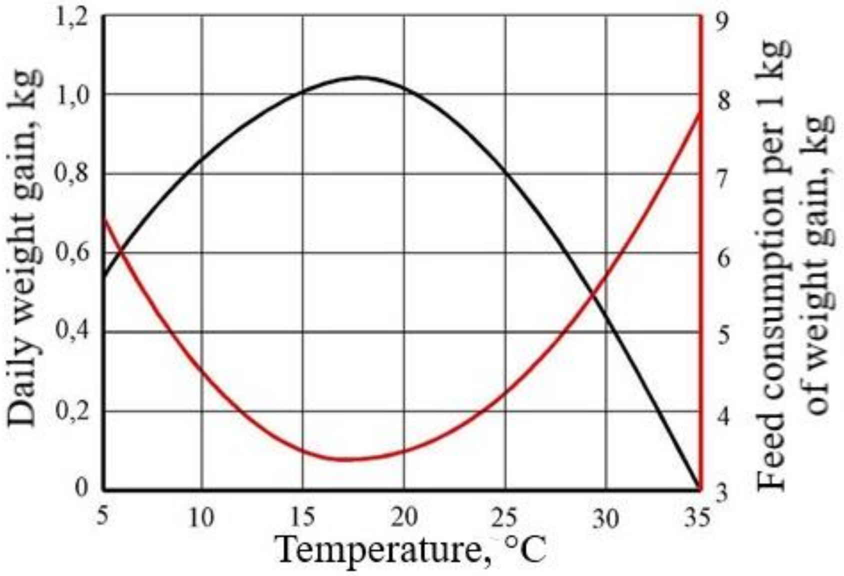


Fig. 1.2. The effect of indoor air temperature on productivity fattening pigs and the need for fodder

In studies [14, 15] it was found that an increase in the air temperature in the room causes significant deviations in the body's activity in cows. First, the metabolism decreases, as a result of heat stress, the appetite decreases, the secretory, enzymatic, and motor functions of the gastrointestinal tract weaken. Thus, at a temperature of: 21 °C - feed consumption begins to decrease and the hope decreases by 4%; 24 °C – milk productivity significantly decreases by 7%; 25 °C – body temperature increases, hope decreases even more by 10%; 33 °C – milk yield sharply decreases, milk fat content drops, death from overheating may occur (Fig. 1.3).

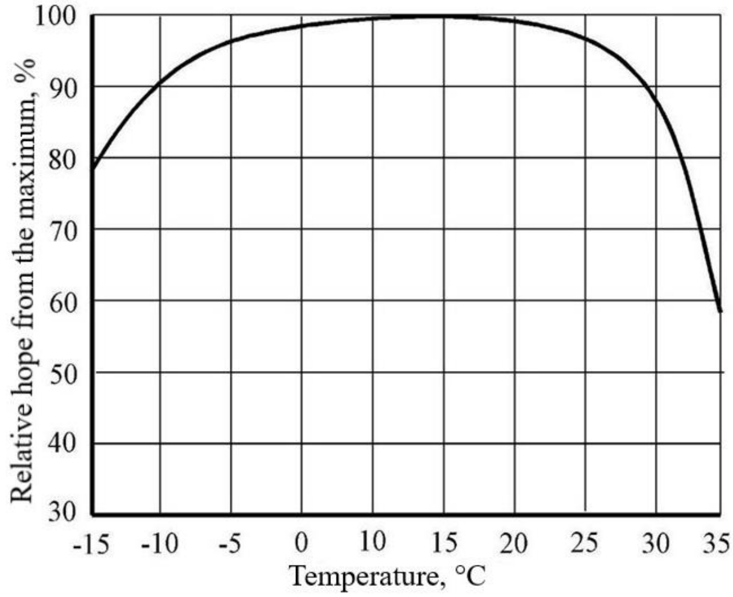


Fig. 1.3. Dependence of the relative productivity of cows from the air temperature in the room

In order to achieve the maximum productivity of animals, it is advisable to provide the microclimate in livestock premises (air exchange and air temperature) from the energy point of view with air heat exchangers, the use of which allows you to save the energy needed to heat the air in the premises [4].

1.2 New technologies for keeping pigs

Today, the total number of pigs in the world is approximately 850 million heads. The following countries have the largest population of these animals: China - 473 million heads, USA - 60 million heads, Brazil - 33 million heads, Germany - 28 million heads, Spain - 24 million heads, Vietnam - 23 million. heads, Poland and Mexico - 18 million heads each, the Russian Federation - 16 million heads, Ukraine - 8.2 million heads and the Republic of Belarus - 3.3 million heads [16, 17, 18].

Denmark ranks first in pork production per person annually, reaching 328.5 kg. Spain is in second place with a result of 81.2 kg, the Netherlands - 77.1 kg, Canada - 62.5 kg, Poland - 54.4 kg, Germany - 52.9 kg, France - 38.1 kg, China - 36 ,0 kg, the

USA – 31.7 kg, the Republic of Belarus – 31.3 kg, Vietnam – 20.9 kg, Ukraine – 12.9 kg and the Russian Federation – 12.2 kg. [1, 2, 3].

According to the data of the State Statistics Service, as of January 1, 2022, the total number of pigs in Ukraine amounted to 5,608.8 thousand heads (not including the temporarily occupied territory of the Autonomous Republic of Crimea, the city of Sevastopol, and part of the area of the anti-terrorist operation) [19]. At the same time, 49% are held in the private sector and 51% in rural areas. enterprises (Fig. 1.4).

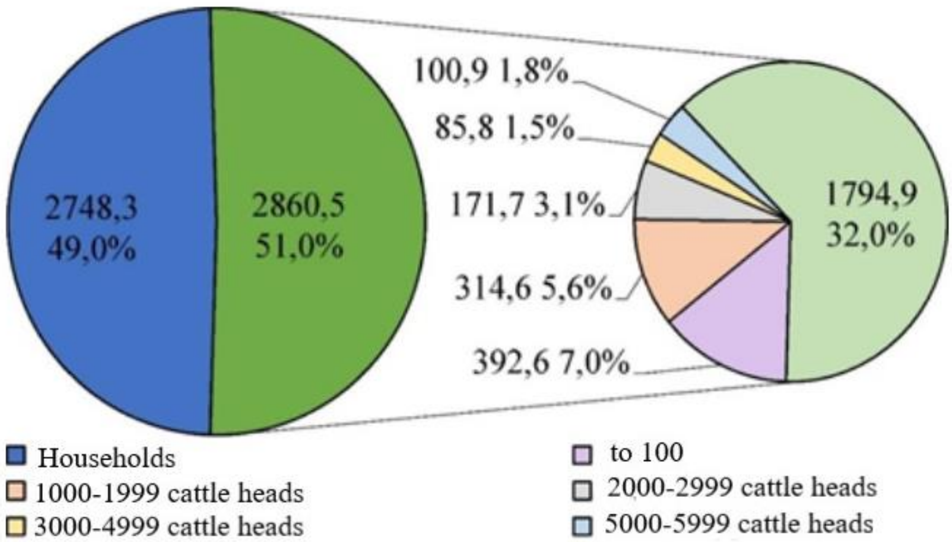


Fig. 1.4. Pig population distribution by categories and types of farms

As for the latter, most of the herd is concentrated in enterprises with a herd of more than 6,000 heads (32%). It is at these enterprises that the second and third category of technologies are used, while the second category prevails. On farms with livestock from 1 to 6 thousand heads (12% of the total number of livestock), the first three categories of technological solutions are used, while traditional technologies prevail for farms with smaller livestock. On farms, up to 1,000 heads (7%) use traditional technologies.

Currently, according to the regulatory framework of Ukraine regarding the organization of pork production, pig keeping technologies are distinguished by systems and methods of keeping. The standards of technological design provide for two systems of keeping pigs: walking and non-walking. The walking system is subdivided into a stand-walking system, a free-walking system, and a camp system. The no-walk system for keeping pigs also has its own options: floor-stand, cage-battery, tiered, container, conveyor. Depending on the system and method of keeping, the volume and planning solutions of premises (pig houses), methods and means of keeping pigs and mechanization of production processes will differ [3, 20, 21].

In addition to systems and methods, retention technologies can also be distinguished by the following features [5, 22]:

– *the presence of bedding*: changeable, deep, without bedding;

– *floor view*: continuous, partially or completely fissured;

– *type of fodder*: wet or liquid mixtures, dry bulk (mixed feed);

– *method of manure removal*: mechanical (scraper conveyors, scraper installations, bulldozer), various methods of hydraulic removal;

– *type of pigsties*: capital, pig houses, hangars (cold storage), etc.

From the point of view *biological component of technology*, its characteristics can be distinguished: reproduction method (tour or year-round uniform (flow)) and cultivation method (single-phase, two-phase and three-phase) [5].

If we summarize technological solutions for pork production in Ukraine, we can distinguish four main groups [5]:

– *traditional technologies* – are based on volume-planning solutions of pig farms, which are based on typical projects of Soviet times. At the same time, the maintenance system can be walking or non-walking, the method of maintenance is floor-stand, the method of reproduction is year-round-uniform, the method of cultivation is mainly three-phase. Keeping livestock of all technological groups on a continuous floor. Differences between different farms are caused, first of all, by the methods of distributing fodder and removing manure laid down in the typical design of the premises;

– *new (Western) technologies* – are characterized by the fact that all livestock are kept on a partially or completely split floor in capital premises, the premises are specialized for different technological groups, divided into isolated sections. At the same time, the maintenance system is no-walking, the method of maintenance is floor-stand, the method of reproduction is year-round-uniform, the method of cultivation is three-phase;

– *low-cost (alternative) technologies* – the most widespread in Ukraine was the group keeping of pigs in hangars (sometimes in capital premises) in large homogeneous groups on deep unchanging litter, feeding them with dry balanced feed to their heart's content with free access to water and the use of natural ventilation to regulate the microclimate. At the same time, the maintenance system is mostly non-walking, the method of maintenance is floor-stand, the method of reproduction is year-round and uniform, the method of cultivation is one- or two-phase;

– *private sector* – a feature of keeping pigs in auxiliary farms of the population, first of all, is a small herd of animals. This leads to the practically absence of means of mechanization of production processes. At the same time, the system of maintenance can be walking or non-walking, the method of maintenance is floor-stand, the method of reproduction is mainly tour, growing is one- or two-phase. Livestock are mostly kept on a solid floor, although there are examples of the use of partially and completely slatted floors.

The most common method of industrial pork production in the world [23, 24] is keeping pigs on a fully or partially slatted floor (new (Western) technologies). The technological process of pork production, according to the specified technology, involves the implementation of a year-round three-phase system with a flow-rhythmic organization of work and a fixed production rhythm. This system is based on the maintenance of separate groups of animals in specialized sections in accordance with the «empty-busy» principle and implementation of differentiated feeding.

1.3 Analysis and classification of technological schemes of heat-utilizers for livestock premises

In order to choose the most suitable for working conditions in livestock premises from all known technological schemes of heat utilizers, it is advisable to perform their classification [25]. This approach will make it possible to exclude those common features that are characteristic of this type and to focus attention on distinguishing features when considering each scheme. In addition, the classification will allow to generalize the physical properties of work for this type of recycler [3].

There are several ways of classifying exhaust air heat utilizers. We will use the traditional classification scheme, in accordance with which the recyclers are divided into three groups [26]:

- heat exchangers with an intermediate coolant;
- recuperators;
- regenerators.

The main feature of heat exchangers with an intermediate heat carrier is the presence of a circulation circuit with or without a pump, which ensures the transfer of thermal energy of the exhaust air to the supply air (Fig. 1.5) [3].

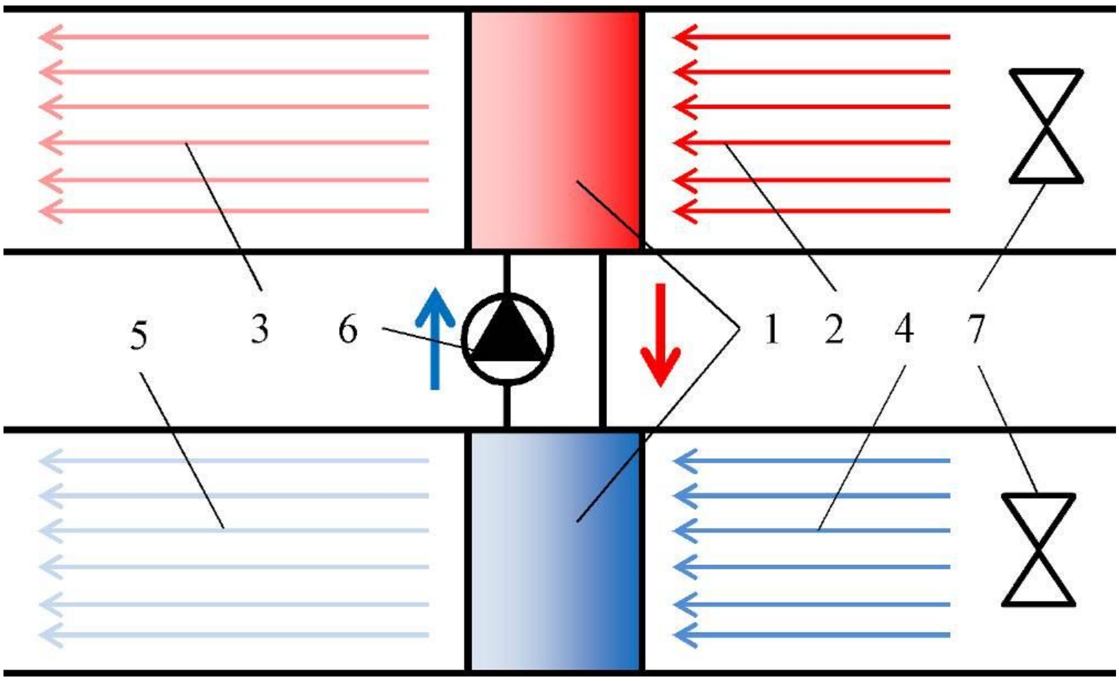


Fig. 1.5. Technological diagram of a heat exchanger with an intermediate coolant: 1 – heat exchange surface; directions of air movement: 2 - from the room; 3 – to the external environment; 4 – from the external environment; 5 – indoors; 6 – pump; 7 – fan

Heat exchangers with an intermediate coolant can be of the recuperative or contact type [27]. In the latter case, the intermediate coolant comes into direct contact with the heat exchange media. Options are also possible, when the coolant is in direct contact with the medium in one channel, and a recuperative heat exchanger is used in the other channel.

Water or another liquid that does not freeze in the working temperature range (aqueous solutions of salts, glycols, freons, etc.) can be used as an intermediate coolant. The circulation circuit of the systems can be open or closed.

Heat pipe heat exchangers represent a special group of heat exchangers with an intermediate coolant. A heat pipe is a chamber in which there is a liquid. One end of the pipe is washed with warm air, the other with cold. Under the influence of the temperature difference at the ends of the tube there is evaporation and condensation of the working fluid, which provides heat transfer. Heat exchangers made of heat pipes can be of the gravity type with the return of the working fluid by gravity (thermosiphons) and wick type, in which the capillary effect occurs [28].

This method of heat utilization ensures air exchange of rooms without mixing of heat exchange flows, which excludes the possibility of harmful impurities from the exhaust air entering the supply air, and is recommended for use in the presence of technical and economic justification for any type of premises, especially when the supply and exhaust channels are significantly distant [29].

The limitation of the use of heat exchangers with an intermediate transmission medium is due to the need to use pumps for the circulation of the coolant, which are characterized by low reliability and require additional electricity costs. In addition, the heat exchange blocks are made of metals, which in the aggressive environment of livestock and poultry premises due to corrosion lose their tightness quite quickly and fail, and therefore have a service life of no more than 3-4 years. The efficiency of the use of these types of utilizers is 60-70% [30].

Switchable regenerators are used in exhaust air heat recovery units (Fig. 1.6). In such recyclers, the nozzle is stationary and is successively washed with warm and cold air. Foil, technical cardboard, gravel, etc. are used as nozzle material.

Rotating regenerative heat exchangers are known, in which heat transfer is carried out by an accumulative mass that passes successively through streams of cooling air and that which is heated [31, 32]. Regenerative rotary heat exchangers are non-sorption and sorption (enthalpic).

In sorption regenerators, the heat exchange surface is covered with a layer of sorbent (lithium chloride, etc.), which absorbs moisture from the exhaust air and transfers it as a result of the desorption process of the supply air.

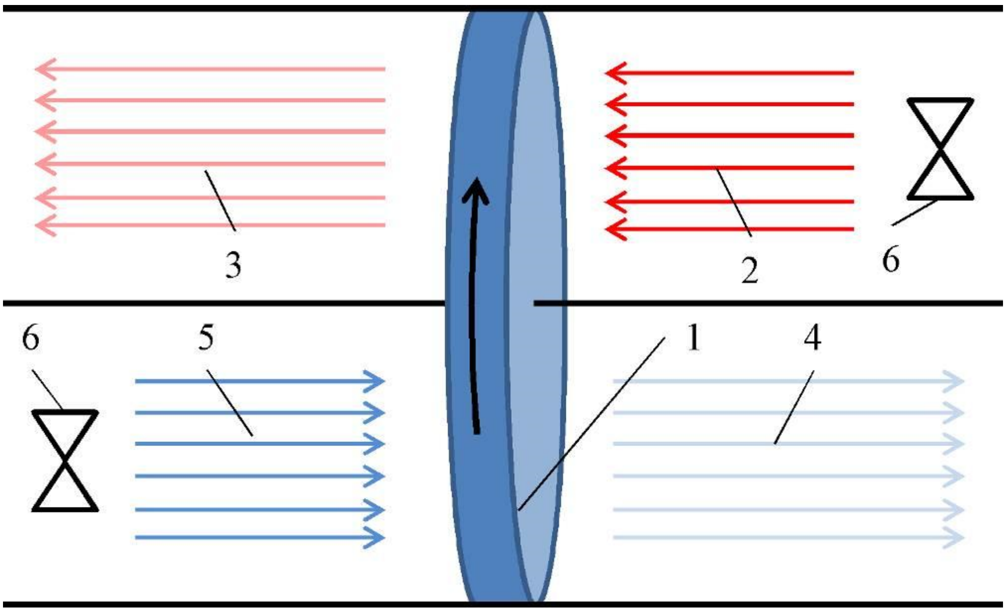


Fig. 1.6. Technological scheme of the regenerator utilizer:
1 – heat exchange regenerative surface; directions of air movement:
2 – from the premises; 3 – to the external environment; 4 – indoors;
5 – from the external environment; 6 – fan

Regenerative rotary heat exchangers with small overall dimensions have a heat utilization efficiency of up to 75-85% and are able to pass significant volumes of air [43]. Such heat recovery devices are used to recover the heat of ventilation emissions both at industrial facilities and in air conditioning systems of residential and office premises.

The rotors of such heat exchangers are made of anti-corrosion steel and have a high cost and metal content. The main disadvantage of regenerative heat exchangers is that the condensate formed in the process of heat exchange, together with pathogenic bacteria and fungi formed on the moistened surface, is transferred to the supply air

[41]. In addition, with this method of heat utilization, partial recirculation of polluted exhaust air occurs [33]. These factors make it necessary to increase the air exchange almost twice. Therefore, in rooms with high pollution and air humidity, which include livestock and poultry rooms, for disposal heat of ventilation emissions, it is impractical to use regenerative rotary heat exchangers. In addition, the technical and economic analysis [34] shows that the use of regenerative heat exchangers in livestock and poultry premises is not justified due to the need to increase air exchange.

In recuperative heat exchangers, heat transfer between air flows is carried out through the wall that separates them.

Many variants of structural solutions of recuperators are used, depending on the conditions of their operation. Fig. 1.7 shows a diagram of one of the types with a cross-current system of coolant movement through slotted channels. Such heat exchange is stationary and continuous, the structure is recuperative of heat exchangers is simple, there is no collision of supply and exhaust air flows, dry heating of supply air takes place, which is especially important for livestock premises saturated with moisture, and also important when using heat exchangers in rooms with increased requirements for air environment parameters, mixing of flows and partial return are not allowed exhaust air into the room [35].

According to the material from which they are made, recuperators can be metal, glass, cardboard, plastic, etc.

According to the direction of air flow, recuperative lamellar direct-surface heat exchangers are divided into direct-flow, counter-flow and cross-flow.

The advantages of recuperative direct-surface heat exchangers include a relatively large area of the heat exchange surface with small overall dimensions, high thermal performance indicators, and the simplicity of construction makes them reliable and durable. The efficiency of heat utilization in recuperative plate heat exchangers reaches 80-90% [42].

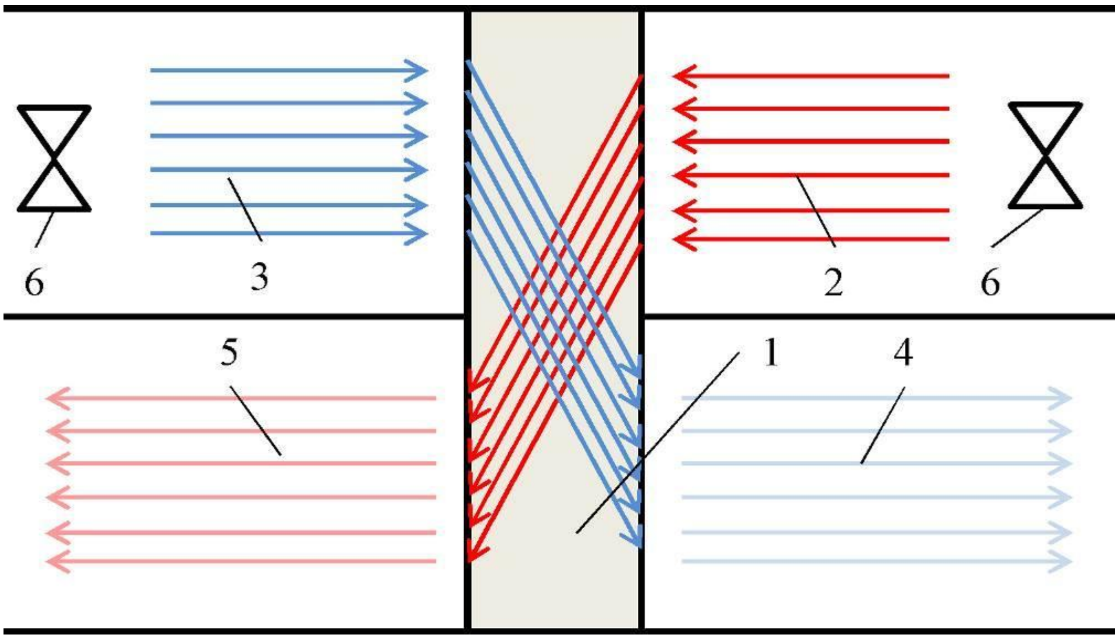


Fig. 1.7. Technological scheme of the recuperative utilizer:
1 – heat exchange regenerative surface; directions of air movement:
2 – from the premises; 3 – from the external environment; 4 – indoors;
5 – to the external environment; 6 – fan

Therefore, for the utilization of heat in the systems of ensuring the regulatory parameters of the air environment of livestock premises, the most promising are recuperative heat exchangers made of polymer materials that are resistant to moisture and chemically active impurities.

1.4 Analysis of designs of recuperative heat utilizers for livestock premises

The recuperative heat exchanger is a surface-type heat exchanger to preserve the heat of the removed gas flows. Unlike the regenerator, the heat exchange in it occurs through a stationary heat exchange surface and the directions of movement of heat carriers do not change [36].

In addition to the heat exchanger, the recuperative heat exchanger often includes two fans for exhaust and supply air flows. Various devices are often included in the design to automate its operation and improve the quality of supply air [37].

Recuperative heat-utilizers are classified according to a number of features [38]: by purpose - with heating of air, gas, liquid; according to the scheme of movement of coolants - direct-flow, counter-flow and cross-flow; by design: lamellar, tubular, ribbed and lamellar-ribbed, etc.

The main criteria for choosing recuperative heat-utilizers are [39, 40]:

- the coefficient of thermal efficiency, which is determined by the formula [41]:

$$\eta_t = \frac{T_{k2} - T_{n2}}{T_{n1} - T_{n2}}, \qquad T_{n1} - T_{k1} = T_{k2} - T_{n2}, \tag{1.1}$$

T_{k2} – the temperature of the supply air at the exit from the heat recovery unit, ºC;

T_{n2} – supply air temperature at the entrance to the heat recovery unit (external air), ºC;

T_{n1} – exhaust air temperature at the entrance to the heat recovery unit (indoors), ºC;

T_{k1} – exhaust air temperature at the exit from the heat recovery unit (exhaust air), ºC;

- sanitary and hygienic indicators: there should be no pollution passing through the recuperator;

- energy efficiency: this value characterizes the specific energy consumption, that is, how much the recuperative heat exchanger consumes to return a unit of heat from the removed air;

- operational characteristics: the structure must be repairable, have a long service life, require minimal maintenance.

- construction cost.

Having compared these parameters, it is possible to analyze the designs of recuperative heat-utilizers for livestock premises.

Plate heat exchangers have become very widespread, thanks to their compact design, the possibility of quick assembly and modernization, simplicity and instant cleaning of impurities. The main elements that make up the collapsible plate heat exchangers (Fig. 1.8) are working plates separated by rubber spacers, end chambers with nozzles, a frame and fastening bolts. Thin sheet steel (0.5-0.6 mm) is used for the production of plates, which for the flow part is made with a corrugated surface, due to which the heat exchange surface and the activity of flow turbulence increase significantly [42, 43].

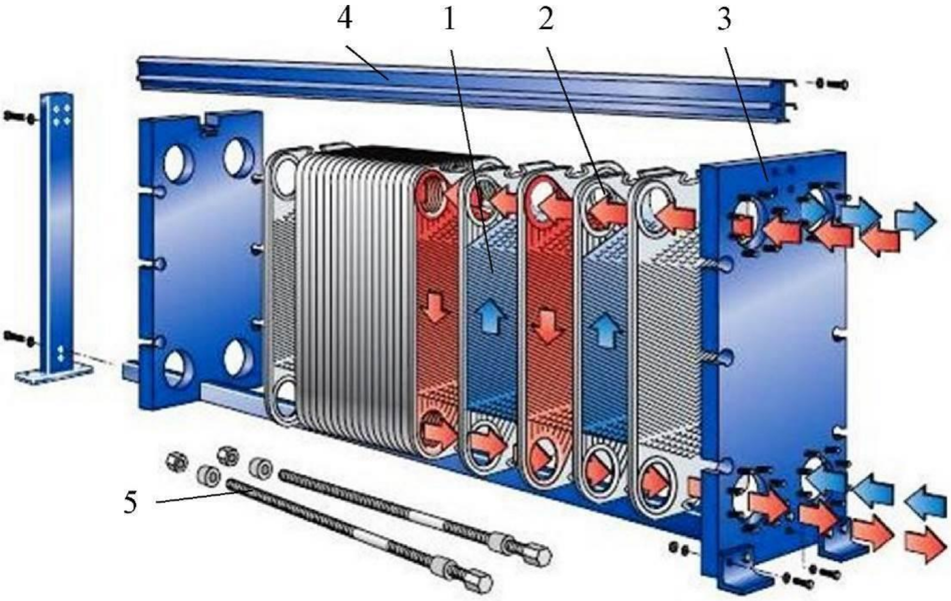


Fig. 1.8. Design schemes of the plate heat exchanger:
1 – working plates; 2 – rubber gaskets;
3 – final chambers with nozzles; 4 – frame; 5 - tie bolts

The maximum temperature of the heat carrier in plate-type collapsible heat exchangers is about 150 °C at a pressure of 2.5 MPa. Due to the large heat exchange surface (20-30 sheets) and the small thickness of one sheet, a high heat transfer coefficient is achieved.

Today, plate recuperative heat exchangers are the most widely used to ensure air exchange in rooms.

The design of plate recuperators (Fig. 1.9) is a set of special aluminum plates with a thickness of 0.1-0.5 mm, which represent the heat exchange surface of the recuperator.

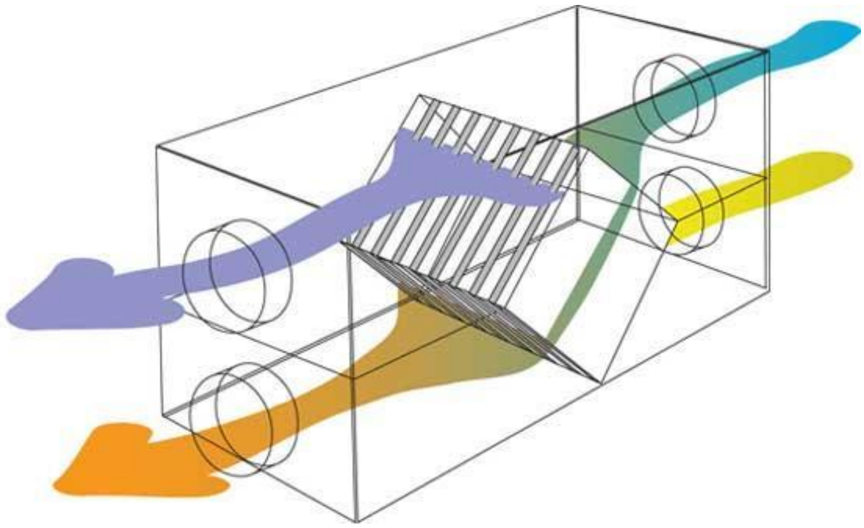


Fig. 1.9. Design scheme of lamellar regenerative heat recovery unit

To increase efficiency, as well as to obtain the best aerodynamic characteristics, recuperator plates have their own specific structure and geometry.

In plate recuperators, there is another most important parameter that largely affects efficiency and aerodynamic characteristics. This is the distance between the plates, which is from 5 to 12 mm. This is due to the optimal combination of two main indicators of efficiency and resistance.

Many companies [44, 45] produce plate recuperators for rectangular (Fig. 1.10, a) and round (Fig. 1.10, b) channels.

a b

Fig. 1.10. Lamellar recuperative heat exchangers
for rectangular (a) and round (b) channels

Plate heat exchangers are used: in municipal energy; in heating stations; in ventilation and air conditioning of buildings.

Advantages of plate heat exchangers: small areas occupied by heat exchange equipment; the possibility of working at low temperature pressures (minimum temperature difference between the heating surface and the one being heated); slow growth of deposits; low pressure losses (reduction of electricity consumption for electric pumps); low labor costs (terms) during repair and maintenance.

Disadvantages of plate heat exchangers: relatively high cost; expensive equipment for maintenance and repair; qualified service personnel.

Plate-ribbed heat exchangers (Fig. 1.11), in contrast to shell-and-tube ones, are among the most compact devices due to the developed heat exchange surface in a limited volume. Plate-ribbed heat exchangers are produced with ribs of different configurations. The most common ribbed surfaces of heat exchange equipment form triangular and rectangular channels for the flow of heat carriers [46].

Plate-ribbed heat exchangers are widely used in drying plants, heating systems, as economizers and air cooling devices [47, 48].

Fig. 1.11. The general appearance of the finned-plate heat exchanger

Advantages of plate-fin heat exchangers: high efficiency of heat exchange per unit surface; rigid construction.

Disadvantages of plate-fin heat exchangers: high construction cost; insignificant main heat exchange surface; a metal with a high coefficient of thermal conductivity is required.

Shell-plate heat exchangers (Fig. 1.12) are a welded package of plates placed in a cylindrical body. The principle of operation is similar to plate heat exchangers. One medium moves between the corrugated plates, and the second medium in the space between the plates and the body [49].

Fig. 1.12. The general view of the shell-plate heat exchanger

The pressure and temperature of water vapor in collapsible plate heat exchangers are limited by the materials of the gaskets to 150-190 °C. The material of the gaskets of heat-utilizers also imposes restrictions on the use of working environments, such as acids, alkalis, etc.

Shell-plate heat exchangers are widely used in: the oil industry; chemical industry; heating points; ventilation and air conditioning; refrigeration industry.

Advantages of shell plate heat exchangers: reliability; compactness; high heat transfer coefficient; resistance to high temperatures (900°C) and pressure (140 bar).

Disadvantages of shell-plate heat exchangers: the impossibility of disassembling the heat exchanger on the side of the plate package, this space is available only for non-disassembly washing with chemical reagents.

In twisted heat exchangers (Fig. 1.13), heat spreads not only through the tube, but also through the intertube space. The design of the heating surface of this type of heat exchanger includes a number of concentric coils, which are enclosed in a casing [50].

Fig. 1.13. General view of a twisted heat exchanger

Twisted heat exchangers are widely used in high-pressure equipment for processes of separation of gas mixtures by the method of deep cooling. These heat exchangers are characterized by the ability to self-compensate.

Advantages of twisted heat exchangers: susceptible to high temperatures and pressure; resistant to deformation.

Disadvantages of twisted heat exchangers: low heat output.

The spiral heat exchanger (Fig. 1.14) is two spiral channels wound from rolled material around the central partition. One of the purposes of spiral heat exchangers is heating and cooling of highly viscous media. The working media of this thermal equipment are liquid-liquid, gas-liquid, gas-gas, liquid-vapor, gas-vapor [51].

Fig. 1.14. General view of the spiral heat exchanger

Field of application: oil and gas industry (heavy oils, oil, gasoline fractions); chemical industry (heat recovery, PVC, acrylic, evaporators); metallurgical industry, coking plants (washing oils, benzenes); mining industry (suspensions, magnesium oxides); food industry (raw sugar juice, alcohol production processes); wastewater treatment and waste water heat recovery.

Advantages of spiral heat exchangers: compact size; insignificant hydraulic resistance; high intensity of heat exchange.

Disadvantages of spiral heat exchangers: manufacturing complexity; service and repair; unacceptable at high pressures.

In shell-and-tube heat exchangers, the main elements are (Fig. 1.15): the body, bundles of small-diameter pipes, pipe grates, nozzles, covers, stress compensation elements.

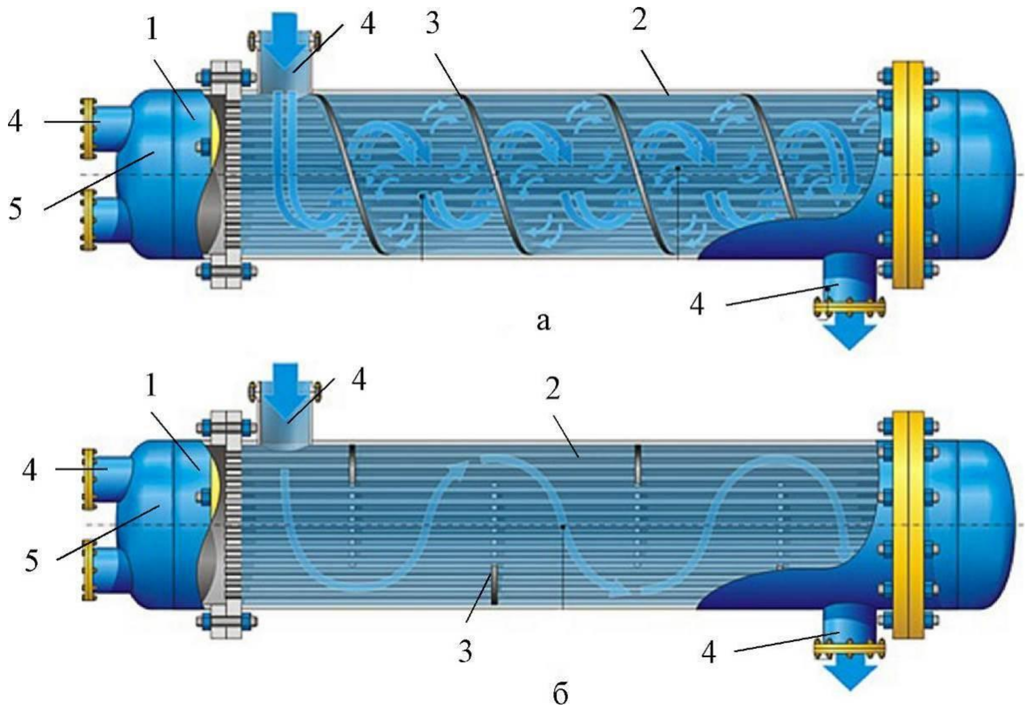


Fig. 1.15. Design schemes of spiral (a) and multi-pass (b) shell-and-tube heat exchanger: 1 – body; 2 – bundles of pipes of small diameter; 3 – pipe grates; 4 – nozzles; 5 – covers

Heat is transferred through the walls of the tubes from medium to medium, one of which circulates inside the tubes, and the other washes them outside. To increase the heat transfer coefficient in shell-and-tube heat exchangers, the direction of movement of the external environment is changed several times with the help of partitions; such a heat exchanger is called multi-pass. Inside the tubes, the speed of movement of the medium and the heat transfer coefficient can be increased with the help of special devices that change the flow direction. Depending on the field of application, these heat exchangers are horizontal, vertical or inclined. Shell-and-tube heat exchangers are used for heat exchange and thermochemical processes between various liquids, vapors and gases - both without change and with a change in their aggregate state, as condensers, heaters and evaporators [52].

Shell-and-tube heat exchangers are used in: thermal energy, oil, gas, chemical and food industries [53].

The general appearance of the shell-and-tube heat exchanger is presented in Fig. 1.16.

Fig. 1.16. General view of the shell-and-tube heat exchanger

One of the types of shell-and-tube heat exchanger is the «pipe-in-pipe» type heat exchanger, the design of which is shown in Fig. 1.17. Individual elements are connected to each other by nozzles and fittings, forming a complete device of the required size. These heat-utilizers were used at low flow rates of coolants and at high pressures [54].

Fig. 1.17. Heat exchanger type «pipe in a pipe»

Advantages of shell-and-tube heat exchangers: the widest range of applications in terms of operating parameters; the lowest requirements for environmental cleanliness; high resistance to pneumatic shocks; relative simplicity of design.

Disadvantages of shell-and-tube heat exchangers: temperature deformations; relatively low heat transfer coefficient.

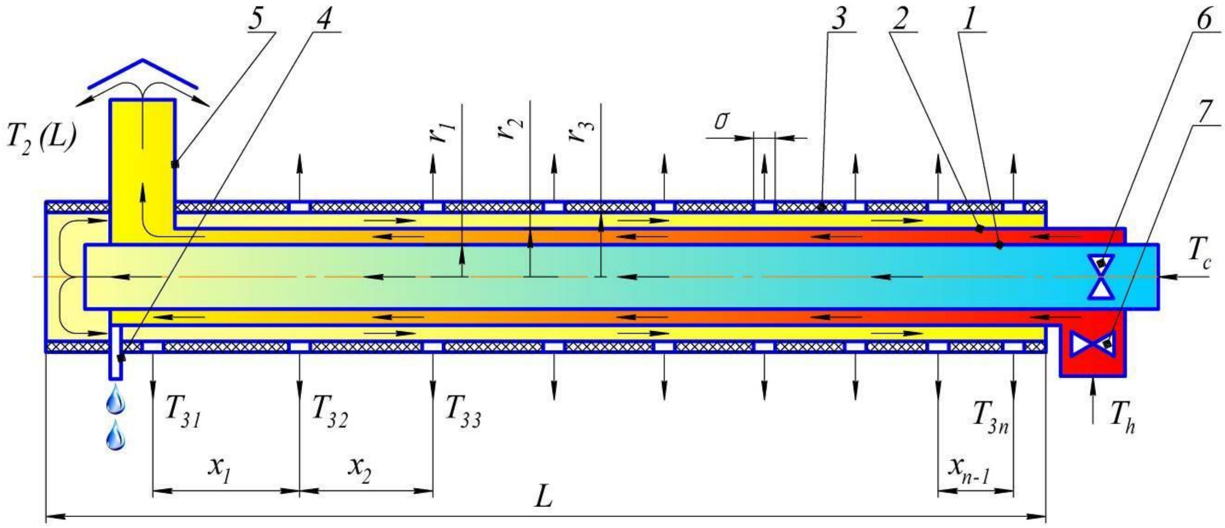
Taking into account the technological conditions of the air in livestock premises (significant dustiness - up to 6 mg/m^3; high humidity - up to 80% [55]; the presence of a high concentration of aggressive components - ammonia to 20 mg/m^3, hydrogen sulfide - up to 10 mg/m^3, of carbon dioxide - up to 0.28% [90]) and the results of the analysis of the designs of recuperative heat utilizers revealed that, in terms of sanitary and hygienic and operational indicators, high energy efficiency and low cost of construction, the most suitable for the ventilation system are shell-and-tube heat utilizers of the «pipe-in-pipe» type [4].

1.5 Technological diagram of a heat-utilizer for livestock premises

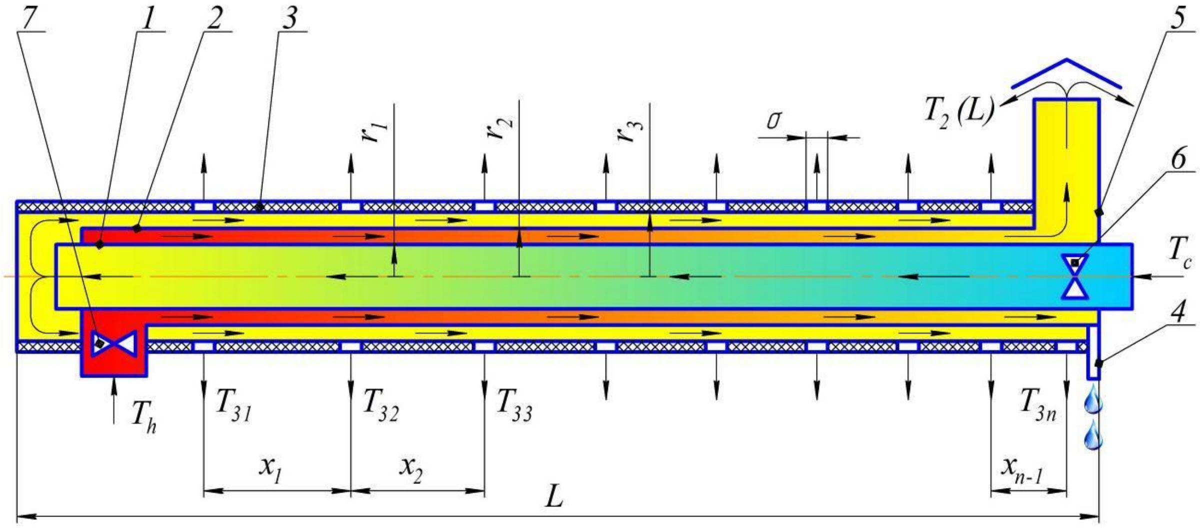
To ensure the microclimate in livestock premises, based on the specified technological conditions and the analysis of the designs of heat-utilizers, two structural and technological schemes of a three-pipe concentric heat-utilizer were developed, which differ in the directions of air flow: forward flow (Fig. 1.18, a) and counterflow [1, 6, 9] (Fig. 1.18, b).

The technological schemes of the three-pipe heat exchanger with direct and counterflow include pipes 1, 2 and 3, which are installed coaxially, a pipe for draining condensate 4, which passes through pipe 3 and is located in the lower part of pipe 2, an exhaust shaft 5, which passes through pipe 3, supply 6 and exhaust 7 fans.

The heat exchanger carries out the technological process as follows. The supply (cold) air is supplied by the fan 6 through the internal pipe 1. The exhaust (warm) air from the room is pumped by the fan 7 into the space between the pipes 1 and 2, which has an annular cross-section.



a



b

Fig. 1.18. Technological diagram of a three-pipe heat exchanger with the main parameters: 1, 2, 3 – pipes; 4 – tube for draining condensate; 5 – exhaust shaft; 6 – supply fan; 7 – exhaust fan

With a technological scheme with direct flow, both flows move in one direction to the exhaust shaft 5, from which the exhaust air exits into the external environment, and the supply air turns around and continues to move in the opposite direction in the space between pipes 2 and 3, which also has an annular cross section.

With a technological scheme with counterflow, the flows move in the opposite direction: the exhaust air leaves the external environment from the exhaust shaft 5, and the supply air turns around and continues to move in the opposite direction in the space between the pipes 2 and 3, which also has an annular cross section.

In this way, the process of heat exchange between supply and exhaust air occurs through the walls of pipes 1 and 2, thanks to which the supply air is heated by a certain amount.

When the exhaust air cools, condensate forms on the outer surface of pipe 1 and the inner surface of pipe 2, for the removal of which pipe 4 serves.

To exclude air cooling in the room, the surface of the outer pipe 3 is thermally insulated.

In the course of research, it is necessary to identify the most effective structural and technological scheme of a three-pipe heat recovery unit (direct flow, counterflow) and justify its following design parameters:

- pipe radii r_1, r_2, r_3;
- the length of the external pipe of the heat exchanger L;
- material and wall thickness of the heat recovery pipe;
- the area of the openings for the supply air outlet σ, their number is n, the pitch of the holes x.

1.6 Analysis of the results of theoretical studies of the heat transfer process in concentric heat exchangers

Many domestic and foreign works [56, 57, 58, 59, 60] are devoted to the study of the process of heat transfer in heat-utilizers. We will analyze those works in which concentric heat exchangers are considered.

In works [61, 62], the method of calculating heat exchange and hydrodynamics for practical problems of natural circulation heat exchange of heated pipes is described.

In [95], the dimensionless dependences for calculating the heat transfer and fluid flow inside the heat exchanger tubes with an exhaust shaft are specified. The disadvantage of the proposed method is the narrow scope of application - for two-pipe heat exchangers.

On the basis of a modification of the implicit numerical method, in [63] an improved method for determining the indicators of non-stationary processes of tubular heat exchangers with cross-flow of heat carriers was developed:

- for the coolant

$$t_{1,i,j}^{k+1} = \left(t_{1,i,j}^{k} + A_1 \frac{\Delta\tau}{\Delta x} t_{1,i-1,j}^{k+1} + B_1 \Delta\tau t_{i,j}^{k}\right)\left(1 + A_1 \frac{\Delta\tau}{\Delta x} + B_1 \Delta\tau\right)^{-1}, \quad 0 < i \le N_1, \quad j = 1,2,\dots,N_2; \tag{1.2}$$

- for the pipe wall

$$t_{i,j}^{k+1} = \left(t_{i,j}^{k} + C_1^* \Delta\tau t_{1,i,j}^{k+1} + C_2^* \Delta\tau t_{2,i,j}^{k}\right)\left(1 + C_1^* \Delta\tau + C_2^* \Delta\tau\right)^{-1}, \quad 0 < i \le N_1, \quad j = 1,2,\dots,N_2; \tag{1.3}$$

Theoretical studies [64] present general information about heat exchange processes in heat exchange devices. The ideal mixing model is adopted as the flow model.

In [105], a mathematical model of the «stirring-stirring» type heat exchanger and its corresponding transient processes according to the diagram in Fig. 1.19 was obtained.

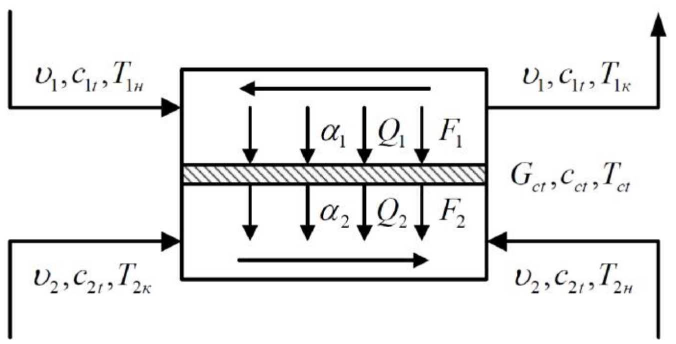


Fig. 1.19. Equivalent circuit of the heat exchanger according to research:
Q – the amount of heat transferred through the heat exchange surface;
α – heat transfer coefficient; G, c, T – weight, heat capacity and wall temperature

The paper [65] presents a ratio for calculating the formation of deposits on the heat transfer surface of heat exchangers. The proposed dependence takes into account the influence of the contaminant concentration, solution speed, wall temperature, pressure and plate parameters, on the intensity of the appearance and growth of deposits, which in turn affects the process of heat transfer between the walls of the heat exchanger.

In studies [66, 67], the problem of the movement of a liquid heat carrier through a heat exchanger was solved, the equivalent scheme of which is presented in Fig. 1.20. In these studies, the equations of the heat balance of the heat exchanger were obtained and the dependences of the temperature change of the coolant during its movement were determined.

Fig. 1.20. Equivalent circuit of the heat exchanger according to research

According to studies [110], the dependences of the temperature distribution on the length of the heat exchanger were obtained:

$$T_{h(x)} = T_{h(in)} - \frac{U_1P_1}{C_h}\left[\frac{G_5}{\lambda_3}\left(e^{\lambda_3 x}-1\right)+\frac{G_6}{\lambda_4}\left(e^{\lambda_4 x}-1\right)\right]- \\ -\frac{U_2P_2}{C_h}\left[\frac{G_7}{\lambda_3}\left(e^{\lambda_3 x}-1\right)+\frac{G_8}{\lambda_4}\left(e^{\lambda_4 x}-1\right)\right] \quad (1.4)$$

$$T_{c1(x)} = T_{h(in)} - \frac{U_1P_1}{C_h}\left[\frac{G_5}{\lambda_3}\left(e^{\lambda_3 x}-1\right)+\frac{G_6}{\lambda_4}\left(e^{\lambda_4 x}-1\right)\right]- \\ -\frac{U_2P_2}{C_h}\left[\frac{G_7}{\lambda_3}\left(e^{\lambda_3 x}-1\right)+\frac{G_8}{\lambda_4}\left(e^{\lambda_4 x}-1\right)\right]-\left(G_5e^{\lambda_3 x}+G_6e^{\lambda_4 x}\right) \quad (1.5)$$

$$T_{c2(x)} = T_{h(in)} - \frac{U_1 P_1}{C_h}\left[\frac{G_5}{\lambda_3}\left(e^{\lambda_3 x} - 1\right) + \frac{G_6}{\lambda_4}\left(e^{\lambda_4 x} - 1\right)\right] - \\ - \frac{U_2 P_2}{C_h}\left[\frac{G_7}{\lambda_3}\left(e^{\lambda_3 x} - 1\right) + \frac{G_8}{\lambda_4}\left(e^{\lambda_4 x} - 1\right)\right] - \left(G_7 e^{\lambda_3 x} + G_8 e^{\lambda_4 x}\right) \qquad (1.6)$$

U –overall heat transfer coefficient;

P – average perimeter;

C – heat power factor;

G –constants of integration;

λ – root of the equation;

T – temperature.

In works [68, 69], the distribution of heat energy in U-shaped heat exchangers with a liquid coolant is determined (Fig. 1.21, a). The temperature distribution along the length of the heat exchanger pipe is established (Fig. 1.21, b).

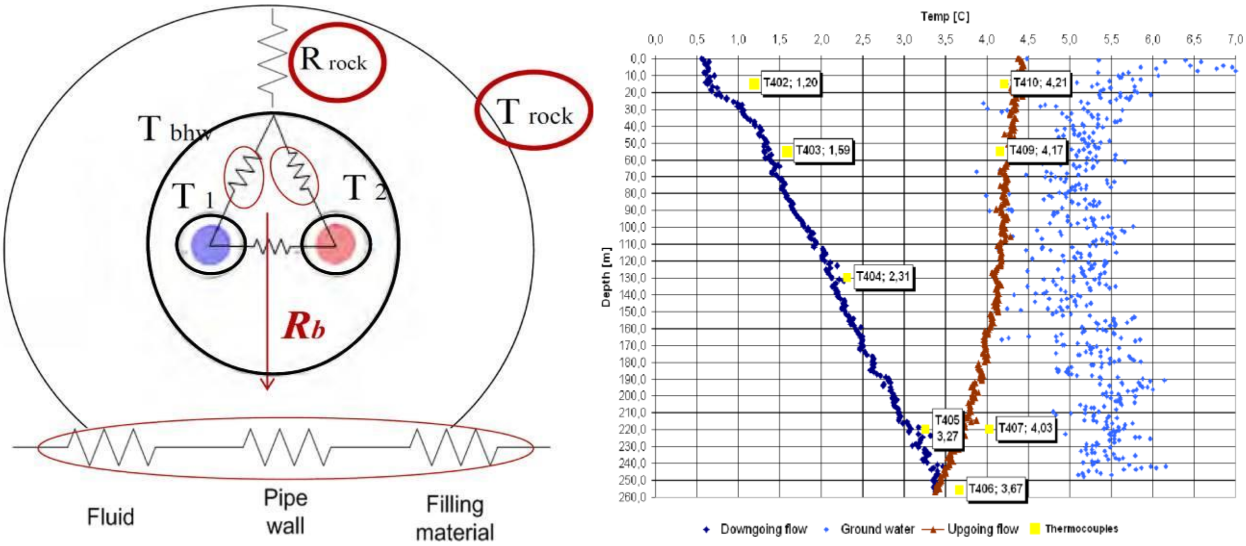


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Fig. 1.21. Equivalent diagram (a) and temperature distribution (b) of the heat exchanger U-shaped with a liquid coolant according to research [95]

The paper [70] proposes an algorithm for calculating heat transfer, which is used in studies of a concentric tube heat exchanger. Research is limited to determining heat transfer coefficients based on experimental results. The proposed algorithm makes it possible to obtain a correlation of heat transfer coefficients for a liquid that circulates through the internal annular space in the transitional flow regime (Fig. 1.22):

$$\alpha_H = 2{,}718 \cdot w_H^{0{,}597} \cdot \rho_H^{0{,}597} \cdot c_{p,H}^{1/3} \cdot \left(\frac{d_{h2}}{\mu_H}\right)^{0{,}264} \cdot \left(\frac{\lambda_H}{L_1}\right)^{0{,}667} \tag{1.7}$$

ρ – heat carrier density;

μ – dynamic viscosity of the coolant;

λ – thermal conductivity;

α – heat transfer coefficient;

w – average speed of the coolant;

c_p –specific heat capacity;

L – the length of the heat exchanger;

d – equivalent diameter of the heat exchanger.

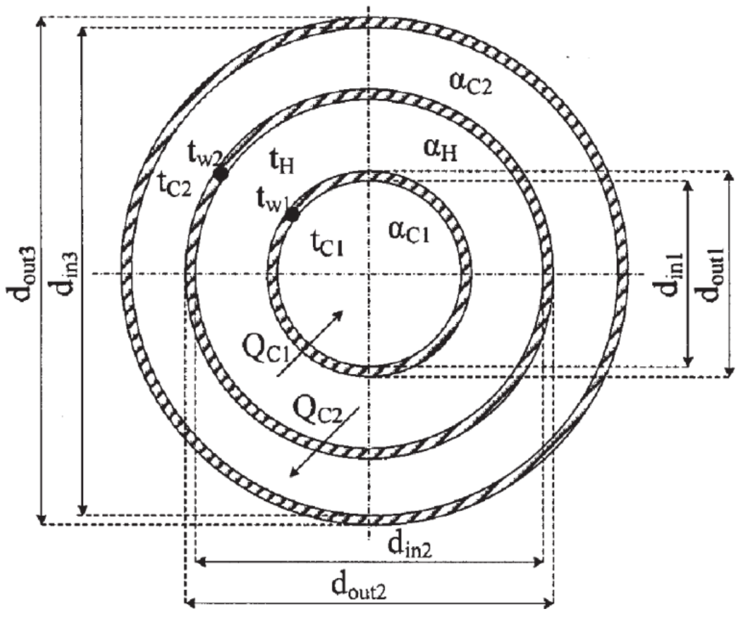

Fig. 1.22. Equivalent diagram of a concentric pipe heat exchanger according to research

Research [71] established a dependence for determining the heat transfer coefficient for a concentric three-pipe system with pipes made of different materials (Fig. 1.23):

$$\frac{1}{U_i} = r_i \left[\frac{1}{k_a r_{ma} / (r_2 - r_1)} + \frac{1}{k_b r_{mb} / (r_3 - r_2)} + \frac{1}{k_c r_{mc} / (r_4 - r_3)} \right], \qquad (1.8)$$

U – heat transfer coefficient;

r – pipe radius;

k –thermal conductivity coefficient.

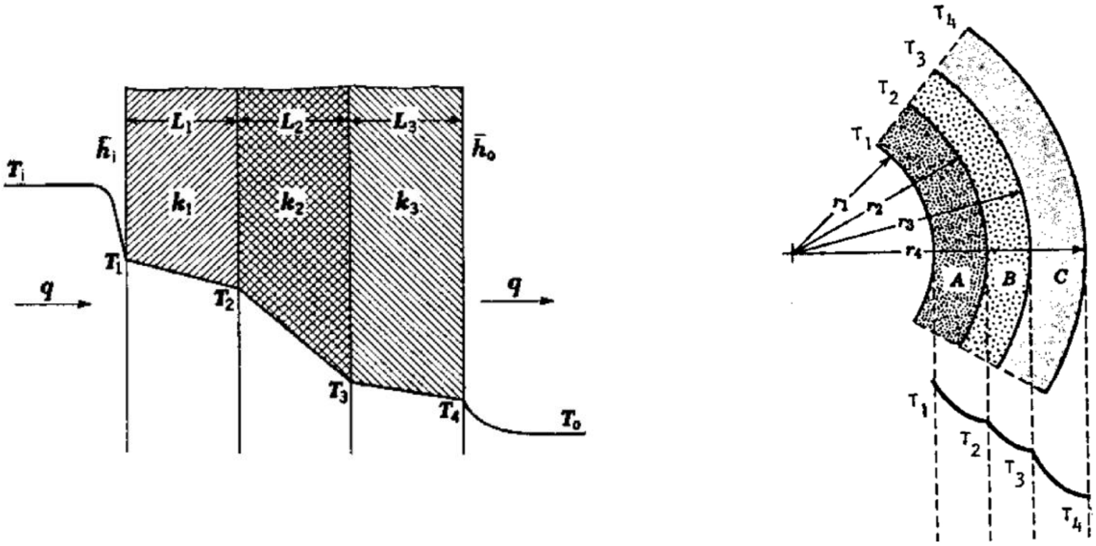

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Fig. 1.23. Calculation scheme of concentric tube heat exchanger according to research

In works [72, 73], a numerical model of the one-dimensional static process of heat and mass transfer in concentric heat exchangers was developed based on the laws of energy conservation. Based on the developed model, an algorithm was created for the software to calculate the temperature distribution in the heat exchanger. However, the specified algorithm is calculated only for liquid and gaseous coolants without taking into account the formation of condensate during cooling.

Theoretical studies [74] of a triple concentric tube heat exchanger with a liquid coolant have established that the ratio of pipe radii is the most important parameter affecting its efficiency. However, the proposed mathematical model does not take into account pneumatic resistance, which significantly affects the mass transfer process.

Thus, the analysis of the results of theoretical studies of the heat transfer process in concentric heat exchangers showed that the authors in their works mainly considered the movement of a liquid or gaseous heat carrier without taking into account the phenomenon of condensation of water vapor at chilled Also, the considered works did not take into account pressure losses during the movement of the coolant through the heat exchanger, which significantly affects the mass transfer process. In addition, in these works, the laminar flow mode of the heat carrier is mainly considered, which indicates the narrow limits of the application of the obtained mathematical models [5].

2 EXPERIMENTAL AND NUMERICAL RESEARCH EFFICIENCY OF MICROCLIMATE PROVISION SYSTEMS IN THE PREMISES

2.1 Methodology of experimental studies of microclimate support systems

The efficiency of removing pollutants from the area animals are kept is expressed by the efficiency of pollution removal (ξ) [14]. Value ξ is determined by local measurements of the concentrations of (gaseous) pollutants emitted in the piggery. Value ξ at an arbitrary point p at time t for any pollution is defined as:

$$\xi_{x,p,t} = \frac{C_{x,e,t} - C_{x,i,t}}{C_{x,p,t} - C_{x,i,t}} \tag{2.1}$$

$\xi_{x,p,t}$ – pollution removal efficiency at point p at time t for pollution x;

$C_{x,e,t}$ – the concentration of pollutant x in the outlet air at the moment of time t (mg/m^3);

$C_{x,p,t}$ – concentration of pollutant x at point p at time t (mg/m^3);

$C_{x,i,t}$ – concentration of pollutant x in the incoming air at time t (mg/m^3).

The efficiency of heat removal from the area animals stay can similarly be expressed through the efficiency of heat removal (ζ). Equation (1) can also be used to calculate ζ by replacing local concentrations with local temperature values.

The efficiency of removal of impurities and heat will be equal to unity in a perfectly mixed air space. In a room with ideal laminar flow from inlet to outlet and homogeneously distributed pollutant and heat sources, the pollutant and heat removal efficiency values decrease from infinity at the inlet to unity at the outlet.

However, in practice, no ventilation system is perfect laminar flow or perfect mixing. All ventilated air spaces exhibit temperature, humidity, pollutant and dust gradients due to airflow patterns and different source locations of pollutants, resulting in pollutant and heat removal efficiency values above or below unity. Value ξ higher units indicate that fresh air first enters the animal housing area and then passes through pollutant sources on its way to the exit, which should indicate effective displacement of air into the animal housing area. Value ξ below one indicates that the concentration of the pollutant in the area the animals are located exceeds the concentration of the pollutant in the outlet air. These lower values can occur when part of the fresh air is removed from the room without causing air displacement in the area the animals are, or when an arbitrary point p is close to the source of pollution. Low values ξ indicate a high level of pollutants.

Research is focused on the determined efficiency of removal of pollutants and heat removed in the area the animals stay. Effective removal of pollutants from the area animals stay ($\xi > 1$) is desirable for low ventilation rates to promote good air quality in the animal housing area and save heating energy as less ventilation air is required. Effective removal of heat from the area animals stay ($\zeta > 1$) is desirable at high ventilation rates, when the ventilation is mainly intended to control the temperature in the animal housing areas.

The contents were measured in three occupied rooms for growing pigs CO_2 and temperature to determine the values ξ and ζ. Experimental studies were carried out under production conditions at the pig farms of FG «Litagor» (Ukraine, Vinnytsia region, Kozyatinsky district, Mykolaivka village), LLC «Subekon» (Ukraine, Vinnytsia region, Tyvrivskyi district, Sutysky village), PSP «Agrofirma Napadivska» (Ukraine, Vinnytsia region, Vinnytsia district, Napadivka village).

The pigsties were built according to classical designs and had practically the same thermal insulation properties. The internal height of the walls of the rooms was 2.20–2.40 m, the highest point of the ceiling was at a height of 4.00–4.20 m. The plan and cross section of the three rooms are shown in fig. 2.1–2.3. In fig. 2.1–2.3 also marked sampling points. The premises generalize the three most common ventilation systems.

Option 1 (FG «Litagor») - the ventilation system of the ground channel (Fig. 2.1). There are 16 groups of 25–30 piglets for rearing in the room. The floor area was 1,08±0,01 m^2/the head, of which 92.8±0.1% is lattice, and 7.2±0.1% is solid. The enclosures were 5.00±0.01 m × 6.50±0.01 m. Heating was provided by heat exchangers with hot water near the outer wall in the premises. The ventilation air passes through the ground channel through 9 ground holes (0.30±0.01 m × 0.30±0.01 m), and then to two exhaust shafts (Ø 0,60±0,01 m), which are placed in the center of the room in the ceiling. Air consumption in exhaust shafts ranges from 3.5±0.1 to 27±0.1 m^3/hour per animal. The average air velocity in the channel ranged from 0.12 m/s to 1.13 m/s. Irrigation nozzles are used for cooling in the summer, which are placed in surface inflow channels, which provide finely dispersed water supply with a capacity of 10–14 kg/h.

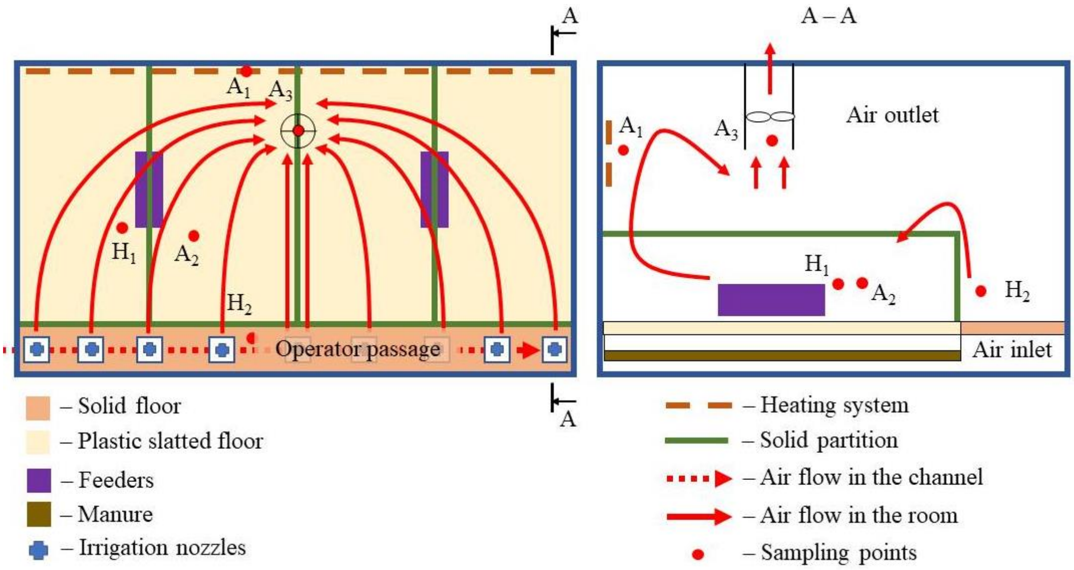


Fig. 2.1. Scheme of the ventilation system of the ground channel (option 1)

The design provides for fresh air to flow at a low speed through the inlet and fill the operator's passage. From there, the air flows through the front of the pen through the partition of the pen (height 1.10±0.01 m) to the animals. In the pen, air will flow to the middle part of the pen, providing fresh air to the area the animals stay and removing contaminants from it. The ventilation process is based on displacement and mixing of air.

Option 2 (PSP «Agrofirma Napadivska») - ceiling ventilation system (Fig. 2.2). There are 16 groups of 20-22 piglets for rearing in the room. The floor area was 0.90±0.01 m^2/head, of which 50.0±0.1% was lattice, and 50.0±0.1% was solid. The pens were 5.70±0.01 m × 3.00±0.01 m. Heating was provided by heat exchangers with hot water near the outer wall in the premises. Ventilation air passes through 32 ceiling supply channels (0.30±0.01 m × 0.50±0.01 m) and then to two exhaust shafts (Ø 0,65±0,01 м), which are placed in the center of the room in the ceiling. Air consumption in exhaust shafts ranges from 3.5±0.1 to 25±0.1 m^3/h per animal. Irrigation nozzles are used for cooling in the summer, which are placed in the wall inflow channels, which provide finely dispersed water supply with a capacity of 14–18 kg/h.

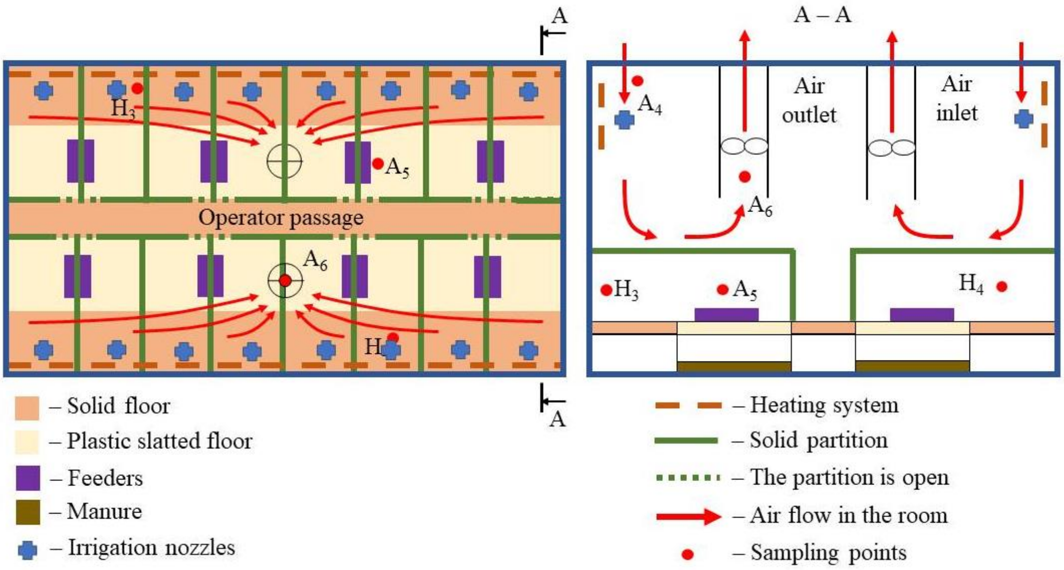


Fig. 2.2. Scheme of the ceiling ventilation system (option 2)

Option 3 («Subekon») - ventilation system through wall channels (Fig. 2.3). There are 8 groups of 24–26 piglets for rearing (200 heads). The floor area was 1.08±0.01 m^2/head, of which 94.3±0.1% was lattice, and 5.7±0.1% was solid. The enclosures had dimensions of 4.5±0.01 m × 5.5±0.01 m. Heating was provided by heat exchangers with hot water near the outer wall in the premises. The ventilation air passes through 12 wall supply channels (0.35±0.01 m × 1.00±0.01 m), and then to three exhaust shafts (Ø 0.65±0.01 m), which are located in the center room in the ceiling. Air consumption in exhaust shafts ranges from 3.5±0.1 to 25±0.1 m^3/h per animal. Irrigation nozzles are used for cooling in the summer, which are placed in the wall inflow channels, which provide finely dispersed water supply with a capacity of 14–18 kg/h.

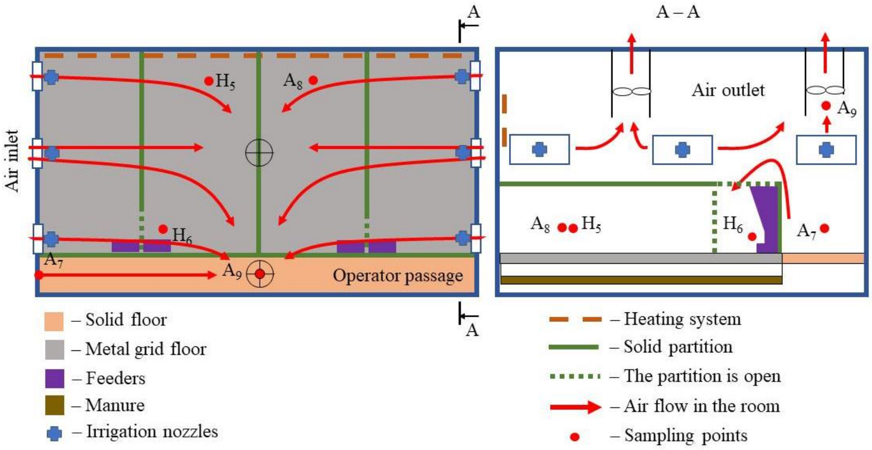


Fig. 2.3. Scheme of a room with a ventilation system

through wall channels (option 3)

Photo of the production conditions of the research, three variants of the ventilation systems of the pig premises are presented in fig. 2.4–2.6.

Control of the intensity of ventilation and heating in all three rooms was based on the measured internal temperature and climate controller settings in winter and summer. Climate settings were copied from those recommended in practice for various ventilation systems and are listed in Table 2.1 for the winter period and Table 2.2 for the summer period.

Table 2.1

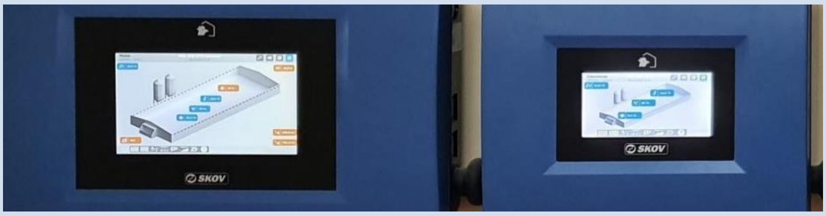
Adjusting the indoor climate in the winter period (December–February)

Ventilation system	Day	Heating	Ventilation set point (°C)	Ventilation per animal (m^3/h)	
				Min.	Max.
Option 1 – the ventilation system of the ground channel	1–14	Enabled [1]	20	3	12
	15–28	Enabled [1]	21	4	15
	29–42	Disabled	22	6	18
	43–56	Disabled	10	9	25
Option 2 – ceiling ventilation system	1–14	Enabled	18	4	13
	15–28	Enabled [1]	19	5	16
	29–42	Disabled	20	7	25
	43–56	Disabled	19	11	35
Option 3 – ventilation system through wall channels	1–14	Enabled [1]	20	3	12
	15–28	Enabled [1]	23	4	15
	29–42	Disabled	18	6	18
	43–56	Disabled	19	9	25

[1] The temperature of the water in the heated system – 40-45 °C.

General view of the premises for keeping pigs

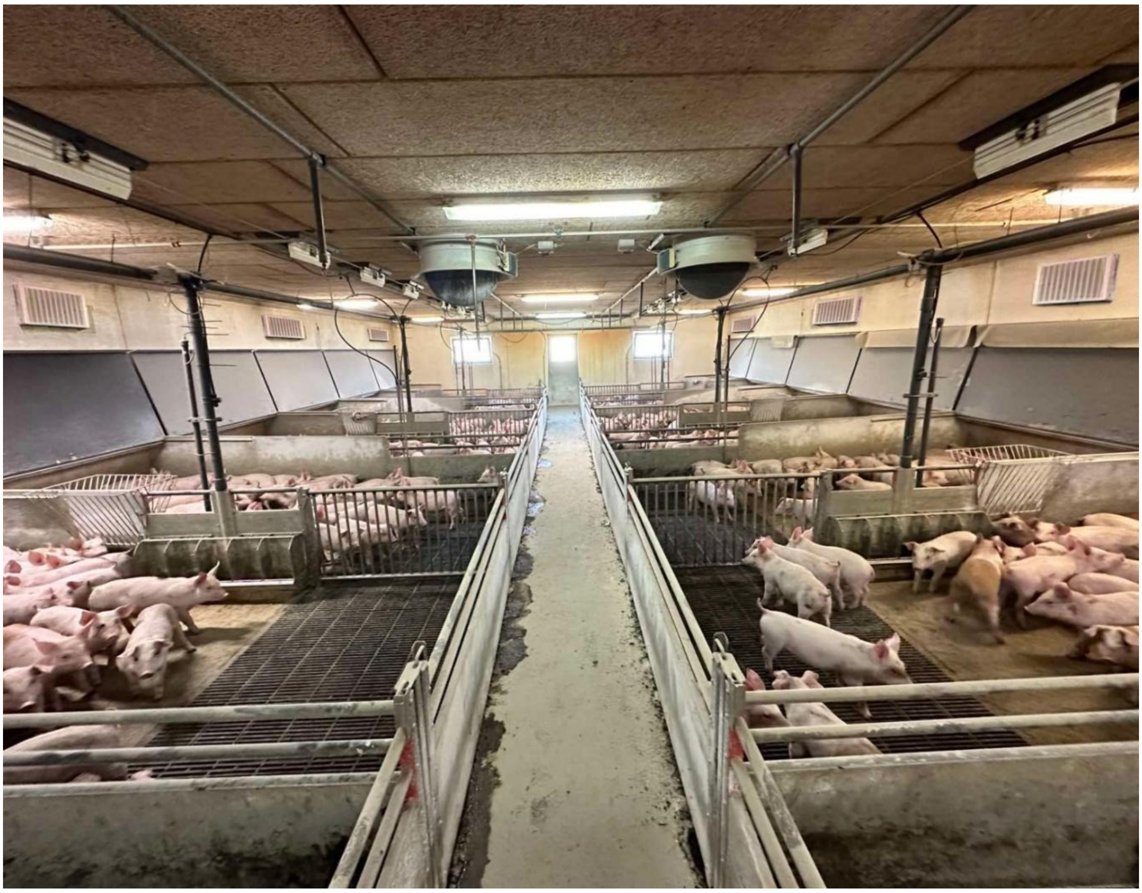
Heat exchanger



Control system of the heating system

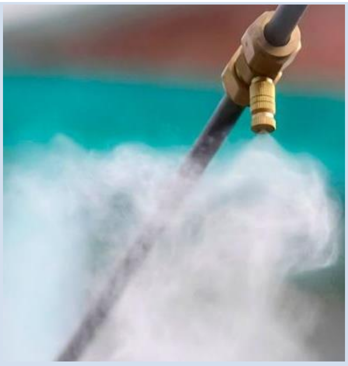
Fig. 2.4. The ventilation system of the ground channel pig premises in production conditions (option 1)

Piglets are weaned from the age of 20–21 days and transferred to group machines for rearing up to 77–78 days of age, namely before transfer to the group of fattening or repair young animals (after reaching a weight of 38 kg). Therefore, the total length of stay of piglets in rearing is 56–57 days on average.



General view of the premises for keeping pigs

Heat exchanger



Irrigation nozzles

Control system of the heating system

Fig. 2.5. Ceiling system of piggery premises
in production conditions (option 2)

Table 2.2

Adjusting the indoor climate in the summer (June–August)

Ventilation system	Day	Cooling	Ventilation set point (°C)	Ventilation per animal (m^3/h)	
				Min.	Max.
Option 1 – the ventilation system of the ground channel	1–14	Enabled [1]	24	3	12
	15–28	Enabled [1]	25	4	15
	29–42	Disabled	24	6	18
	43–56	Disabled	22	9	25
Option 2 – ceiling ventilation system	1–14	Disabled	25	4	13
	15–28	Enabled [1]	26	5	16
	29–42	Enabled [1]	27	7	25
	43–56	Enabled [1]	26	11	35
Option 3 – ventilation system through wall channels	1–14	Disabled	27	3	12
	15–28	Enabled [1]	24	4	15
	29–42	Enabled [1]	24	6	18
	43–56	Enabled [1]	25	9	25

[1] Water consumption from each irrigation nozzle is 14–18 kg/h

On the first day after weaning, the piglets were placed in a room with a temperature of 23–25 °C. For the winter period (starting from February), the temperature in the room was heated with the help of a heating system, in the summer period - with the help of an air irrigation system.

Five sampling points were located in each room, three of which are in the area the animals stay (Figs. 2.1–2.3).

Considerations for determining the location of these sampling points in the animal housing area were as follows: the presence of a sampling point may not affect the behavior of animals in a group machine; differences in concentration are expected CO_2 or temperatures within the animal range, sampling points should be spaced so that this can be measured; at least one of the sampling points must be located in the area animals are expected to lie down; the location of sampling points should be practical for the farm owner.

General view of the premises for keeping pigs

Heat exchanger

Irrigation nozzles

Management system heating systems

Fig. 2.6. The ventilation system through wall channels of piggery premises in production conditions (option 3)

Three indoor sampling points were connected to the TM-32/H-5T system (Ukrrele, Ukraine) and Arduino Nano (Arduino Software) for concentration measurement CO_2 and temperature using, respectively, the sensor of carbon gas content SEN0159 (SEN0159, China), accuracy - ± 5% and temperature sensor DS18B20 (Dallas Semiconductor, USA), accuracy - ± 0.1 °C (Fig. 2.7). Data were recorded consecutively every hour on a personal computer using the software of the company Ukrrele (Ukraine). These sampling points were located at the air inlet, air outlet, and animal housing area approximately 0.20 ± 0.03 m above the floor and are marked with the letter "A" in Fig. 2.1–2.3.

Fig. 2.7. Sensor connection diagram

Two more sampling points in the area the animals were kept were connected to a manual measuring system, multi-gas detector WALCOM MGD-04 (PRC), accuracy ± 5%, to measure the concentration CO_2, which was registered once a day during the first week of the batch and three times a week later (alternately in the morning and in the afternoon) (Fig. 2.8, a). These sampling points were located approximately 0.20±0.03 m above the floor and marked with the letter "H" in Fig. 2.1–2.3. In addition, once at the highest and lowest ambient temperatures, the premises of the pigsty were photographed with a FLIR T200 thermal imager (Teledyne FLIR, USA) to determine the temperature distribution gradient (Fig. 2.8, b).

Fig. 2.8. Multi-gas detector WALCOM MGD-04 (a),
FLIR T200 thermal imager (b) and Benetech GM8903 anemometer (c)

Air samples were collected from automatic and manual sampling points and transported through a system of Teflon pipes to the sensor CO_2 outside the premises in such a way that animal behavior and air flow characteristics are not affected. The thermocouples and Teflon tubing system were protected from animals by perforated iron tubing. In the ventilation exhaust shaft, the intensity of ventilation was continuously measured using an anemometer Benetech GM8903 (PRC) (Fig. 2.8, c). Lying behavior of animals was recorded once a week by the average number of piglets lying within a circle with a diameter of 1 m around each sampling point.

The breed of piglets is a three-breed hybrid of PIC genetics (F1 × PIC 337).

All data were collected in the period December–March and June–August 2022-2023. Every day, the light in the premises was on from 7:00 a.m. to 4:00 p.m.

The territory of Ukraine is in a temperate climate zone in the region of moderately continental climate [60], which is characterized by hot summers and cold winters. Analysis of changes in air temperature and humidity in the environment of the Vinnytsia region. (Fig. 2.9) makes it possible to determine duration of the summer and winter period of 2022 [82]

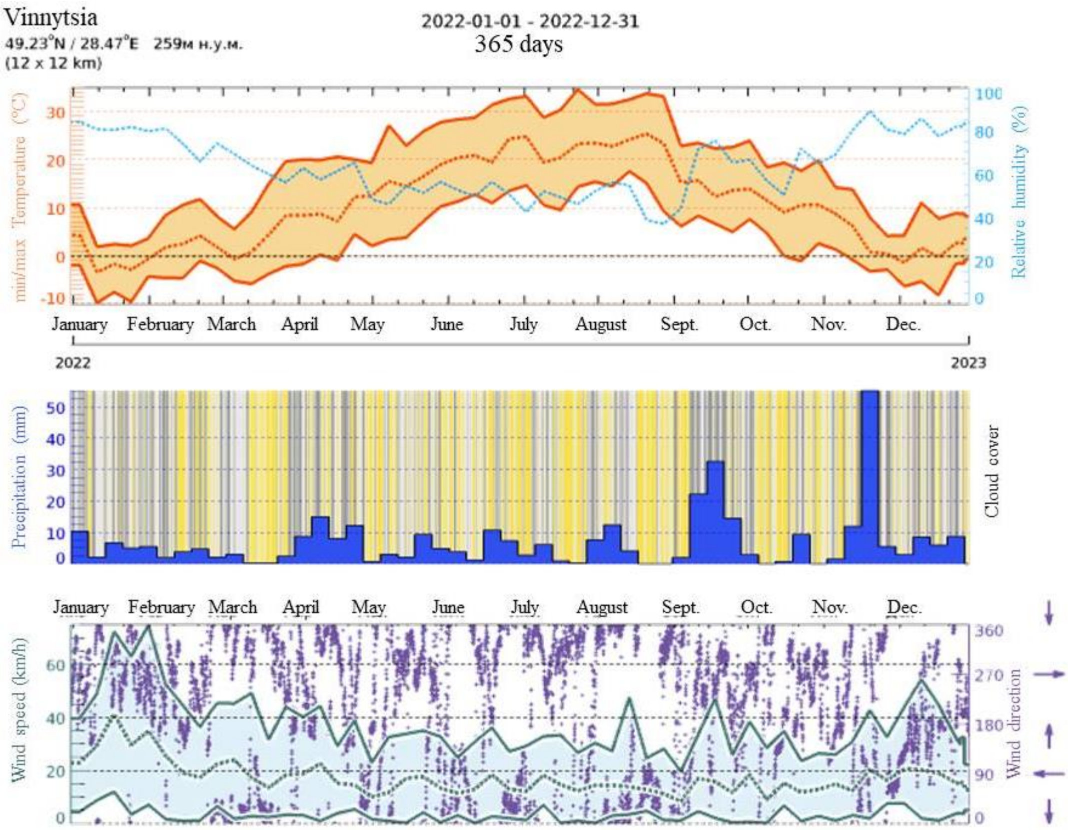
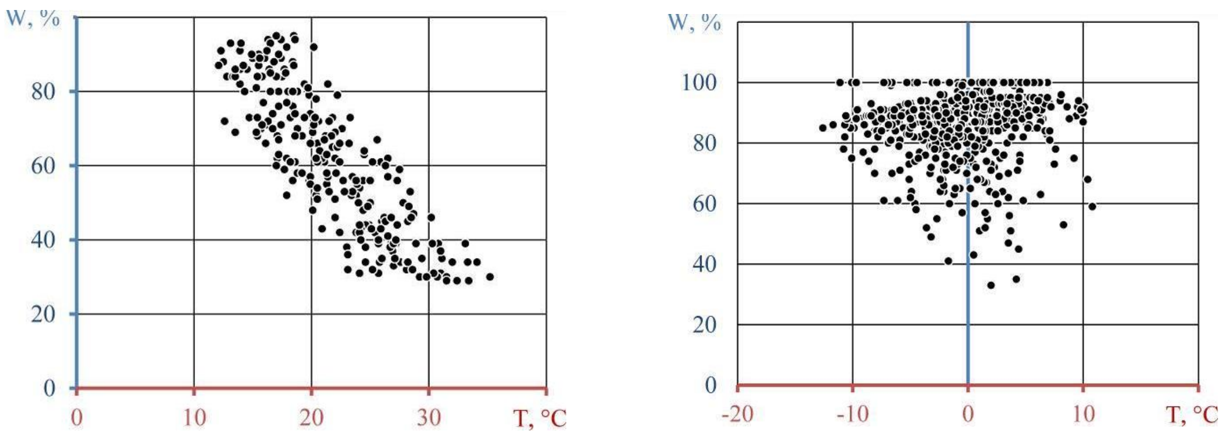


Fig. 2.9. Dynamics of air temperature, humidity
and other environmental parameters of Vinnytsia (2022)

From fig. 2.9 and more extensive data shows that temperature and humidity fluctuate quite significantly during the day. At the same time, there is a certain dependence between temperature and humidity, which can be represented in the form of graphs in fig. 2.10. However, it is clearly manifested in the summer period (from June to August), in the winter period it observes a high value of humidity from 80 to 100%.

Taking into account the need for constant air exchange between the piggery room and the environment, the requirements for microclimate indicators for different sex-age groups obviously make it necessary to increase the temperature of the environment that enters the room in the winter, and on the contrary, to cool it in the summer.



а б

Fig. 2.10. Dependencies between air temperature and environmental humidity in the summer (a) and winter (b) periods of Vinnytsia (2022)

In addition, due to the high value of air humidity (from 80 to 100%) in the environment in the winter period, it is necessary to dry it during its movement into the room. And in summer, at high air temperatures (more than 30 °C), on the contrary, humidify the supply air.

2.2 Theoretical studies of pneumatic losses of a three-pipe concentric heat exchanger

Since the developed structural and technological schemes of the three-pipe concentric heat exchanger (Fig. 1.18) have the same structural elements, the pneumatic losses that occur when the air flow passes through them are the same for both options. Then we will consider the calculation scheme of a three-pipe concentric heat exchanger with counterflow (Fig. 2.11) [7].

At each section of the air flow through the air ducts of the three-pipe concentric heat exchanger, part of the total pressure that goes to overcome the pneumatic resistance forces is lost because, due to the molecular and turbulent viscosity of the moving air, the mechanical work of the resistance forces is transformed into heat [75].

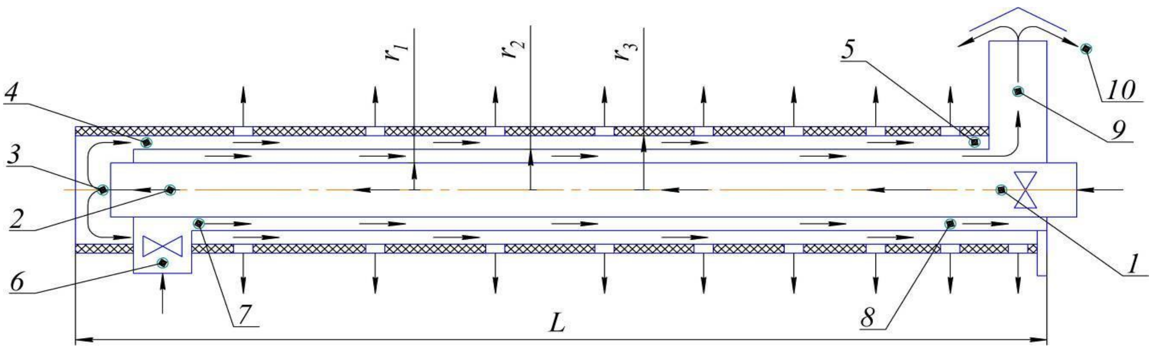


Fig. 2.11. Scheme for pneumatic calculation
of a three-pipe concentric counterflow heat exchanger

There are two types of pneumatic resistance in the duct [76]:

– frictional resistance $\otimes p_f$;

– local resistance $\otimes p_l$.

Pneumatic friction is caused by the viscosity (both molecular and turbulent) of air, which occurs during its movement, and is the result of the exchange of the amount of movement between molecules (in laminar movement), as well as between individual particles (in turbulent movement) of adjacent layers of air, moving at different speeds [68].

Local resistances occur when there is a local violation of the normal flow, separation of the flow from the walls, vortex formation and intense turbulent mixing of the flow in places of changes in the air duct configuration or when meeting and flowing around obstacles. These phenomena enhance the exchange of momentum between moving air particles (i.e., friction), increasing energy dissipation [68].

We will build a mathematical model assuming that the ducts are made of polyethylene, due to its low cost and low coefficient of thermal conductivity.

Consider the pneumatic pressure losses in each section of a three-pipe concentric heat exchanger according to Fig. 2.11 [7].

Frictional pressure losses in the air duct of constant section in sections 1-2, 4-5, 7-8 [77]

$$\Delta p_{f\ 1-2} = \kappa_1 \frac{L}{d_1} \frac{\rho(T_1)\ {v_1}^2}{2}, \tag{2.2}$$

$$\Delta p_{f\ 7-8} = \kappa_2 \frac{L}{d_2} \frac{\rho(T_2)\ {v_2}^2}{2}, \tag{2.3}$$

$$\Delta p_{f\ 4-5} = \kappa_3 \frac{L}{d_3} \frac{\rho(T_3)\ {v_3}^2}{2}, \tag{2.4}$$

$|_i$ – coefficient of friction resistance;

L – duct length, m;

d_i – effective diameter of the i-th duct, m [78]:

$$d_i = \frac{4A_i}{p_i}, \tag{2.5}$$

A_i – cross-sectional area of the ith duct, m^2: for the 1st – $A_1 = \pi \cdot r_1^2$, 2nd – $A_2 = \pi \cdot \left(r_2^2 - r_1^2\right)$, on the 3rd – $A_3 = \pi \cdot \left(r_3^2 - r_2^2\right)$;

p_i – total perimeter of the cross section of the i-th duct, m: for the 1st – $p_1 = 2\pi r_1$, 2nd – $p_2 = 2\pi\left(r_2 + r_1\right)$, on the 3r – $p_3 = 2\pi\left(r_3 + r_2\right)$;

$\rho(T_i)$ – air density in the ith duct at constant pressure, which is related to its temperature, kg/m^3:

$$\rho(T_i) = \rho_{н.у.} \frac{273}{T_i}; \tag{2.6}$$

$\rho_{н.у.}$ – air density under normal conditions ($T_{н.у.}$ = 273 K, $P_{н.у.}$ = 101325 Pas), $\rho_{н.у.}$ =

1,293 kg/m^3 [79].

v_i – air velocity in the ith duct, m/s:

$$v_i = \frac{V_i}{A_i}; \tag{2.7}$$

V_i – volume flow rate of air in the ith duct, m^3/s.

As a result of the calculation according to formula (2.5), we obtain the effective diameters of air ducts: for the 1st – $d_1 = 2r_1$, 2nd – $d_2 = 2(r_2 - r_1)$, on the 3r – $d_3 = 2(r_3 - r_2)$.

According to experiments, the coefficient of friction resistance during turbulent motion depends on the Reynolds number and the roughness of the duct walls. According to A.D. Altshulem, it is equal to [80]

$$\kappa_i = 0{,}11\ \sqrt[4]{\frac{68}{Re_i} + \frac{\psi}{d_i}}, \tag{2.8}$$

ψ – equivalent roughness of the duct walls, for polyethylene we accept ψ = 0,1 mm [66];

Re_i – Reynolds number for the air flow in the i-th duct:

$$Re_i = \frac{d_i \cdot v_i \cdot \rho(T_i)}{\mu}; \tag{2.9}$$

μ –dynamic air viscosity, μ = 18,27·10^{-6} H·s/m^2 [66].

According to [66] pressure loss in the knee (section 12)

$$\Delta p_{16\text{-}7} = 4\alpha \sin^2 \frac{\theta_{6-7}}{2} \frac{\rho(T_2)\, v_2^2}{2}, \tag{2.10}$$

$$\Delta p_{18\text{-}9} = 4\alpha \sin^2 \frac{\theta_{8-9}}{2} \frac{\rho(T_2)\, v_2^2}{2}, \tag{2.11}$$

〈 – impact mitigation factor for a knee of constant cross-section 〈 = 0,55;

⎝ – knee angle, according to the Fig. 1: ⎝$_{6\text{-}7}$ =⎝$_{8\text{-}9}$ = 90°.

Pressure losses in a spatial (annular) turn by 180° during injection (sections 2-3-4 and 9-10) [69]

$$\Delta p_{12\text{-}3} = \zeta_{2-3} \frac{\rho(T_1)\, v_1^2}{2}, \tag{2.12}$$

$$\Delta p_{\mathrm{l}\,9-10} = \zeta_{9-10}\,\frac{\rho(T_2)\ v_2^2}{2}, \tag{2.13}$$

ζ – coefficient of local resistance for spatial (circular) rotation by 180° during injection, according to [73] $\zeta_{2\text{-}3} = \zeta_{9\text{-}10} = 2$.

The average pressure losses that occur when air passes through the holes in section 4-5 according to studies [73] can be calculated by the formula:

$$\Delta p'_{\mathrm{l}\,4-5} = \xi_{4-5}\,\frac{\rho(T_3)\ v_3^2}{2}, \tag{2.14}$$

$\xi_{4\text{-}5}$ – according to [75], we accept the hole flow coefficient $\xi_{4\text{-}5} = 4$.

Pressure losses in a three-pipe concentric heat exchanger are defined as the sum of all pressure losses

$$\begin{gathered}\Delta p = \Delta p_{\mathrm{f}\,1-2} + \Delta p_{\mathrm{l}\,2-3-4} + \Delta p_{\mathrm{f}\,4-5} + \Delta p'_{\mathrm{l}\,4-5} + \\ +\Delta p_{\mathrm{l}\,6-7} + \Delta p_{\mathrm{f}\,17-8} + \Delta p_{\mathrm{l}\,8-9} + \Delta p_{\mathrm{l}\,9-10}.\end{gathered} \tag{2.15}$$

Assuming the equality of volume flow of air in all ducts $V_i = V$ the power required to pump air through a three-pipe concentric heat exchanger is determined by the formula

$$N_f = \frac{V\ \Delta p}{\eta_n}, \tag{2.16}$$

$\eta_{п}$ – full fan efficiency, $\eta_{п} = 0{,}8$ [81].

According to the analysis of research on ventilation systems [81], we accept the condition of equality of the effective diameters of air ducts to ensure uniformity of pressure flows. However, taking into account the presence of distribution holes on the external pipeline, its effective diameter should be 8-12% larger [74]. Considering the above, we have

$$2r_1 = 2(r_2 - r_1) = 2 \cdot 1{,}1 \cdot (r_3 - r_2), \tag{2.17}$$

$$\begin{cases} r_1 \approx 0{,}343 \cdot r_3, \\ r_2 \approx 0{,}686 \cdot r_3. \end{cases} \tag{2.18}$$

According to the obtained dependencies (2.2)-(2.17) and by varying the structural and technological parameters, namely the radius of the external air duct r3, its length L and the supply of air flow V in a wide range, we obtain the dependence of the change in power, which is necessary for pumping air through a three-pipe concentric heat exchanger from of the above-mentioned factors (Fig. 2.12-2.13).

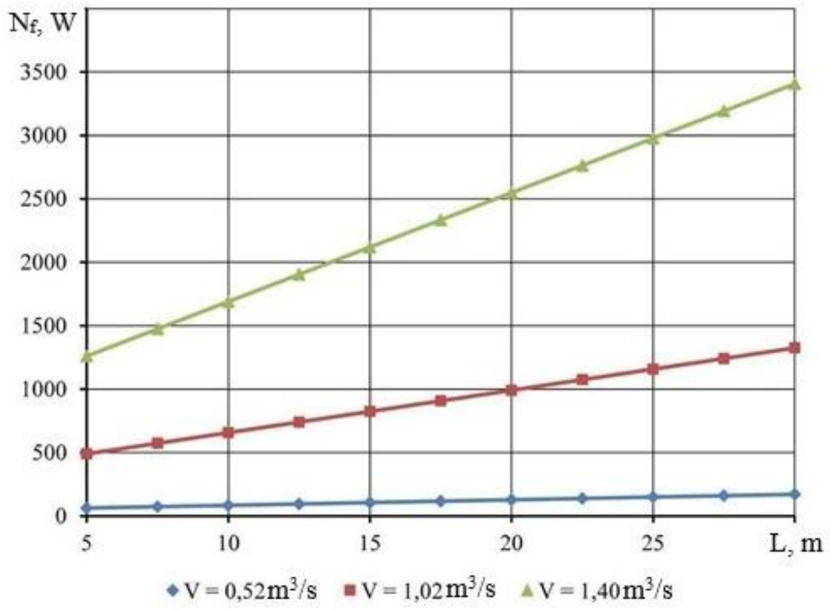


Fig. 2.12. Dependence of power change Nf pneumatic loss from the length of the heat exchanger L and air flow supply V at a fixed value of the radius of the external duct r_3 = 0,47 m

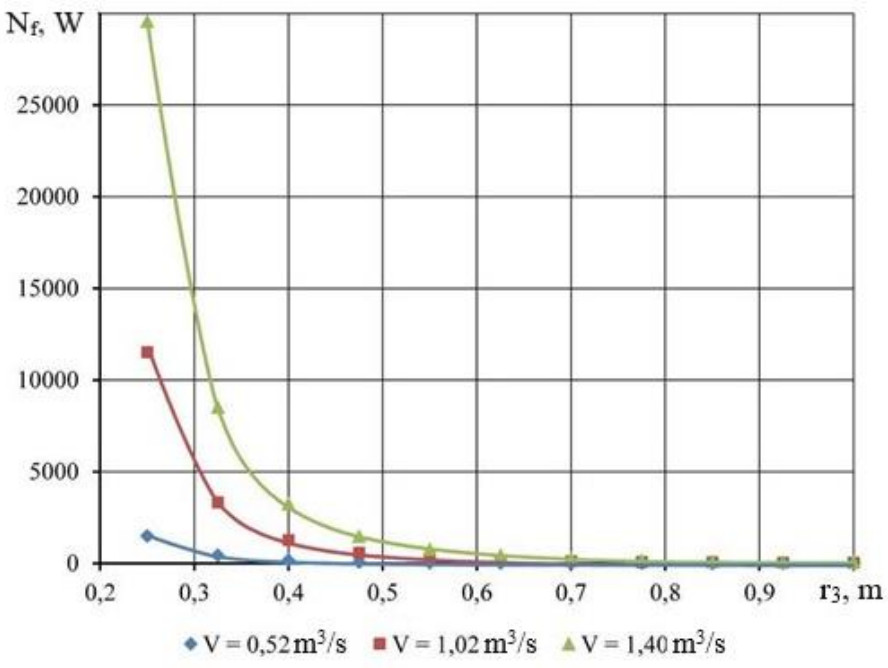


Fig. 2.13. Dependence of power change Nf pneumatic loss from the radius of the external duct r_3 and air flow supply V at a fixed value of the length of the heat exchanger L = 7,5 m r_3, m N_f, W m^3/s

Analyzing figures 2.12-2.13, it was found that the power N_f air loss is directly proportional to the length of the heat exchanger L and inversely proportional to the radius of the external duct r_3 [7].

2.3 Mathematical model of the heat transfer process in a three-pipe concentric heat exchanger

To develop a mathematical model of the heat transfer process in a three-pipe concentric heat exchanger, we accept the following assumptions:

1) the process of heat transfer through the walls of air ducts takes place only through their thickness;

2) the process of heat transfer through the walls of air ducts occurs instantly;

3) due to thermal insulation of the external duct, the process of heat transfer through its wall is absent;

4) due to a slight change in the pressure of the air flow (Δp = 10-200 Pa [82]) when it moves through the air duct, we consider the thermodynamic process of the system as isobaric;

5) in the first, we consider the approximate heat transfer process without taking into account the phenomenon of condensation of water vapor with subsequent correction of the obtained values;

6) the air flow in the ducts is uniform and isotropic.

To study the process of heat transfer in a three-pipe concentric heat exchanger, we will make a calculation scheme: counterflow (Fig. 2.14, a), forward flow (Fig. 2.14, b).

As the abscissa axis, the axis of the three-pipe heat exchanger with the origin of coordinates in the center of its end section is chosen.

In the first, we will consider the approximate process of heat transfer in a three-pipe heat exchanger without taking into account condensation. On the section of the heat exchanger with a length of dx in each pipe, a heat flow is observed, which occurs due to the formation of a temperature gradient. According to the Newton-Richmann law [83] we have:

$$dQ_i = Q_i(x + dx) - Q_i(x) = m_i C p_d T_i(x), \qquad (2.19)$$

Q – heat flow, W;

x – coordinate, m;

i – duct number;

m_i – air mass flow rate in the i-th duct, kg/s;

C_p – specific heat capacity of air, C_p = 1006 J/(kg·K) [84];

T_i – air flow temperature in the i-th duct, K

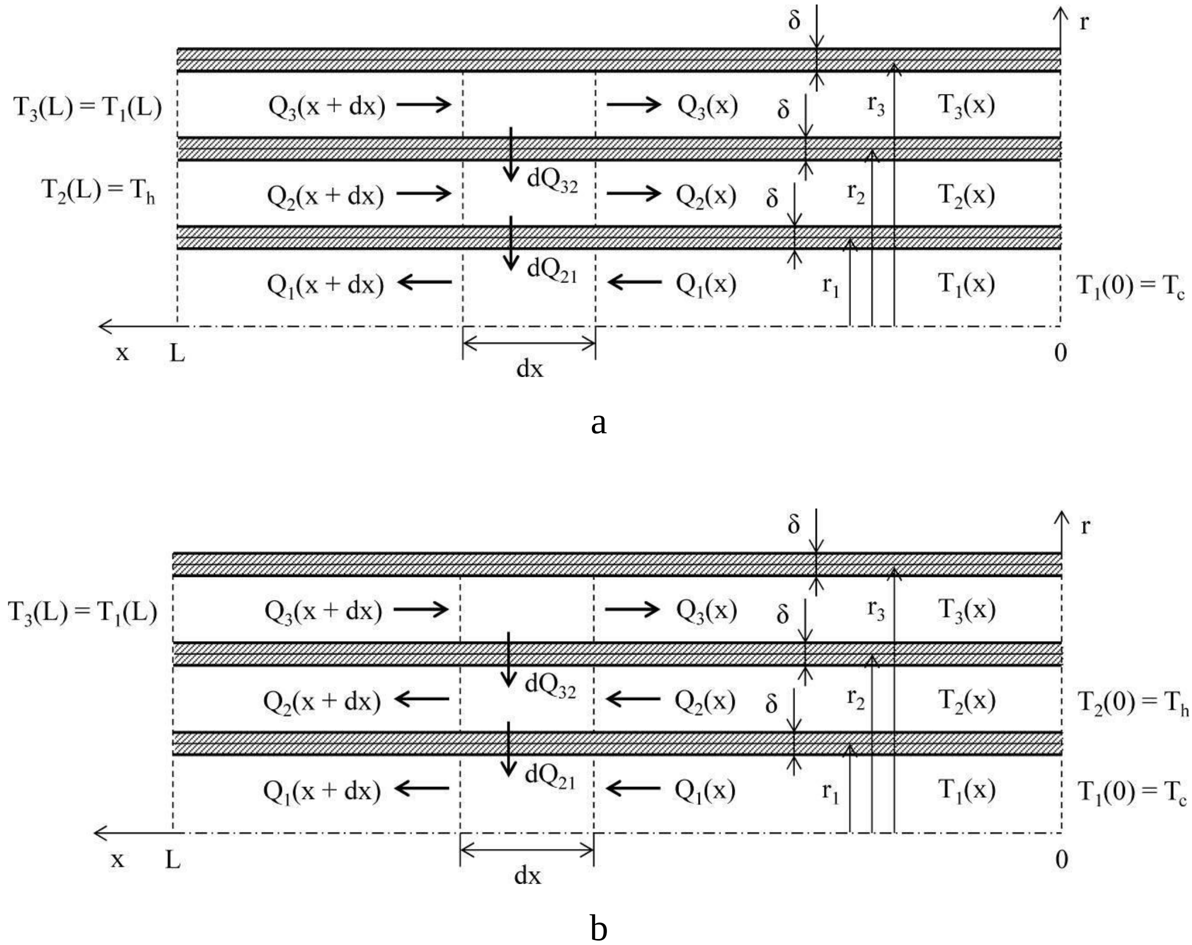


Fig. 2.14. Calculation diagram of the heat transfer process in a three-pipe heat exchanger: counterflow (a) and forward flow (b)

Due to the temperature difference of each air duct, the heat transfer process occurs through its walls. According to the equation of heat transfer through a cylindrical wall [85] we obtain:

$$dQ_{ij}(x) = \pi K_j \, (T_i(x) - T_i(x))dx, \tag{2.20}$$

j – duct number;

Q_{ij} – heat flow through the duct wall, J/s;

K_j – linear heat transfer coefficient, W/(m·K).

According to the law of conservation of energy for the area of the heat exchanger dx (Fig. 2.14), we obtain the following identities (the upper sign for counterflow, the lower for forward flow):

$$\begin{cases} \dot{Q}_1(x+dx) - \dot{Q}_1(x) - d\dot{Q}_{21}(x) = 0, \\ \mp\left(\dot{Q}_2(x+dx) - \dot{Q}_2(x)\right) + d\dot{Q}_{21}(x) - d\dot{Q}_{32}(x) = 0, \\ \dot{Q}_3(x+dx) - \dot{Q}_3(x) - d\dot{Q}_{32}(x) = 0. \end{cases} \tag{2.21}$$

Substituting (2.18)-(2.19) into (2.20), we obtain a system of differential equations for the heat transfer process in a three-pipe heat exchanger without taking condensation into account:

$$\begin{cases} \dot{m}_1 C_p \mathrm{dT}_1(x) - \pi\mathrm{K}_1\left(T_2(x) - T_1(x)\right)\mathrm{dx} = 0, \\ \mp\dot{m}_2 C_p \mathrm{dT}_2(x) + \pi\mathrm{K}_1\left(T_2(x) - T_1(x)\right)\mathrm{dx} - \pi\mathrm{K}_2\left(T_3(x) - T_2(x)\right)\mathrm{dx} = 0, \\ \dot{m}_3 C_p \mathrm{dT}_3(x) - \pi\mathrm{K}_2\left(T_3(x) - T_2(x)\right)\mathrm{dx} = 0. \end{cases} \tag{2.22}$$

or

$$\begin{cases} \dot{m}_1 C_p \frac{\mathrm{dT}_1(x)}{\mathrm{dx}} - \pi\mathrm{K}_1\left(T_2(x) - T_1(x)\right) = 0, \\ \mp\dot{m}_2 C_p \frac{\mathrm{dT}_2(x)}{\mathrm{dx}} + \pi\mathrm{K}_1\left(T_2(x) - T_1(x)\right) - \pi\mathrm{K}_2\left(T_3(x) - T_2(x)\right) = 0, \\ \dot{m}_3 C_p \frac{\mathrm{dT}_3(x)}{\mathrm{dx}} - \pi\mathrm{K}_2\left(T_3(x) - T_2(x)\right) = 0. \end{cases} \tag{2.23}$$

The linear heat transfer coefficient for a cylindrical duct can be calculated by the formula [86, 87]:

$$K_i = \frac{1}{\frac{1}{\alpha_i(2\mathrm{r}_i - \delta)} + \frac{1}{\alpha_i(2\mathrm{r}_i + \delta)} + \frac{1}{2\lambda}\lg\left(\frac{2\mathrm{r}_i + \delta}{2\mathrm{r}_i - \delta}\right)}, \tag{2.24}$$

α_i –coefficient of heat transfer in the ith duct, W/(m^2·K);

δ – duct wall thickness, m;

r_i –radius of the i-th duct, m;

λ – specific thermal conductivity of the duct wall, for polyethylene λ = 0,3 W/(m·K) [88].

The coefficient of heat transfer in the i-th duct depends on the Nusselt number and is calculated according to the formula [73]:

$$\alpha_i = \frac{\lambda \cdot Nu_i}{d_i}, \tag{2.25}$$

Nu_i – Nusselt number for air flow in the i-th duct;

d_i – effective diameter of the i-th duct, m.

The effective diameter of the duct is determined by the formula (2.4).

According to studies [74], the Nusselt number for the air flow depends on the Reynolds number and is determined by the dependence:

$$Nu_i = 0{,}018 \cdot Re_i^{0,8} \tag{2.26}$$

Re_i – the Reynolds number for the air flow in the i-th duct is determined by the formula (2.9).

The mass flow of air in the i-th duct can be calculated by the formula:

$$m_i = V_i \cdot \rangle(T_i). \tag{2.27}$$

According to Fig. 2.14 (a), the boundary conditions for the system of differential equations (2.23) are

$$\begin{cases} T_1(0) = T_c, \\ T_2(L) = T_h, \\ T_1(L) = T_3(L), \end{cases} \tag{2.28}$$

L – duct length, m;

T_c – temperature at the entrance of the 1st duct, K;

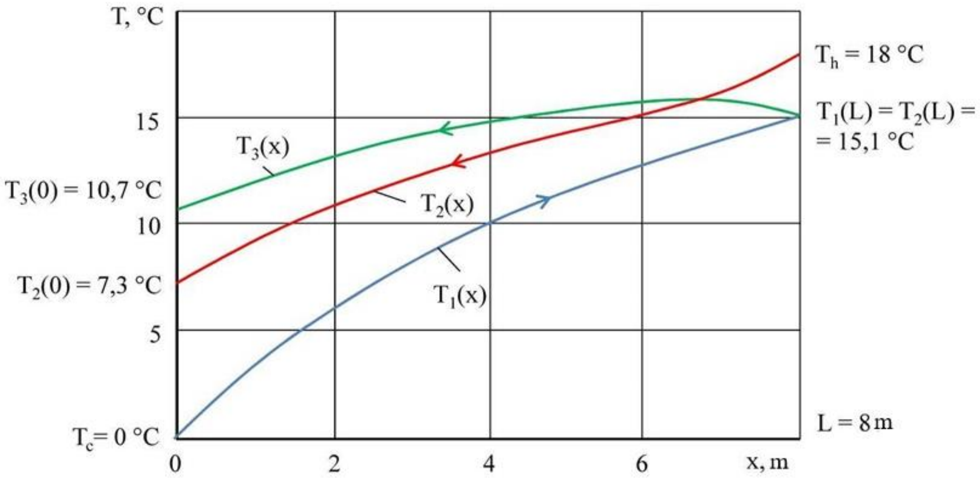
T_h – temperature at the entrance of the 2nd duct, K.

According to Fig. 2.14 (b), the boundary conditions for the system of differential equations (2.22) are

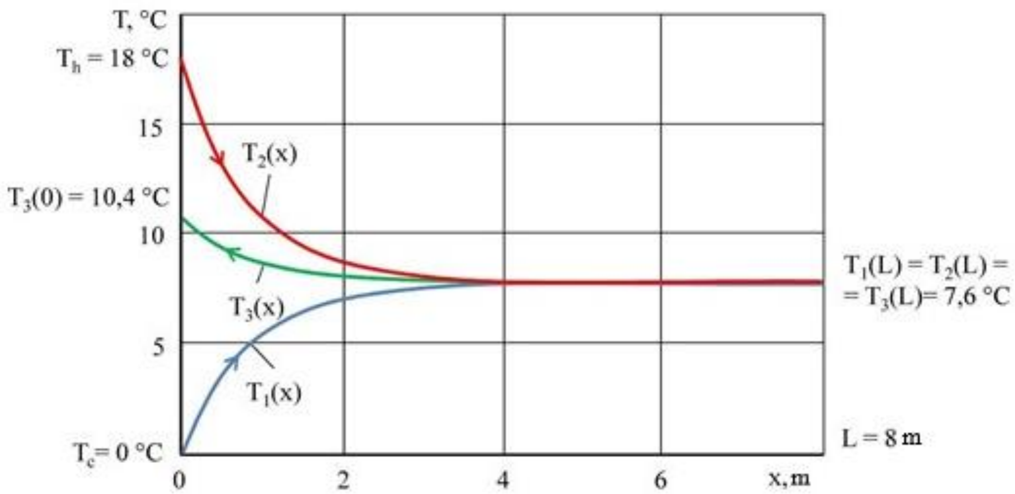
$$\begin{cases} T_1(0) = T_c, \\ T_2(0) = T_h, \quad . \\ T_1(L) = T_3(L) \end{cases} \tag{2.29}$$

We will solve the system of differential equations (2.23) together with the dependencies (2.19)-(2.29) in the Mathematica software package, which uses the numerical method of least squares [89].

Accepting design and technological parameters (L = 8 m; r_1 = 0,25 m; r_2 = 0,172 m; r_3 = 0,086 m; $V_1 = V_2 = V_3$ = 0,14 m^3/s; T_c = 273 K; T_h = 291 K; δ = 0,0002 m) of a three-pipe concentric heat exchanger, we obtain the temperature distribution of air flows along its length with counterflow (Fig. 2.15, a) and forward flow (Fig. 2.15, b).



a



b

Fig. 2.15. Temperature distribution of air flows in a three-pipe heat exchanger1 with counterflow (a) and forward flow (b) along its length

As can be seen from Fig. 2.15 (a), the cold flow of air from the environment is heated in the 1st duct from the temperature T_c to $T_1(L) = T_2(L)$. This happens due to the transfer of heat from the 2nd duct, the air from the room is cooled from the temperature T_h to $T_2(0)$. Next, the air flow from the environment passes through the 3rd duct, it is first heated and then cooled to a temperature of $T_3(0)$. This is caused by the contact of the 2nd air duct with the cooled air flow in the 2nd air duct.

Another situation is shown in Fig. 2.15 (b) the cold flow of air from the environment is heated in the 1st duct from the temperature T_c to $T_1(L) = T_2(L) = T_3(L)$.

To take into account the phenomenon of condensation during the operation of a three-pipe concentric heat exchanger, we calculate the temperature at which this phenomenon begins to occur (dew point). According to studies [90], the dew point is calculated by the formula:

$$T_r(T,W) = \frac{b\left(\frac{a(T-273)}{b+T-273}+lg\,W\right)}{a-\frac{a(T-273)}{b+T-273}+lg\,W} + 273, \tag{2.30}$$

a, b – empirical coefficients, a = 17,27, b = 237,7 °C [90];

W – relative air humidity, we accept W = 0,6 [91].

Analysis of the obtained solution of the system of differential equations (2.22) shows that condensate is formed in the 2nd duct. Solving Eq

$$T_2(x_T) = T_r(T_h, W), \tag{2.31}$$

we get the value x_T during which the phenomenon of condensation in the 2nd air duct stops.

According to studies [92], the heat flow that occurs due to the formation of a temperature gradient along the length of the 2nd air duct, taking into account condensate, can be calculated as follows:

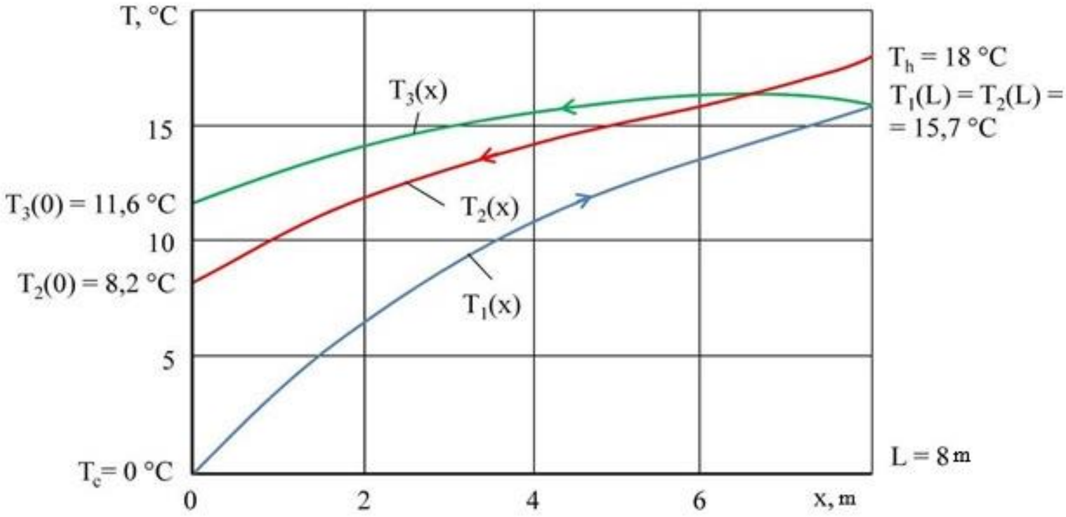
$$d\dot{Q}_2(x) = \dot{Q}_2(x+dx) - \dot{Q}_2(x) = \left(1+\frac{x_T}{L}\right)\dot{m}_2 C_p dT_2(x), \tag{2.32}$$

By substituting (2.18)-(2.19) and (2.31) into (2.20), we obtain a system of differential equations for the heat transfer process in a three-pipe heat exchanger, taking into account condensation:

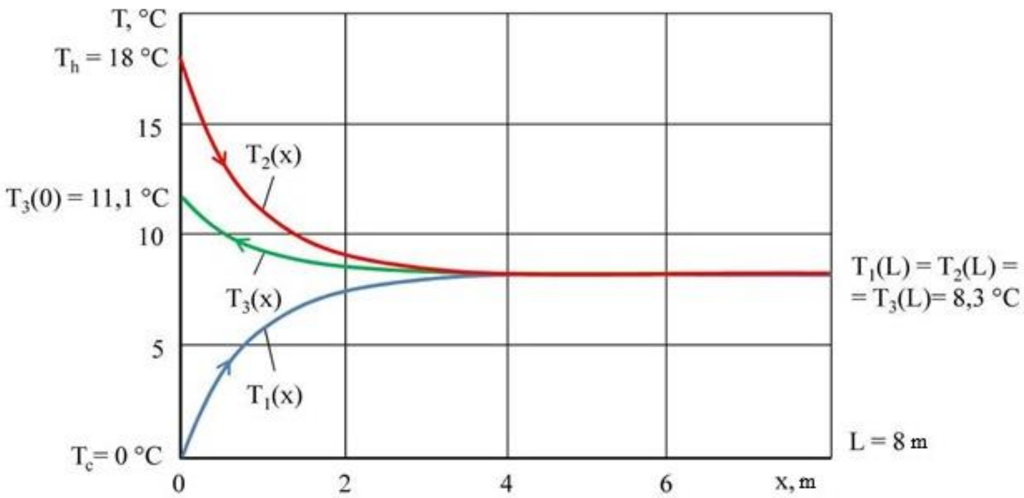
$$\begin{cases} \dot{m}_1 C_p \dfrac{dT_1(x)}{dx} - \pi K_1\big(T_2(x) - T_1(x)\big) = 0, \\ \mp\left(1+\dfrac{x_T}{L}\right)\dot{m}_2 C_p \dfrac{dT_2(x)}{dx} + \pi K_1\big(T_2(x) - T_1(x)\big) - \pi K_2\big(T_3(x) - T_2(x)\big) = 0, \\ \dot{m}_3 C_p \dfrac{dT_3(x)}{dx} - \pi K_2\big(T_3(x) - T_2(x)\big) = 0. \end{cases} \tag{2.33}$$

We will solve the system of differential equations (2.33) together with the dependencies (2.20)-(2.28) in the Mathematica software package, which uses the numerical method of least squares.

Accepting the structural and technological parameters of the three-pipe concentric heat exchanger, we obtain the distribution of the temperature of the air flows along its length, taking into account condensation (Fig. 2.16).



a



b

Fig. 2.16. Temperature distribution of air flows in a three-pipe heat exchanger with counterflow (a) and forward flow (b) along its length, taking into account condensation

Comparing figures 2.15 and 2.16, it is clear that the mathematical model, taking into account the process of condensation in the 2nd duct, gives an increase in the temperature of the air flow by 3-10%.

The analysis of the results of theoretical studies of the heat transfer process in the structural and technological schemes of the three-pipe concentric heat exchanger with counterflow (Fig. 2.16, a) and direct flow (Fig. 2.16, b) showed that the option with counterflow is more effective, because it provides a temperature gradient within from 11.6 to 15.7 °C compared to direct current - from 8.3 to 11.1 °C. The average value of the temperature in counterflow is 13.7 °C, and in direct flow it is 9.7 °C.

Taking into account the above, we accept the structural and technological scheme of a three-pipe concentric heat exchanger with counterflow of air flows.

For the energy evaluation of the heat transfer process in a three-pipe concentric counterflow heat exchanger, we calculate the thermal power used to heat the air flow in the 2nd ducts, according to the Newton-Richmann law [93]:

$$N_Q = \dot{m}_2 C_p (T_3(0) - T_c). \tag{2.34}$$

For the accepted design and technological parameters1 of a three-pipe concentric heat recovery unit with a counterflow, the thermal power is N_Q = 1,96 kW.

2.4 Optimization of design and mode parameters of a three-pipe concentric counterflow heat exchanger

To optimize the structural parameters of the three-pipe concentric counterflow heat exchanger, we will use the developed mathematical model of the heat transfer process (Section 2.2).

For further calculations of the radii of the heat recovery pipelines, we will use formula (2.18).

Variations in the design and operating parameters of the three-pipe concentric heat exchanger were carried out within the limits of:

– the length of the external pipe of the heat exchanger L = 5-30 m;

– the radius of the external pipe of the heat exchanger r_3 = 0,25-1,00 m;

–volume flow of air V = 0,14-1,4 m^3/s;

– ambient temperature T_c = 0-10 °C.

The useful heat capacity of the heat recovery unit, which is determined by the formula, was chosen as the optimization criterion:

$$\Delta N = N_Q - N_f. \tag{2.35}$$

The step-by-step selection of each of the above-mentioned parameters made it possible to obtain graphical dependences of their influence on the useful heat capacity of the heat-utilizer (Fig. 2.17-2.19).

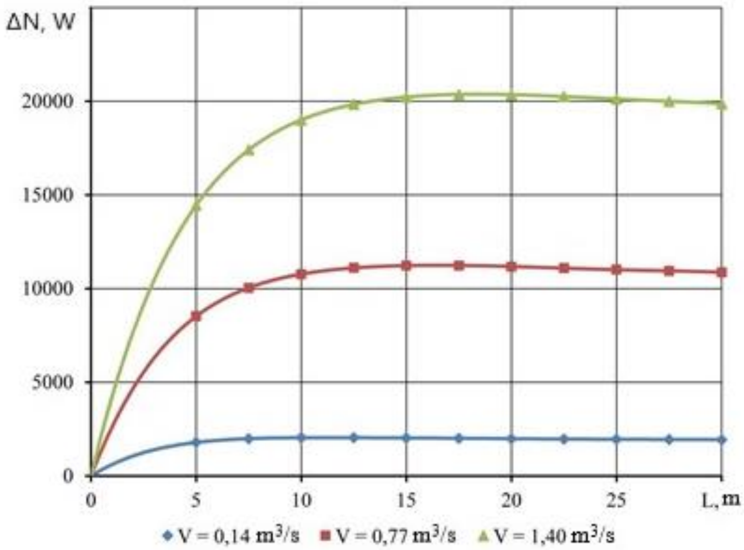


Fig. 2.17. Dependence of the useful thermal power of the heat utilization device ΔN from the length of its outer pipe L and volume flow of air V at a fixed value r_3 = 1 m, T_c = 0 °C

Fig. 2.17 shows that the useful thermal power increases to the maximum value with an increase in the length of the heat exchanger, and then slightly decreases, that is, the optimum is observed. This optimum occurs due to the increase in pneumatic losses during air flow through a long pipeline (Fig. 2.12).

According to Fig. 2.18, with a small value of the radius of the external air duct of the heat exchanger, a negative value of the useful thermal power is observed, which is caused by large pneumatic losses. However, when the radius of the external air duct increases, the optimum of useful thermal power appears.

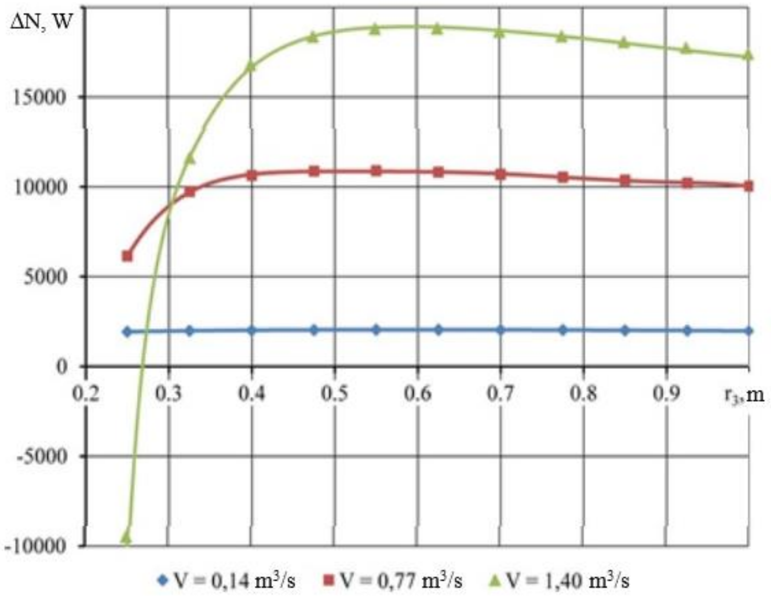


Fig. 2.18. Dependence of the useful heat capacity
of the heat utilizer ΔN from the radius of its outer pipe r_3
and volume flow of air B at a fixed value L = 7,5 m, Tc = 0 °C

The dependence of the useful thermal capacity of the heat exchanger on the ambient temperature (Fig. 2.19) is linear. As the value of the ambient temperature increases, the useful heat capacity of the heat exchanger decreases.

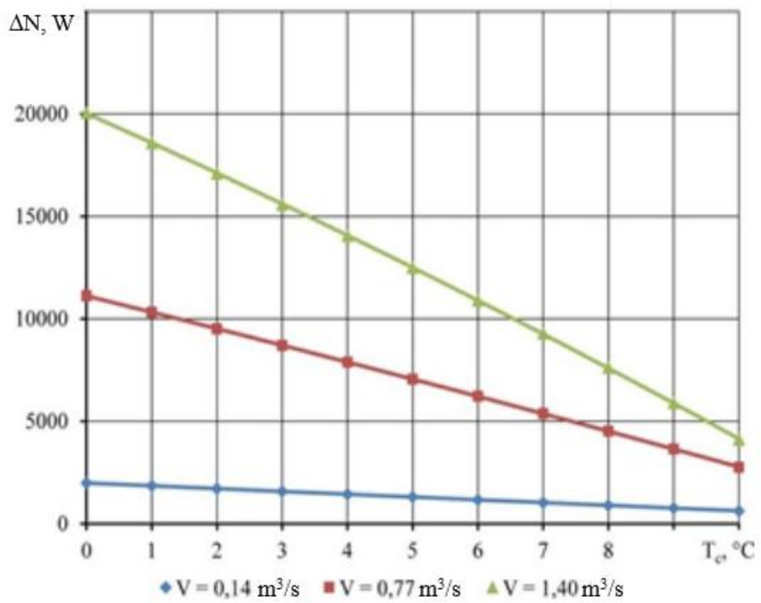


Fig. 2.19. Dependence of the useful heat capacity
of the heat utilizer ΔN from the ambient temperature T_c
and volumetric air flow V at a fixed value of L = 7.5 m, r_3 = 0,7

A system of equations was created to optimize the design parameters:

$$\begin{cases} \Delta N \to \max, \\ L \to \min, \\ r_3 \to \min. \end{cases} \tag{2.36}$$

Assuming ambient temperature conditions T_c = 0 °C at different volumetric air flows V, the solution of the system of equations (2.33) is the data summarized in the table 2.3.

By approximating the data in Table 2.3, we obtain the dependence of the design parameters of the heat exchanger on the volumetric flow rate of the air passing through it under the condition of the greatest useful thermal power:

$$r_3 = 0{,}3619 \cdot V + 0{,}1523, \tag{2.37}$$

$$L = 14{,}776 \cdot V + 3{,}7335, \tag{2.38}$$

$$\Delta N = 13713 \cdot V - 144{,}92. \tag{2.39}$$

Table 2.3

Optimum structural and operational parameters of a three-pipe concentric counterflow heat exchanger made of polyethylene

V, m^3/s	L, m	r_3, m	ΔN, W
0,14	5,8	0,2	1906
0,27	7,74	0,24	3551
0,39	9,46	0,3	5222
0,52	11,43	0,35	6976
0,64	13,19	0,39	8583
0,77	15,13	0,43	10315
0,9	16,99	0,47	12071
1,02	18,85	0,53	13866
1,15	20,71	0,55	15503
1,27	22,53	0,62	17380
1,4	24,39	0,66	19179

Graphical interpretation of dependencies (2.37)-(2.39) is presented in the figures 2.20-2.22.

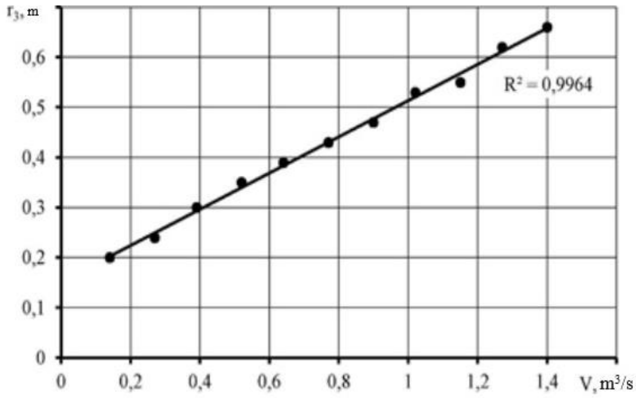


Fig. 2.20. Dependence of the radius of the heat exchanger r_3 on the volume flow rate of air V passing through it under the condition of the greatest useful thermal power

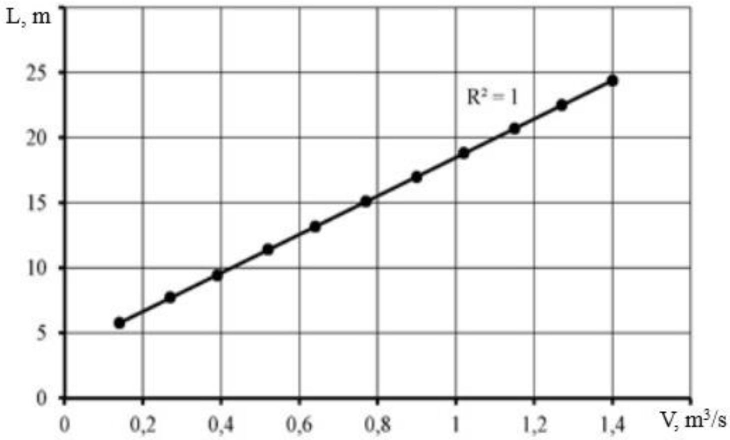


Fig. 2.21. Dependence of the length of the heat exchanger L on the volumetric flow rate of air V passing through it under the condition of the greatest useful heat capacity

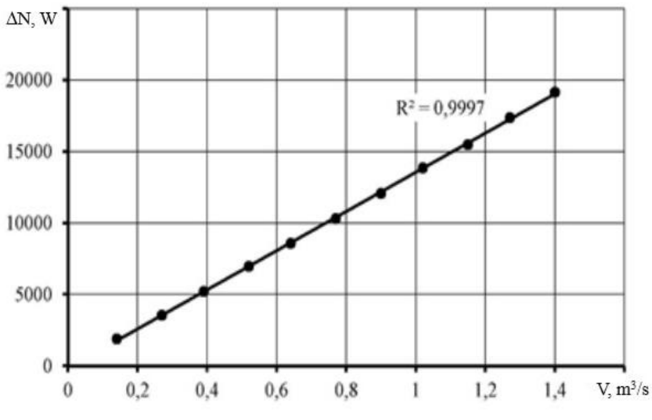


Fig. 2.22. Dependence of the useful thermal power ΔN
on the volume flow of air V passing through the heat exchanger

2.5 Justification of the geometry of the location of the holes in the air duct of a three-pipe concentric heat exchanger

Let's consider the calculation scheme for determining the geometry of the location of the holes in the air duct of a three-pipe concentric heat exchanger (Fig. 2.23). As the abscissa axis, the axis of the three-pipe heat exchanger with the origin in the center of its end section is chosen. The heat exchanger has a length L, along which is located *n* holes of the same plane σ. The speed of the air flow at the beginning of the duct is v_n. It is necessary to establish how the distance between the holes changes along the length of the heat exchanger to ensure uniform distribution of air through the holes [6].

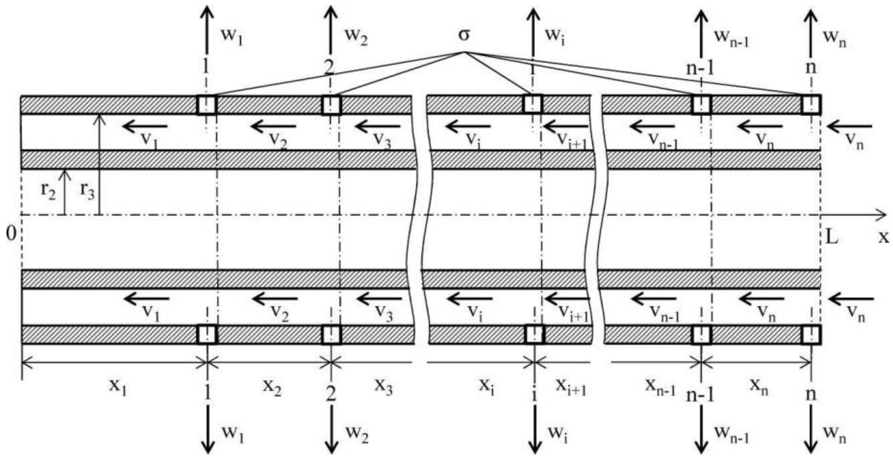


Fig. 2.23. Calculation scheme for determining the geometry of the location of the holes in the duct of a three-pipe heat exchanger

We number all the holes against the movement of the air flow and draw cross-sections behind each hole [6].

The speed of the air flow passing through the i-th hole is determined according to the Torricelli formula [96]:

$$w_i = \phi\sqrt{\frac{2}{\rho}\Delta \mathrm{p}_i}, \tag{2.40}$$

ρ – air density, kg/m^3;

Δp_i – pressure loss at the i-th opening, Pa;

φ –orifice flow rate, φ = 0,65 [94].

From equation (2.40) we express Δp_i:

$$\Delta \mathrm{p}_i = \frac{\rho}{2}\left(\frac{w_i}{\phi}\right)^2, \tag{2.41}$$

The condition for uniform distribution of air is:

$$\frac{\sigma \cdot w_i}{x_i} = \frac{A \cdot v_n}{L}, \tag{2.42}$$

σ – hole area, m^2;

L – length of the heat exchanger, m;

x_i – distance between the i-th and (i-1)-th holes, m;

v_n – air flow rate at the beginning of the duct, m/c:

$$v_n = \frac{V_0}{A}; \tag{2.43}$$

V_0 – volume flow of air at the beginning of the duct, m^3/s;

A –cross-sectional area of the duct, m^2:

$$A = \pi(r_3^2 - r_2^2); \tag{2.44}$$

r_2, r_3 – duct radii, m.

From equation (2.42) we express w_i:

$$w_i = \frac{x_i \cdot A \cdot v_n}{\sigma \cdot L}. \tag{2.45}$$

According to the law of conservation of mass, the amount of air consumption at a given section i-i should be constant:

$$A \cdot v_{i-1} + \sigma \cdot w_i = A \cdot v_i, \tag{2.46}$$

v_i –air flow speed at the i-th section of the air duct, m/s.

From equation (2.46) we express v_{i-1}:

$$v_{i-1} = v_i - \frac{\sigma}{A} \cdot w_i. \tag{2.47}$$

Let's write the Bernoulli equation for the (i-1)-th and i-th section of the air duct [127]:

$$\Delta p_i + \frac{\rho \cdot v_i^2}{2} = \Delta p_{i-1} + \frac{\rho \cdot v_{i-1}^2}{2} + \kappa \frac{x_{i-1}}{d_e} \frac{\rho \cdot v_{i-1}^2}{2} + 2 \cdot \alpha \cdot \frac{\rho}{2}(v_i - v_{i-1})^2, \tag{2.48}$$

$d_e = 2 \cdot (r_3 - r_2)$ – effective diameter, m;

| – coefficient of frictional resistance, | = 0,01717 [95];

〈 – impact mitigation factor 〈 = 0,4 [94].

Substituting (2.41), (2.45), (2.47) into equation (2.48) and expressing x_i we get [1]:

$$x_i = \frac{L\phi^2\sigma^2 v_i(\kappa x_{i-1} + d_e)}{v_n(\phi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha d_e) - A^2 d_e)} -$$

$$-\frac{\sqrt{L^2\phi^4\sigma^4 v_i^2(\kappa x_{i-1}+d_e)^2-(L^2\kappa\phi^2\sigma^2 v_i^2 x_{i-1}+A^2 v_n^2 d_e x_{i-1}^2)(\phi^2\sigma^2(\kappa x_{i-1}+d_e+2\alpha d_e)-A^2 d_e)}}{v_n(\phi^2\sigma^2(\kappa x_{i-1}+d_e+2\alpha d_e)-A^2 d_e)}. \qquad (2.49)$$

Dependence (2.48) relates the distance xi to the previous distance x_{i-1} and the speed of the air flow at the i-th section of the air duct v_i.

To determine the distance x_i, air flow speed w_i, the area of the holes σ and their number n, we will develop a technique that consists of the following stages [6]:

1. Setting parameters L, φ, 〈, ⏐, d_e, v_n, A, w_1, v_1.

2. Determination of the step of variation of the area of the holes σ = 2.001·j, j – hole number.

3. Distance calculation x_i according to the formula (2.44), i is the hole number.

4. Calculation of air flow rate w_i according to the formula (2.45).

5. Calculation of the sum of distances $L_{calc} = \sum_{i=1}^{n} x_i$.

6. Fulfillment of the condition: if the sum of the distances L_{calc}>L, then the action of point 7 is performed, otherwise point 3 is performed.

7. Determination of the number of holesn = i.

8. To ensure the necessary convergence of the total length of the heat exchanger, we fulfill the condition: if the modulus of the difference of the sum of the distances and the accepted length of the heat exchanger L_{calc} - L < 0,01, then the action of point 9 is performed, otherwise point 2 is performed.

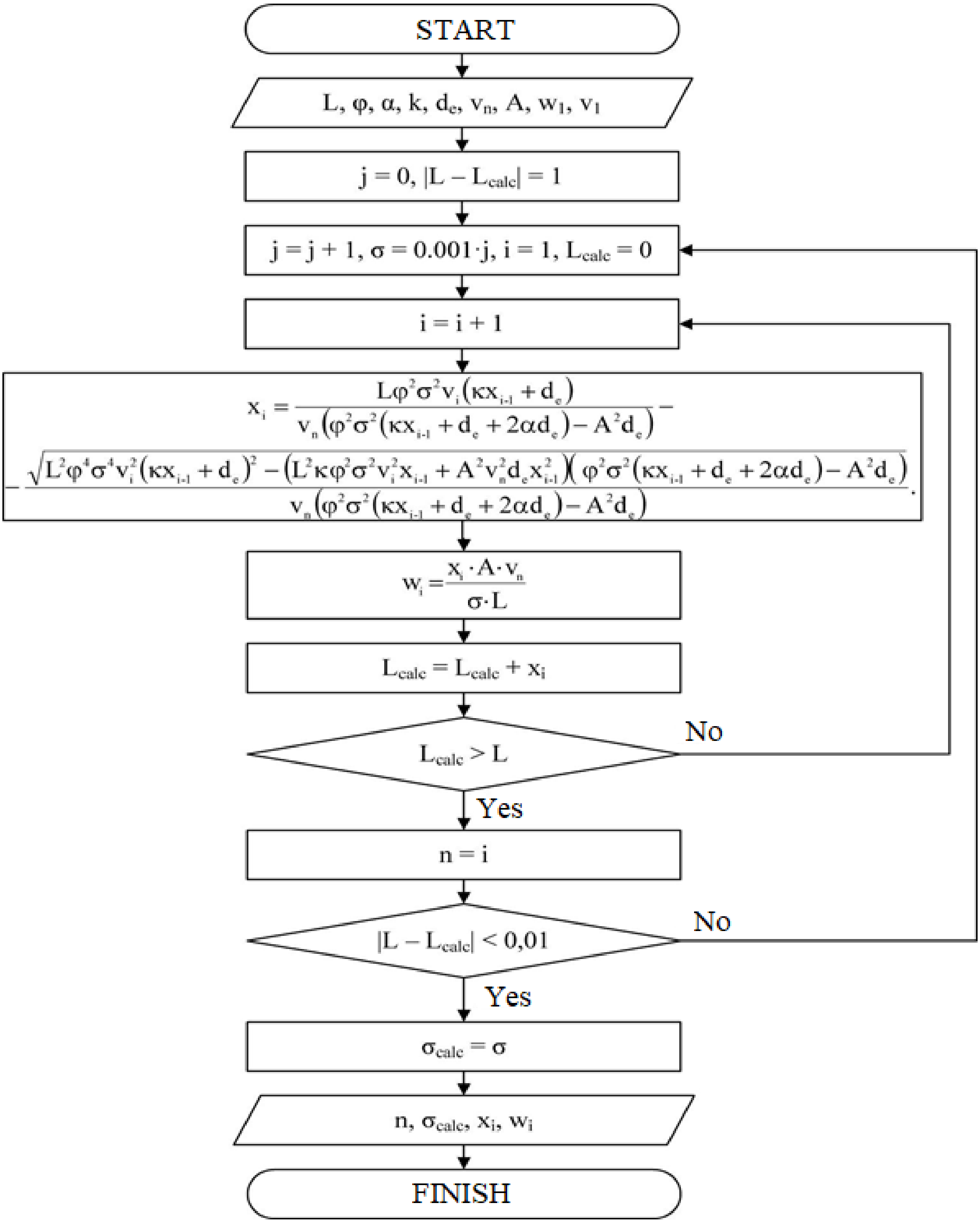
9. Determination of the area of the holes σ_{calc} = σ.

10. Definition of parameters n, σ_{calc}, x_i, w_i.

The developed technique and the algorithm based on it (Fig. 2.24) are implemented in the software package Mathematica.

Accepting design and technological parameters (L = 5,8 m; φ = 0,65; 〈 = 0,4; ⏐ = 0,01717 m; r_2 = 0,14 m; r_3 = 0,2 m; V_0 = 0,14 m^3/s; x_1 = 0,9 m; v_1 = 0 m/s) the number of holes of a three-pipe concentric heat exchanger is determined n = 7 and their area

σ = 0,011 m^2, as well as the distribution of the distance between the holes according to Fig. 2.25 and the air velocities through them (Fig. 2.26).



$$x_i = \frac{L\varphi^2\sigma^2 v_i(\kappa x_{i-1} + d_e)}{v_n(\varphi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha d_e) - A^2 d_e)} - \frac{\sqrt{L^2\varphi^4\sigma^4 v_i^2(\kappa x_{i-1} + d_e)^2 - (L^2\kappa\varphi^2\sigma^2 v_i^2 x_{i-1} + A^2 v_n^2 d_e x_{i-1}^2)(\varphi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha d_e) - A^2 d_e)}}{v_n(\varphi^2\sigma^2(\kappa x_{i-1} + d_e + 2\alpha d_e) - A^2 d_e)}.$$

$$w_i = \frac{x_i \cdot A \cdot v_n}{\sigma \cdot L}$$

Fig. 2.24. Algorithm for calculating the geometry of the location of holes in the air duct of a three-pipe heat exchanger

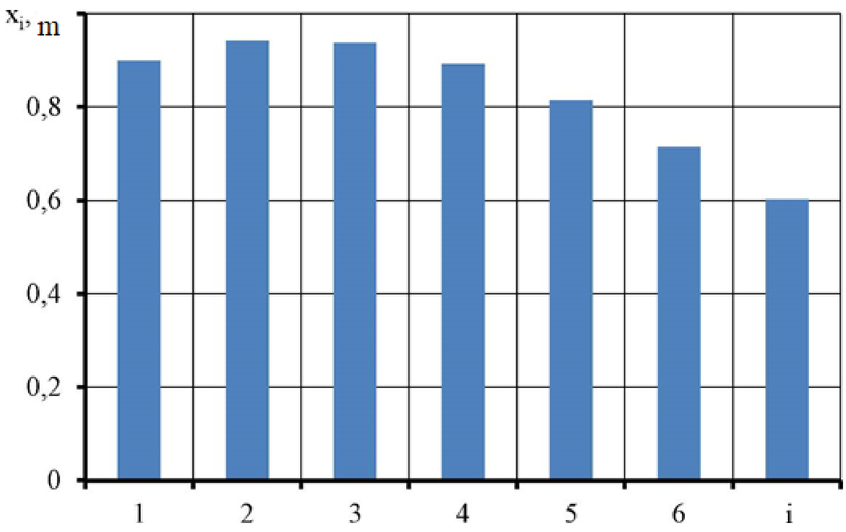


Fig. 2.25. Distribution of the distance between the holes

Analyzing Fig. 2.25 together with Fig. 2.23, it can be stated that the distance between the holes gradually decreases from 0.94 to 0.6 m in the direction opposite to the air flow. However, at the end of the heat recovery duct, there is a slight decrease in distance of 0.04 m, which is caused by the backflow of air impinging on the blocked end. A similar phenomenon is observed with the distribution of air velocities through openings (2.26) [6].

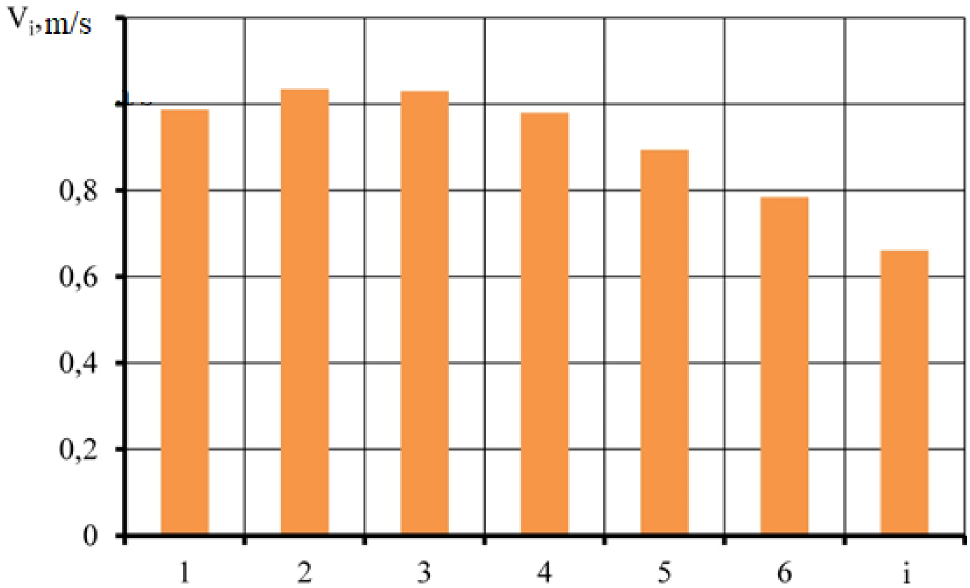


Fig. 2.26. Distribution of air velocities through openings

3 RESULTS OF EXPERIMENTAL RESEARCH VENTILATION SYSTEMS

3.1 Results of experimental studies of the ventilation system of the ground channel

At this stage of experimental research, the scope of research was expanded to the entire building for keeping pigs, a ventilation system with a ground channel was implemented.

The study of microclimate parameters was carried out in the room for keeping piglets at the rearing farm of the «Litagor» farm. The appearance and design of the room and elements of the ventilation system are shown in fig. 3.1.

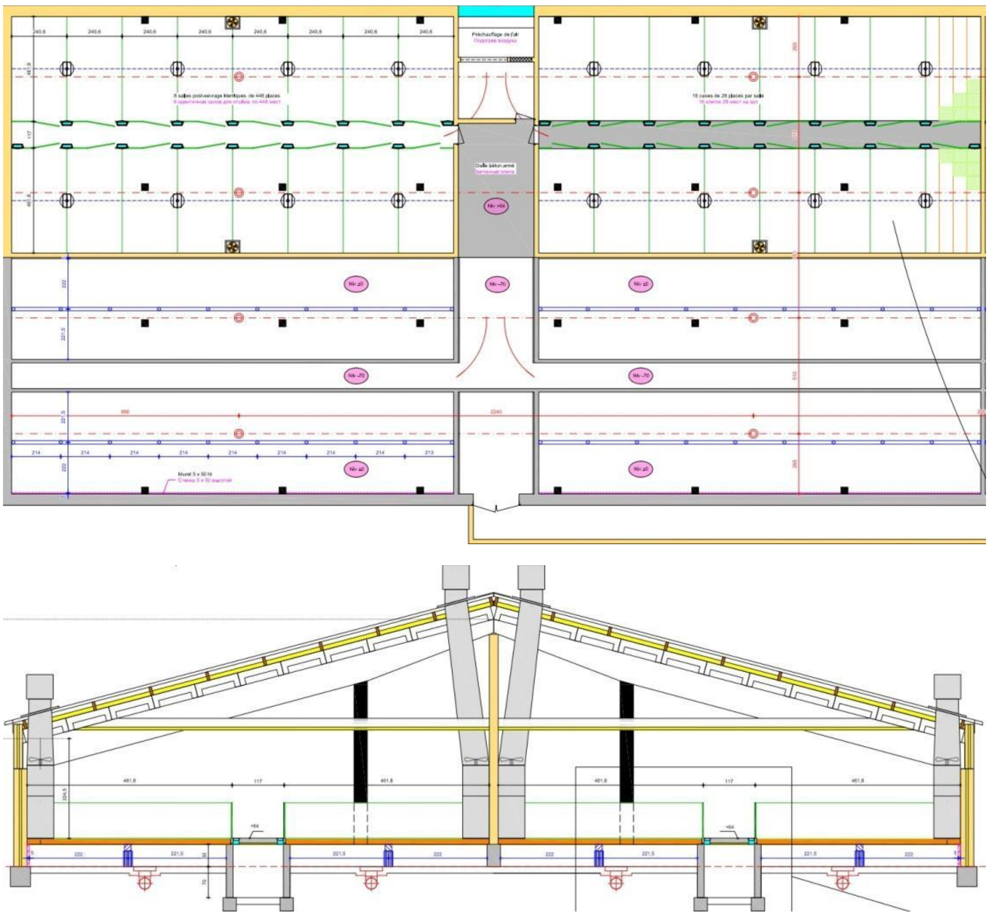


Fig. 3.1. Project of a room with a ventilation system with a ground channel for keeping piglets for growing

The room has a ventilation system that creates negative pressure. This system provides regulation of the microclimate in the room according to the following scheme: air from the environment enters through the adjustment holes located on one side of the room, and then moves under the passages. In the passageways there are equally spaced holes through which air enters the interior of the room.

On the sides of the sections there are exhaust shafts that take air out of the room, creating negative pressure.

Research was conducted in the summer period, namely in July, when the ambient temperature reached 35–38 °C.

Measurements of the specified parameters were carried out at different heights above the floor (30 cm, 70 cm, 160 cm) in the area the animals stay in each machine. In addition, the temperature of the pig's skin at the withers was also measured, as well as the surface temperature of both the solid and the split floor.

Air temperature was measured at 16 different points (group machines). Based on the obtained data, a graph was drawn up, which shows the dependence of the air temperature on the location of the group machine (Fig. 3.2).

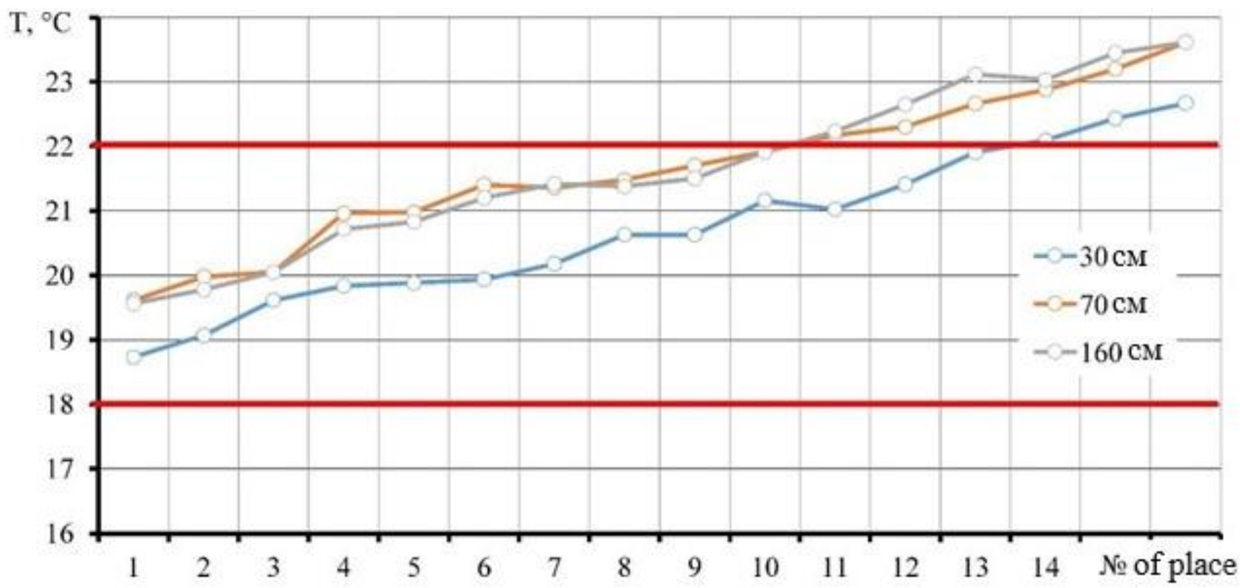


Fig. 3.2 Graph of air temperature changes in group machines

Table 3.1 shows the minimum, maximum, and arithmetic mean values, as well as the deviation of indicators from the arithmetic mean value.

Table 3.1.

Distribution of air temperature in the pig house

Place	Max, °C	Min, °C	Mean, °C	Δ, °C
Solid floor	22,9	21,3	22,1	0,5
Cracked floor	26,2	24,4	25,3	0,5
30 cm	22,7	18,7	20,7	1,2
70 cm	23,6	19,6	21,6	1,2
160 cm	23,6	19,6	21,7	1,3
Animal skin	37,2	36,0	36,6	0,4

Table 3.1 shows that the warmest air is observed at a height of 160 cm from the floor. This may be because warm air tends to rise because it is lighter than cold air. From fig. 3.2 shows that from the 13th group machine, the air temperature in the animal cutting area (height 30 cm from the floor) does not meet the recommended limits. This indicates the need to improve the indoor temperature control system, especially with regard to the comfort and health of the pigs.

In the group machines, the relative humidity of the air was measured, which was in the range from 60.6% to 95.2%, depending on the height above the floor. The obtained data are indicated in the table. 3.2 and in fig. 3.3. As altitude decreases, relative humidity increases. This can be explained by the location of manure channels and the release of moisture from the litter. Comparing the obtained data with the recommended ones, it was found that the air humidity at a height of 30 cm is overestimated, since the recommended range of relative humidity is from 60% to 80%.

Table 3.2.

Distribution of air humidity in the pigsty

Place	Max, %	Min, %	Mean, %	Δ, %
30 cm	95,2	80,8	88,4	3,4
70 cm	88,1	69,3	77,8	4,1
160 cm	76,6	60,6	68,7	3,5

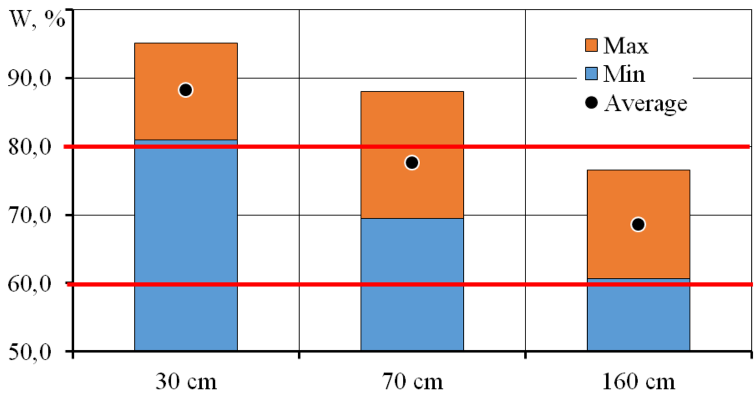


Fig. 3.3. Histogram of the dependence of air humidity on altitude

Indoor air velocities were measured, ranging from 0.12 m/s to 0.81 m/s, depending on the height at which the measurements were taken. The obtained results are presented in the table. 3.3 and illustrated in fig. 3.4. At a height of 30 cm and 70 cm, the air speed is generally within the permissible values, not exceeding 0.2 m/s.

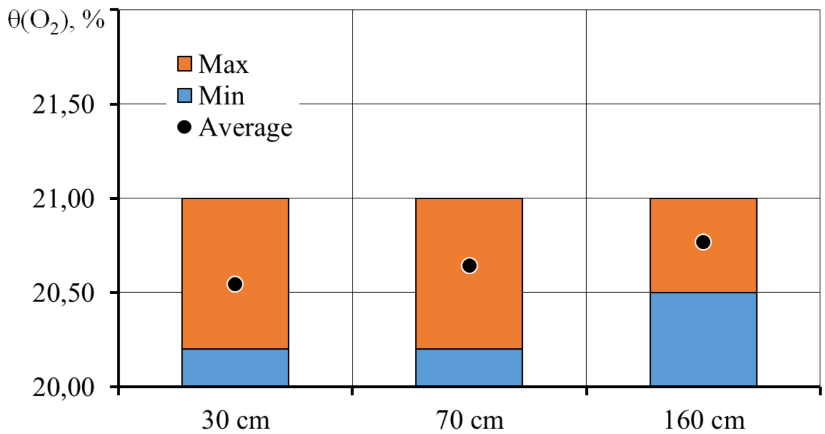


Fig. 3.4. Histogram of dependence of air speed on height

The room uses a ventilation system with negative pressure. Fresh air is supplied from ventilation ducts located under the aisles, which contributes to its immediate impact on the livestock and changes in temperature in the lower part of the room. As the height in the room increases, so does the air speed. This is due to the fact that the exhaust flow of air is carried out in the upper part of the room through exhaust shafts, and the lower part of the room is blocked by obstacles that create resistance to air movement.

Table 3.3.

Distribution of air velocity in the pig house

Place	Max, m/s	Min, m/s	Mean, m/s	Δ, m/s
30 cm	0,42	0,12	0,22	0,12
70 cm	0,53	0,21	0,31	0,11
160 cm	0,81	0,53	0,64	0,12

Indoors, the concentration of ammonia in the air ranged from 5.5 to 16.4 mg/m^3, depending on the height at which the measurements were taken. The maximum permissible concentration of ammonia in the room is 20 mg/m^3. As the altitude decreased, the concentration of ammonia in the air increased. This is explained by the presence of manure channels and pig waste products, which contribute to the release of ammonia into the air. The results of measurements of the amount of ammonia in the air of the room are shown in the table. 3.4 and illustrated in fig. 3.5.

Table 3.4.

Distribution of ammonia content in pig house air

Place	Max, mg/m^3	Min, mg/m^3	Mean, mg/m^3	Δ, mg/m^3
30 cm	16,4	10,4	12,7	1,1
70 cm	13,5	7,7	10,4	1,4
160 cm	9,6	5,5	7,5	1,3

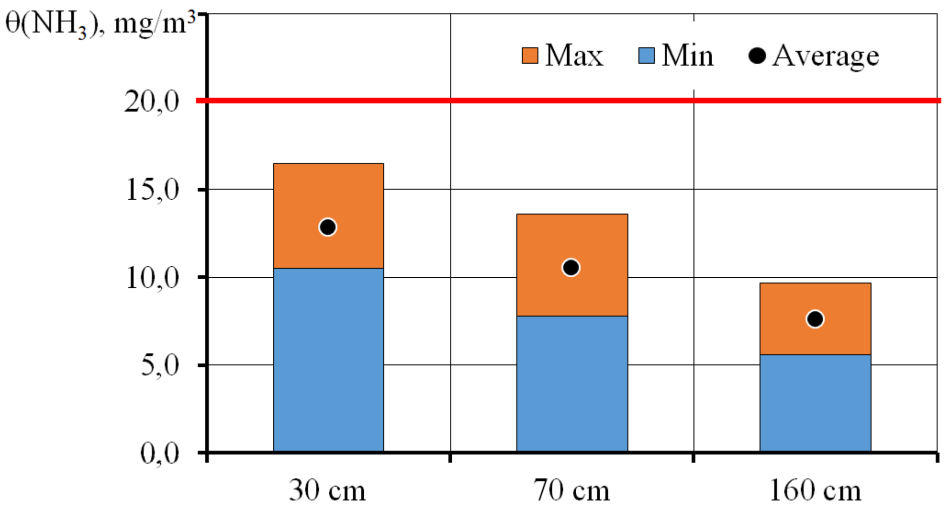


Fig. 3.5. Histogram of the dependence of the proportion of ammonia in the air

The concentration of carbon dioxide in the indoor air ranged from 0.11% to

0.42%, depending on the altitude at which the measurements were taken. The maximum permissible concentration of carbon dioxide in the room is 0.2%. As the height increased, the concentration of carbon dioxide in the air increased. This can be explained by the location of the gas emission sources at the higher levels of the room. The results of measuring the amount of carbon dioxide in the room air are presented in the table. 3.5 and shown in fig. 3.6.

Table 3.5.

Distribution of carbon dioxide content in the air of a piggery

Place	Max, %	Min, %	Mean, %	Δ, %
30 cm	0,24	0,11	0,18	0,03
70 cm	0,36	0,17	0,26	0,05
160 cm	0,42	0,31	0,34	0,04

The concentration of hydrogen sulfide in the room air ranged from 1.31 to 5.17 mg/m^3, depending on the height at which the measurements were taken. The maximum permissible concentration of hydrogen sulfide in the air is 10 mg/m^3. As the height increased, the concentration of hydrogen sulfide in the air increased.

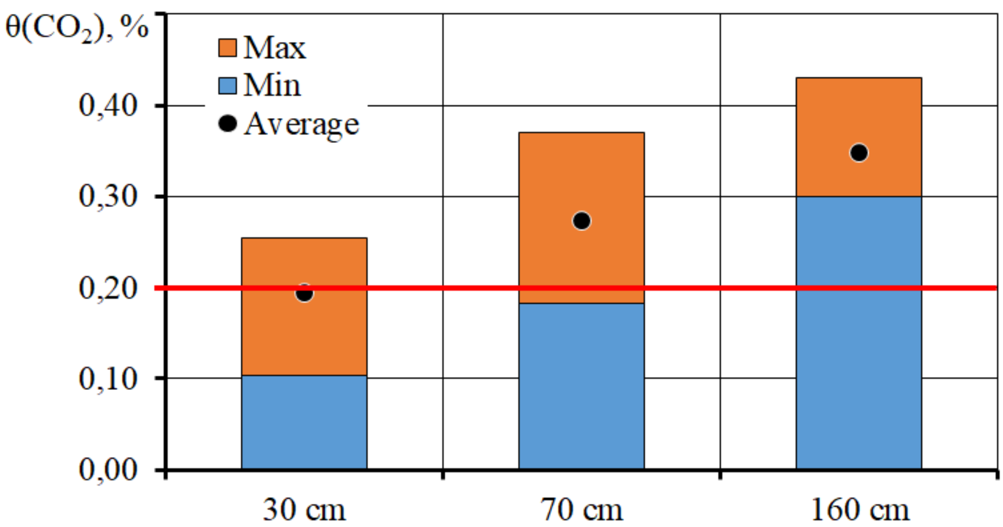


Fig. 3.6. Histogram of the dependence of the proportion of carbon dioxide in the air on altitude

This can be explained by the fact that hydrogen sulfide is released from manure, which is located in manure channels or in litter, and its concentration increases with height above the floor. The results of measurements of the amount of hydrogen sulfide in the air of the room are given in table. 3.6 and presented in fig. 3.7.

Table 3.6.

Distribution of the content of hydrogen sulfide in the air of the pig house

Place	Max, mg/m^3	Min, mg/m^3	Mean, mg/m^3	Δ, mg/m^3
30 cm	5,17	3,35	4,66	0,42
70 cm	4,11	2,01	3,01	0,53
160 cm	2,81	1,31	1,60	0,21

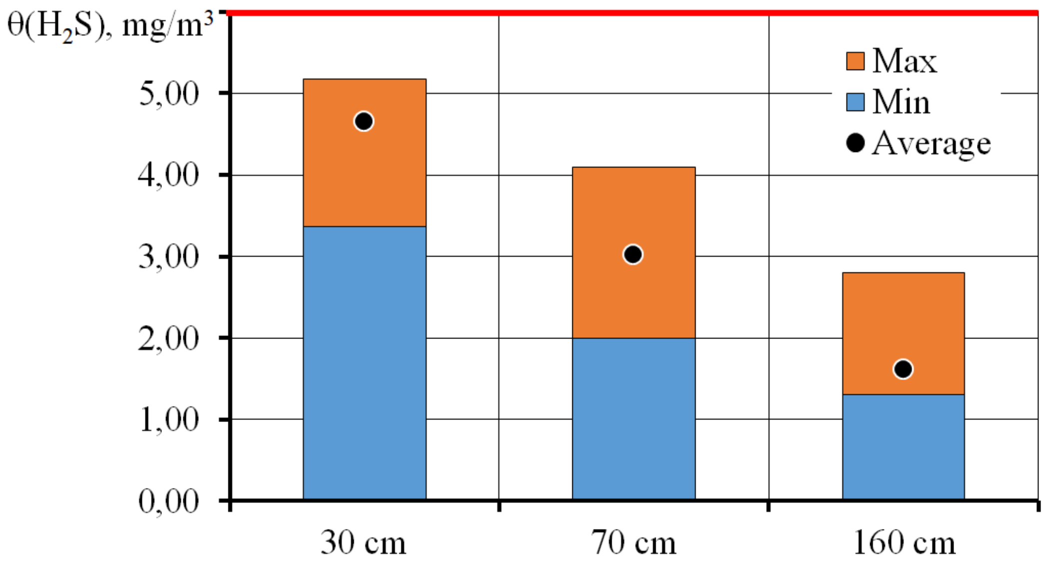


Fig. 3.7. Histogram of the proportion of hydrogen sulfide in the air as a function of altitude

The concentration of oxygen in the air of the room was in the range from 20.1% to 21.2%, depending on the height at which the measurements were made. With decreasing altitude, there is a decrease in the concentration of oxygen in the air, as it is absorbed by pigs in group machines. The results of measurements of the amount of oxygen in the room air are presented in table. 3.7 and illustrated in figures 3.8.

Table 3.7.

Distribution of oxygen content in the air of the pig house

Place	Max, %	Min, %	Mean, %	Δ, %
30 cm	21,1	20,1	20,9	0,12
70 cm	21,2	20,2	20,8	0,11
160 cm	21,1	20,5	20,7	0,13

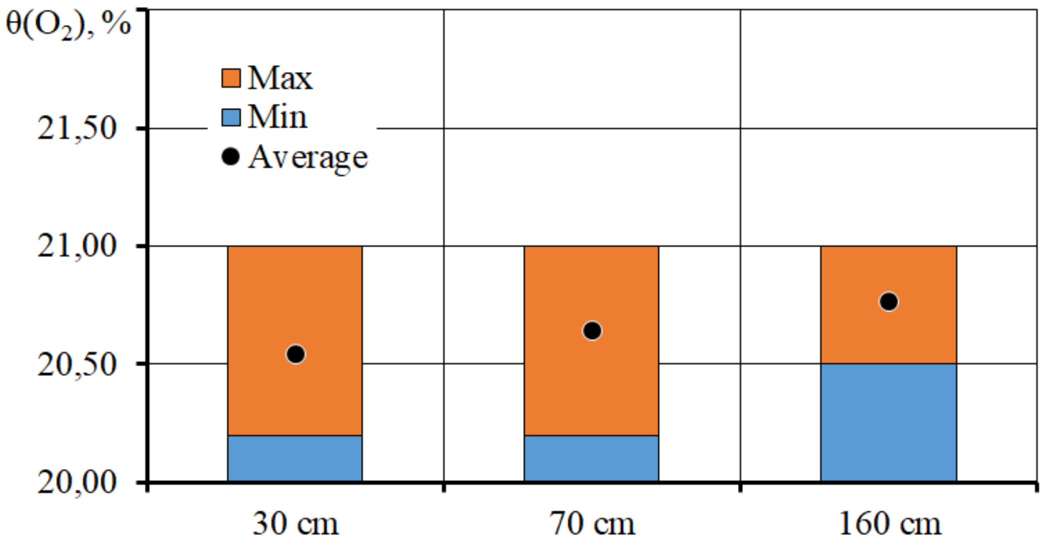


Fig. 3.8. Histogram of the dependence of the proportion of oxygen in the air on altitude

According to the results of the inspection of the room for keeping piglets on growing with a microclimate system with an above-ground channel, the problem of not ensuring the normalized temperature in the area the animals are located after the placement of the 13 group machine along the channel was revealed. It was also found that the relative humidity of the air at the height the animals are located exceeds the recommended standards and reaches 95%, while the recommended humidity of the air for piglets in rearing is no more than 80%. Most indicators of air composition (content H_2S, CO_2, H2S and O_2) in the area the animals stay is within normal limits.

3.2 Justification of the structural and technological scheme of the mechatronic system for ensuring the microclimate of piggery premises

To solve the problems, a structural and technological diagram of a mechatronic system for ensuring the microclimate of piggery premises was developed (Fig. 3.9).

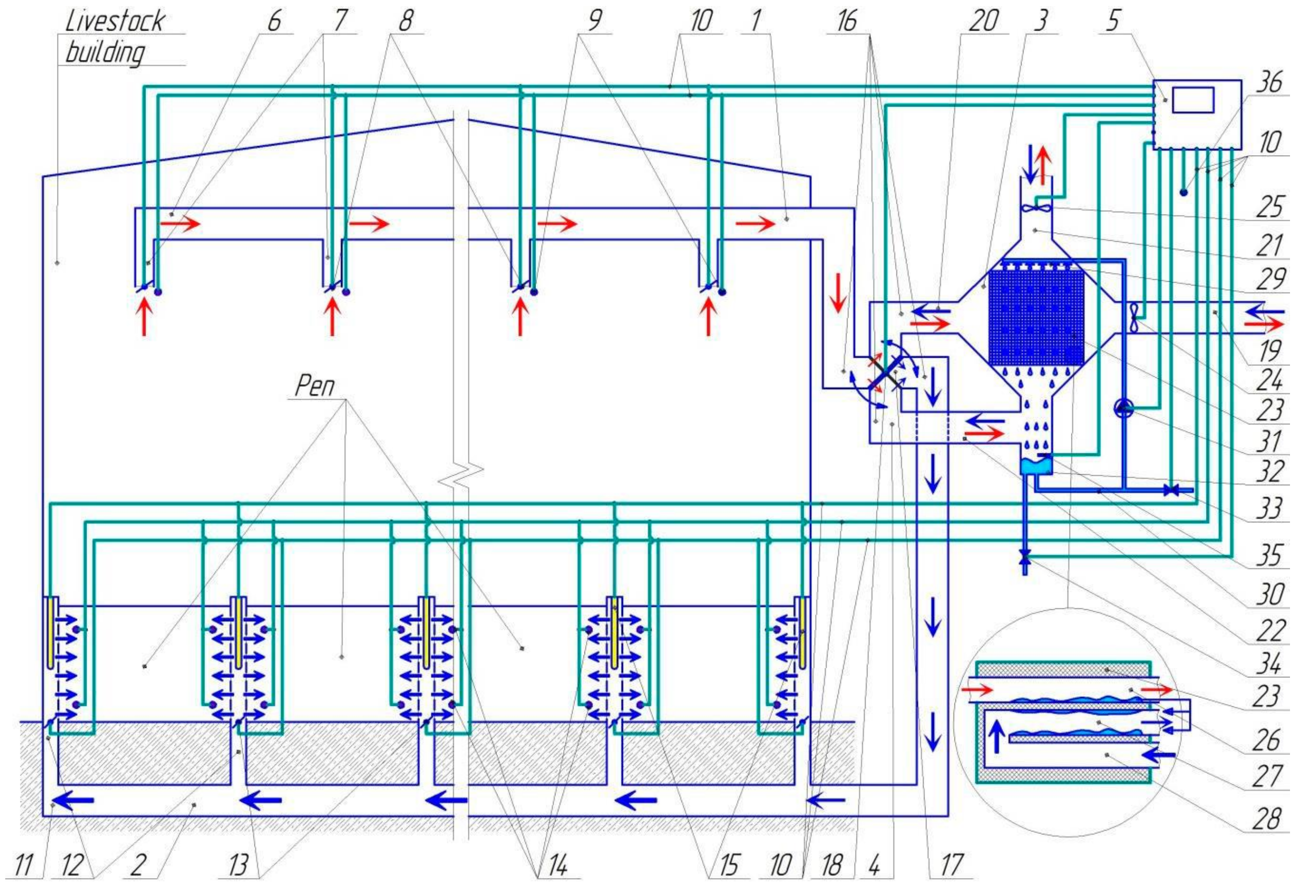


Fig. 3.9. Structural and technological diagram of a mechatronic system ensuring the microclimate of piggery premises:

The mechatronic system for ensuring the microclimate of piggery premises works as follows. The operator on the control unit 5 sets the specified ranges of local microclimate parameter values (temperature, humidity, air flow rate) for each machine animals are kept.

Also, the operator on the control unit 5 sets the limit values of air quality according to the content of hydrogen sulfide, carbon dioxide, and ammonia. Next, the mechatronic system for ensuring the microclimate of piggery premises is launched.

Information on the values of temperature, humidity and air quality (content of hydrogen sulfide, carbon dioxide and ammonia) from temperature, humidity and air quality sensors 9, temperature and air humidity sensors 14, external temperature and humidity sensors 36 is transmitted to the control unit by means of electrical wires 10 5. This information is compared with each other and with the data set by the operator.

If the temperature set by the operator is higher than the temperature on the outside of the pig house (winter period), the control unit transmits a signal to the rotary damper with a servo drive 18, which is set in such a position that allows connecting the contaminated air intake system 1 with the internal nozzle for the working air 20 of the indirect-evaporative type 3 heat exchanger.

In turn, the ventilation system for injecting clean air 2 is connected to the internal outlet for recycled air 22 of the heat exchanger of the indirect-evaporative type 3. The control unit 5 starts the injection fan 24 in the direction of air supply from the piggery to its outer side, and the exhaust fan 25 in the opposite direction of air supply from the outside to the piggery room.

Depending on the air quality above the machines, which is determined using the temperature, humidity and air quality sensors 9 and the limit values set by the operator, the control unit 5 by means of electrical wires transmits a signal to the intake dampers with servo drives 8. If the measured air quality values are lower for the limit values set by the operator, the intake valve with servo drive 8 closes. In the opposite case, the intake valve with servo drive 8 opens to an angle that is directly proportional to the corresponding difference in air quality values and limit values. Air is sucked into the air intake nozzles 7 and is formed into a flow that moves along the central air intake duct 6 of the contaminated air intake system 1. Then the air flow enters the nozzles 16 and the central cavity 17 of the four-way valve 4. After that, the air flow enters the internal nozzle for working air 20 and further to the set of cross channels 23 of the indirect-evaporative type heat exchanger 3. This air flow passes through the working channels 26, it cools and reduces humidity, forming condensate. This process takes place due to heat exchange through the walls that connect the working channels 26 and wet channels 27. Next, the cooled air flow is supplied to the outside of the piggery room through the external nozzle for working air 19.

Simultaneously with the above, the cold dry air flow from the outside of the pig house enters the external outlet for recycled air 21. Next, the cold dry air flow moves to the set of cross channels 23 of the indirect-evaporative heat exchanger type 3, it continues its movement first along the dry channel 28 , and then along the wet channel 27. In the dry channel 28, the cold dry air flow is heated due to heat exchange through the walls connecting the wet channels 27 and the dry channels 28, and in the wet channel 27, the already warm dry air flow is enriched with moisture and continues to heat up.

Excess moisture under the influence of gravity flows downwards into the water intake tank 32. At the initial moment of time, the control unit 5 by means of electric wires closes the electromagnetic valve for draining water 34. In addition, the control unit 5 turns on the water pump 31, which pumps the collected water from of the water intake tank 32 and directs it through the pipeline system 30 to the water supply nozzles 29. Water from the water supply nozzles 29 washes the wet channels 27. The water level in the water intake tank 32 is determined using a level sensor 35, which by means of electrical wires 10 transmits this information to the control unit 5. If the water level is lower than the specified one, the control unit 5 opens the electromagnetic faucet for adding water 33 and water from the water consumption system of the piggery enters the pipeline system 30. In the opposite case, the control unit 5 closes the electromagnetic faucet for adding water 33 and opens the electromagnetic faucet for draining water 34 and water from the water intake tank 32 enters the manure removal system.

After the wet channels 27, the warm, moist air flow enters the internal nozzle for recycled air 22, which is connected to the nozzles 16 and the central cavity 17 of the four-way valve 4. Next, the warm, moist air flow enters the central air duct for air injection 11 of the clean air injection ventilation system 2, through nozzles for injection air 12 enters the middle of the pigsty directly into the machines in which the animals are kept. Due to the fact that the central duct for air injection 11 is located under the floor below the level of freezing of the soil, the process of geothermal heating of the flow of warm moist air additionally occurs.

Information from temperature and air humidity sensors 14 by means of electrical wires 10 is sent to the control unit 5, it is compared with the parameters of the local microclimate set by the operator in each machine separately. If additional heating of the air flow is necessary in a certain machine, the control unit 5 turns on the corresponding heating element 15 and fully opens the discharge valve with the servo drive 13. If the air temperature is higher than or equal to the required one, the control unit 5 turns off the corresponding heating element 15 and partially closes the discharge valve with servo drive 13. The degree of closing of the discharge valve with servo drive 13 is directly proportional to the difference between the set and measured temperature.

Consider the case when the temperature set by the operator is lower than the temperature on the outside of the piggery (summer period). The control unit transmits a signal to the rotary damper with a servo drive 18, which is set in such a position that allows connecting the system of intake of polluted air 1 with the internal nozzle for recycled air 22 of the heat exchanger of the indirect-evaporative type 3. In turn, the ventilation system of injection of clean air 2 connects to the internal nozzle for working air 20 of the heat exchanger of the indirect-evaporative type 3. The control unit 5 starts the discharge fan 24 in the direction of air supply from the outside of the piggery room to the middle of the piggery room, and the exhaust fan 25 in the opposite direction of air supply from the piggery room on its outer side.

Depending on the air quality above the machines, which is determined using temperature, humidity and air quality sensors 9 and the limit values set by the operator, the control unit 5 by means of electrical wires transmits a signal to the intake dampers with servo drives 8. If the measured air quality values are lower for the limit values set by the operator, the intake valve with servo drive 8 closes. In the opposite case, the intake valve with servo drive 8 opens to an angle that is directly proportional to the corresponding difference in air quality values and limit values. Air is sucked into the air intake nozzles 7 and is formed into a flow that moves through the central air intake duct 6 of the contaminated air intake system 1. Next, the air flow enters the nozzles 16 and the central cavity 17 of the four-way valve 4. After that, the air flow enters the internal outlet for recycled air 22 and further to a set of cross channels 23 of the indirect-evaporative type heat exchanger 3. This air flow passes through the dry channel 28 and the wet channel 27, it is heated due to heat exchange through the walls that communicate with the working channel 26. Then the flow air is supplied to the outside of the piggery room through the external outlet for recycled air 21.

Simultaneously with the above, the warm dry air flow from the outside of the pig house enters the external working air nozzle 19. Next, the warm dry air flow moves to the set of cross channels 23 of the indirect-evaporative heat exchanger type 3, it continues to move along the working channels 26, cools and reduces humidity, forming condensation. This process occurs due to heat exchange through the walls that connect the working channels 26 and wet channels 27.

Excess moisture under the influence of gravity flows downwards into the water intake tank 32. At the initial moment of time, the control unit 5 by means of electric wires closes the electromagnetic valve for draining water 34. In addition, the control unit 5 turns on the water pump 31, which pumps the collected water from of the water intake tank 32 and directs it through the pipeline system 30 to the water supply nozzles 29. Water from the water supply nozzles 29 washes the wet channels 27. The water level in the water intake tank 32 is determined using a level sensor 35, which by means of electrical wires 10 transmits this information to the control unit 5. If the water level is lower than the specified one, the control unit 5 opens the electromagnetic faucet for adding water 33 and water from the water consumption system of the piggery enters the pipeline system 30. In the opposite case, the control unit 5 closes the electromagnetic faucet for adding water 33 and opens the electromagnetic faucet for draining water 34 and water from the water intake tank 32 enters the manure removal system.

The cooled moist air flow after the wet channels 27 enters the internal nozzle for working air 20, which is connected to the nozzles 16 and the central cavity 17 of the four-way valve 4. Then the cooled moist air flow enters the central air duct for air injection 11 of the clean air injection ventilation system 2, through nozzles for injection air 12 enters the middle of the pigsty directly into the machines in which the animals are kept. Due to the fact that the central duct for air injection 11 is located under the floor below the level of soil freezing, the process of geothermal cooling of the flow of cooled moist air additionally occurs.

Information from temperature and air humidity sensors 14 by means of electrical wires 10 is sent to the control unit 5, it is compared with the parameters of the local microclimate set by the operator in each machine separately. The control unit 5 turns off all the heating elements 15. At the initial time, the control unit 5 fully opens all discharge dampers with servo drives 13. If the air temperature is lower than required in a certain machine, then the control unit 5 partially closes the discharge damper with a servo drive 13. The degree of closing of the discharge dampers with a servo drive 13 depends directly on the difference between the set and measured temperatures.

4 ANALYTICAL STUDIES OF TECHNICAL AND TECHNOLOGICAL EQUIPMENT AND WASHING SYSTEMS OF MILKING INSTALLATIONS

4.1 Factors and consequences of ineffective washing of milking installations

According to the results of analytical studies, it was established that the milk released from the udder is practically sterile (with the exception of the first jets, which form a so-called «microbial plug» and must be milked separately). However, changes in the physical and chemical composition of milk occur during the passage through the milking equipment. The movement of milk through the milk duct causes its bacterial contamination, and by the time it enters the milk container, a certain microflora is already formed in it. The quantitative and qualitative composition of this microflora changes and develops depending on the conditions of storage and transportation of milk, which affects the sanitary and hygienic indicators of the raw material during its transfer for processing.

The change in the bacterial insemination of milk during its movement through the milking line of the milking plant based on the averaging of data by V.T. Dmytriv [96] is presented in fig. 4.1., according to which it is possible to note the growth of bacterial insemination of milk as it progresses through the technological line.

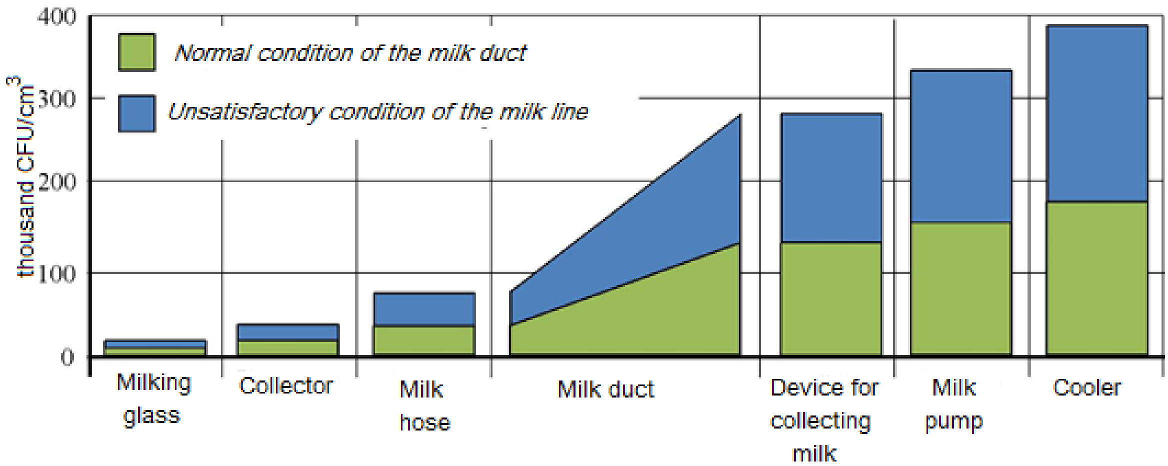


Fig. 4.1. Changes in the bacterial insemination of milk during its movement through the milk line of the milking plant

Inadequate cleaning of milking equipment and the lack of operational methods of controlling the quality of washing lead to contamination and accumulation of deposits on the internal surfaces of milk duct systems (Fig. 4.2).

Microbial contamination of milk occurs all the way from milking to processing. The speed of accumulation and the dynamics of the development of certain types of microorganisms depends on the sanitary condition of milk lines, milk storage conditions and temperature regime [97].

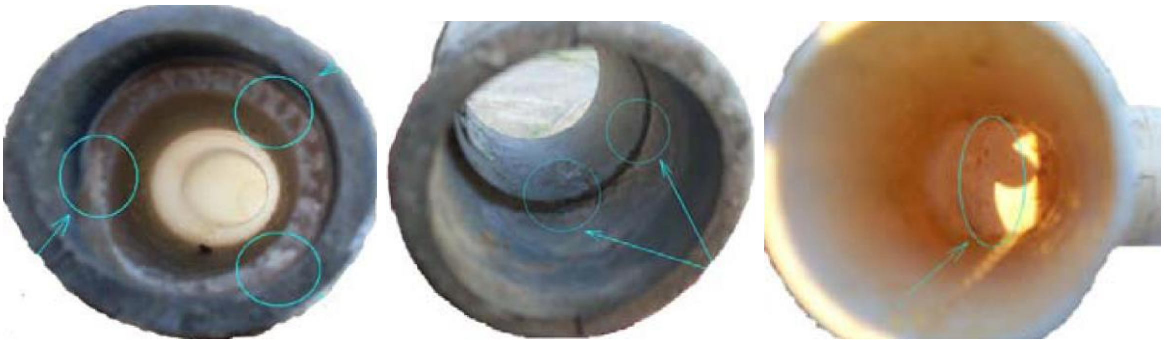


Fig. 4.2. Fragments of mechanical pollution in separate areas of milk duct systems [97]

If the process of cleaning and disinfection of the milk line of the milking plant is not performed sufficiently, milk residues accumulate on its surface in a short time (20–40 min.), creating a favorable environment for the reproduction of microorganisms. For example, under such conditions, lactic acid bacteria double their number in about 40 minutes, and bacteria of the Escherichia coli group - in 20 minutes at a temperature of 30°C. Thus, in the period between milkings (9 hours with two milkings), the amount of microflora can increase approximately 17 thousand times. Bacteria remaining after disinfection in the amount of 2% in a medium contaminated with lipids and proteins can recover their numbers in about 3.5 hours.

The quality of milk and the safety of its consumption largely depend on the cleanliness and sterility of the milk line of the milking plant.

Changes in the composition of milk caused by the metabolic activity of microorganisms, characterized by the appearance of taste and aromatic substances, displacement pH, a decrease in the stability of casein (reduced stability to heat, spontaneous coagulation) [98].

A change in the initial properties of milk as a result of bacterial processes is possible only when the number of microorganisms exceeds 100 thousand CFU/cm^3 and are clearly manifested when the number of microorganisms is more than 500 thousand CFU/cm^3. Therefore, the permissible level of various groups of microorganisms in milk is of great importance [98].

The most important operation for the care of the milk line of the milking plant and milk equipment is its washing. The main task of washing milking equipment is to remove various impurities (milk residues, dirt, bacterial accumulations and other particles and substances) from its inner surface, which is in contact with milk. Moreover, the milk film and fat are not only a favorable environment for the rapid reproduction of bacteria, but also the cause of premature wear of rubber parts.

4.2 Analysis of technical and technological support for washing milking installations

4.2.1 Types of milking plants and milking equipment

All types of milking plants (milking plants in which milking takes place in buckets or directly into cans; milking plants equipped with milk pipes; milking plants with recording milk meter) according to ISO 3918 [99], ISO 5707 [100], ISO 6690 [101] are a complex hydraulic network that includes several types of hydraulic tracts that differ in their parameters. These include (Fig. 4.3):

- milk ducts, through which the flow of the milk-air mixture moves;
- vacuum lines with single-phase air flow;
- milk collectors (or milking buckets, cans, recording milk meters), where, thanks to the large volume of internal space, the flow rate approaches zero and the liquid and gaseous phases (milk and air) are separated;
- milking machines in which a pulsating mode of flow of both milk and air is used to create pulsations.

The entire hydraulic system of a typical milking plant is a non-hermetic system with a corresponding volume. Air leakage into the hydraulic system is associated with:

- air consumption for pulsations in milking machines;
- insufficient tightness between milking cups and teats;
- air flowing into the milking cups with an open valve when putting on and removing them from the teats;
- leakage of pipeline connections.

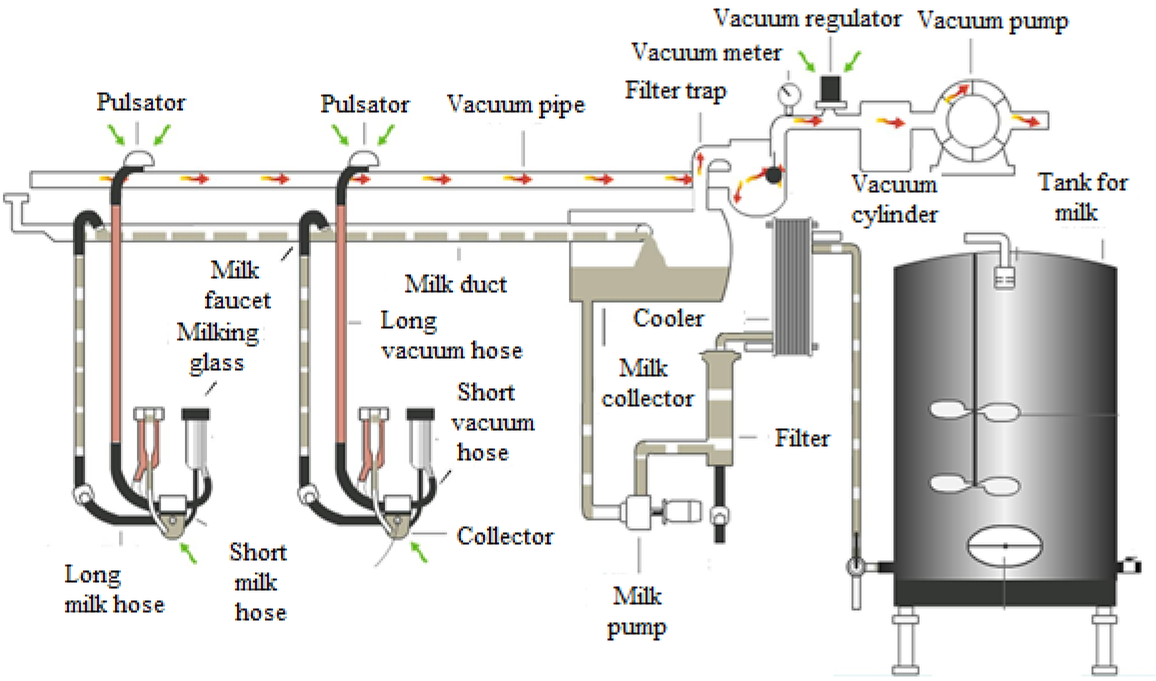


Fig. 4.3. General scheme of the milking plant

Equipment for milking cows from the organization of the production process can be divided into the following categories: for milking in stalls in portable buckets (AD-100B, DAS-2B); for milking in stalls through a milk duct (ADM-8A, UDM-200); for milking on pastures and playgrounds (UDS-ZB, UDL-F-12, K-R-10); for milking in milking parlors («Tandem», «Yalynka», «Carousel», «Parallel»), for milking in mobile milking units (UDP-1; AID-2; UDY-1) [102].

Due to the fact that currently milking machines are usually used for milking in stalls through a milk pipe and for milking in milking parlors, we will consider their washing systems in the future. In its general form, it can be represented as in fig. 4.4.

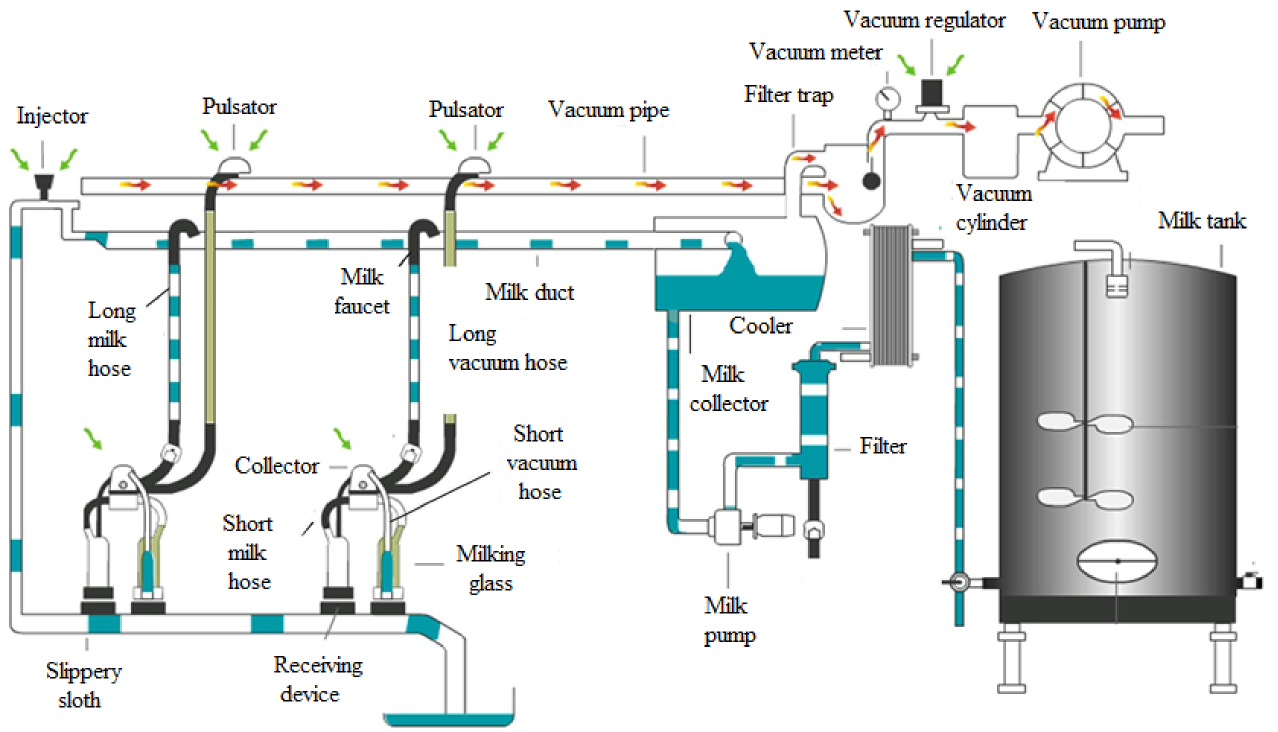


Fig. 4.4. The general diagram of the milking plant flushing system

Analyzing different technological schemes of milking, starting from manual and ending with an automated milking system, the trend of development of milk-conducting communication systems is observed. Modern milk lines are becoming more and more hermetic, their length is increasing, as a result of which the surface area with which the milk is in contact increases dramatically. This complicates the washing process, which can lead to a decrease in the quality of the obtained milk.

The calculation [102] shows that during manual milking, the contact area of milk with the equipment is only 7 m^2, on installations with portable buckets, the total surface in contact with milk is about 20 m^2, on milking installations in stalls through the milk pipe - up to 100 m^2, in milking parlors - up to 45 m^2 (Fig. 4.5). As a result of the increase in the surface area in contact with milk, the possibility of bacterial contamination of milk increases and, accordingly, the costs of resources for washing and cleaning the milk line increase.

As can be seen from the histogram (Fig. 4.5), with the ratio of the level of automation of the milking process and the area of the smallest contact of the equipment with milk, the use of milking parlors is relevant. Note that in research [102], it was proven that bacterial contamination is much lower on installations with a short milk duct of the «Jalynka», «Tandem», «Parallel» or «Carousel" type than on installations with a stall milk duct. Thus, to carry out an effective washing process, it is important that the design of the milking unit has a less branched milk line with the shortest length.

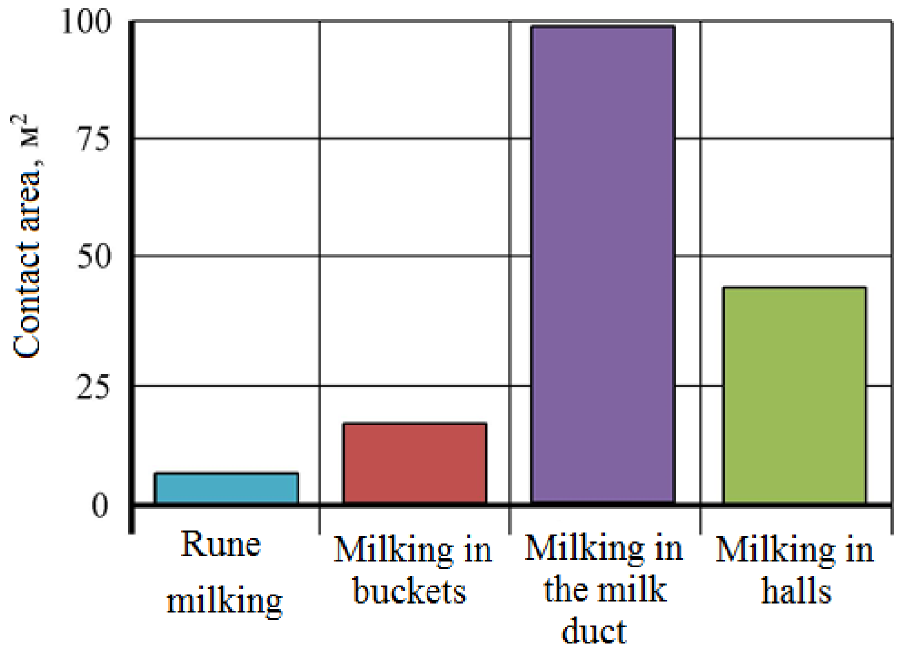


Fig. 4.5. Histogram of the contact area of milk with milking equipment

According to the ISO 5707 standard [100], the design of the milking plant must also provide:

- cleaning of milk residues and deposits on the inner surface of the milk line;
- cleaning the surfaces and cavities of the milk line from residues of detergents and disinfectants;
- reduction of bacterial contamination of surfaces to an acceptable level.

Nodes and parts of communications of the milking installation that come into contact with milk must be made of materials intended for these purposes. According to the ISO 4288 [103] standard, the surface should have roughness 2.5 microns. The surface roughness of the welds should not exceed 16 microns. Full draining of liquid from all parts of the milk line should be ensured.

4.2.2 Classification of the washing system of milking installations

Washing systems, which are equipped with modern milking installations, can be classified according to the method of passage of the washing solution, according to the frequency of filling the milk duct system, according to the method of formation of the cork flow of the washing solution, according to the automation of the technological process (Fig. 4.6).

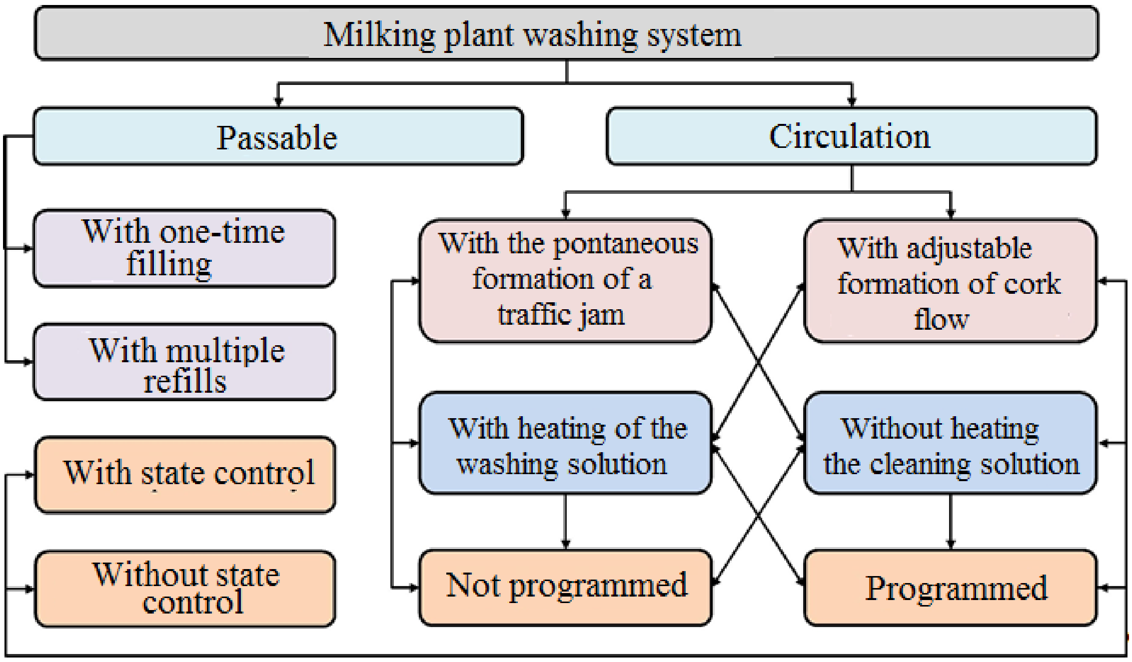


Fig. 4.6. Classification of the washing system of milking installations

According to the method of passage of the washing solution, there are through-through (one-time suction method) and circulation systems for washing milking units. The continuous method of washing milking units is used before milking when rinsing the milk duct systems, as well as when milking in a bucket. After milking, when performing the technological process of washing milking units with a milk pipe (upper or lower), the circulation method is used [104].

According to the frequency of filling of the milk duct system, there are washing systems with one-time and multiple filling. Flushing systems with repeated filling are used mainly on milking installations with a short length of milk duct systems, which do not require a large amount of water for flushing.

To increase the efficiency of circulation washing, systems with spontaneous and regulated formation of a cork flow of washing solution are used. The latter provide a reduction in the specific consumption of water, electricity, consumption of detergents and disinfectants.

Depending on the type of equipment used for milking, programmable and non-programmable flushing systems are used. These systems are used for automatic washing of milking equipment. The possibility of programming allows you to set certain time parameters of the flushing system and adapt it to different types of milking installations.

When washing the milking equipment, systems with heating and without heating of the washing solution are used. In winter, it is better to use a system with heating, which allows you to keep the temperature of the washing solution at the necessary level for effective washing of the milking unit. In the presence of a programmable control system, special sensors are used to determine the state of cleanliness of the milk line of the milking plant.

When washing milking machines, almost all washing systems provide for two options for washing stands: with washing heads or bowls for each milking cup, or a common tank the milking cups are immersed.

4.2.3 Analysis of known designs of washing systems of milking plants

Many works, including O. O. Dergacheva, V. I. Andriychuk, L. V. Savchuk, A. P. Palii, and D. J. Reinemann, were devoted to the study of the problem of washing the milk ducts of milking plants.

In his work, A. I. Fenenko [105] considers the process of removing milk fat particles under the influence of tangential friction forces that overcome the adhesive adhesion of milk fat to the surface of the milk line. Scientists [105, 106] more fully conducted studies of the structure of the film of milk contaminants.

D. V. Pogorelov [107] in his work to create a cork flow regime during washing proposes a washing method with preliminary accumulation of washing liquid in a container located between the washing pipeline and the milk pipeline (Fig. 4.7).

The scheme works as follows: liquid under vacuum is sucked by milking devices from the washing device into the washing pipeline 3 and then enters the tank 4, which is connected to the vacuum through the controlled valve 6, while the lower valve 5 is closed. As a result of filling the tank by the amount V_p the upper level sensor is triggered, while valve 6 closes and valve 5 opens and the liquid begins to flow into the milk pipe at a high speed. Since the distribution valve is closed, the main flow is directed into the ringed milk pipe, and a smaller one (about 5%) flows through the hose 9 into the milk tank 2. As soon as the liquid level in the reservoir 4 drops to the lower level, valve 5 closes and valve 6 opens. A rarefaction is again created in tank 4 and the liquid is sucked into it with high intensity. The liquid moves through the milk duct under pressure at high speed when the pipe section is completely filled. At the same time, the volume of liquid in the tank is regulated V_p should be approximately equal to the volume of the milk container, so that it does not overflow, since the instantaneous supply of liquid to the milk line will be much greater than the throughput of the milk pump of milk container 2.

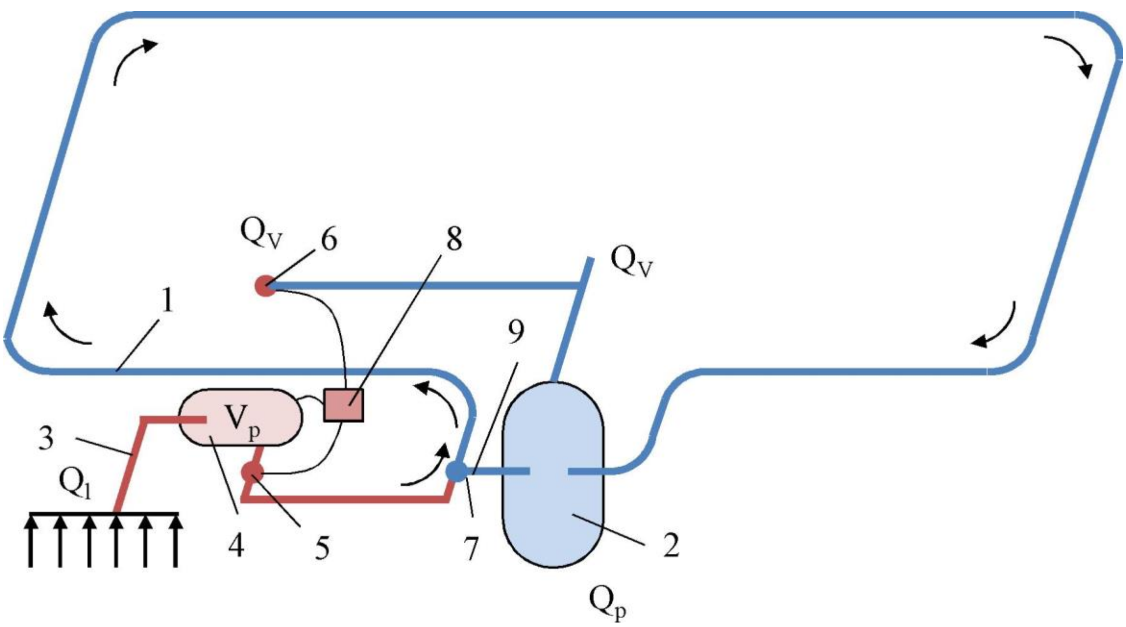


Fig. 4.7. Scheme of circulation washing of the milk duct
with preliminary accumulation of washing liquid [163]:
1 - milk duct; 2 – milk container; 3 – washing pipeline; 4 – tank;
5,6 – controlled valves; 7 – three-way faucet; 8 – control unit; 9 – hose

The disadvantage of the presented design is the excessive consumption of the washing solution and the inability to control the hydrodynamic mode of movement of the air plug, which can cause a sufficiently strong hydraulic shock that can destroy the elements of the milk line.

In order to intensify the process of washing the milk duct from contamination in his work, Yu. Linnyk [108] suggests during circulation washing and disinfection to periodically supply additional air with the help of a pulse amplifier based on the collector of a three-stroke milking machine and elastic plugs for mechanical cleaning of the inner surface of the milk duct (Fig. 4.8) [108].

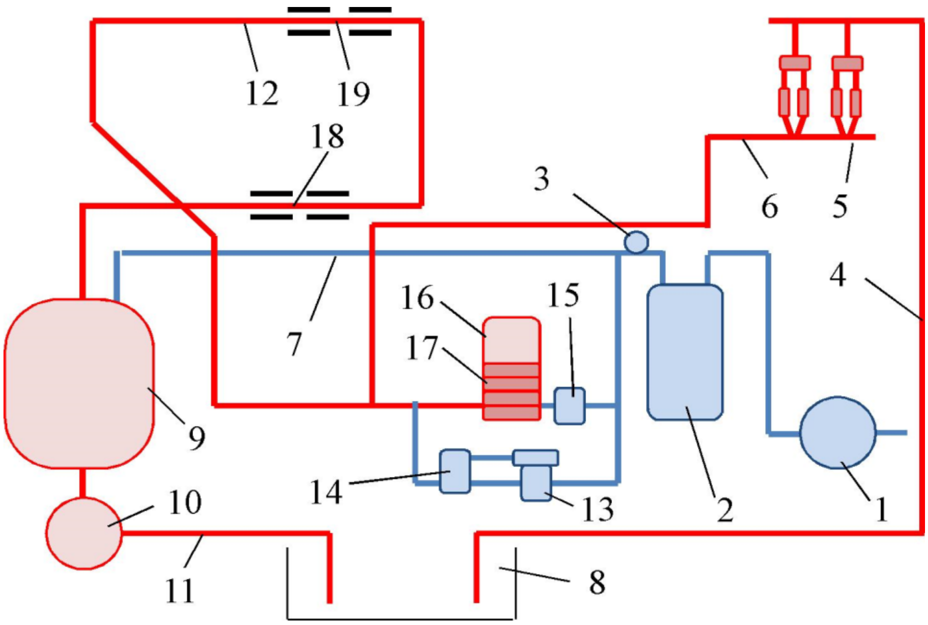
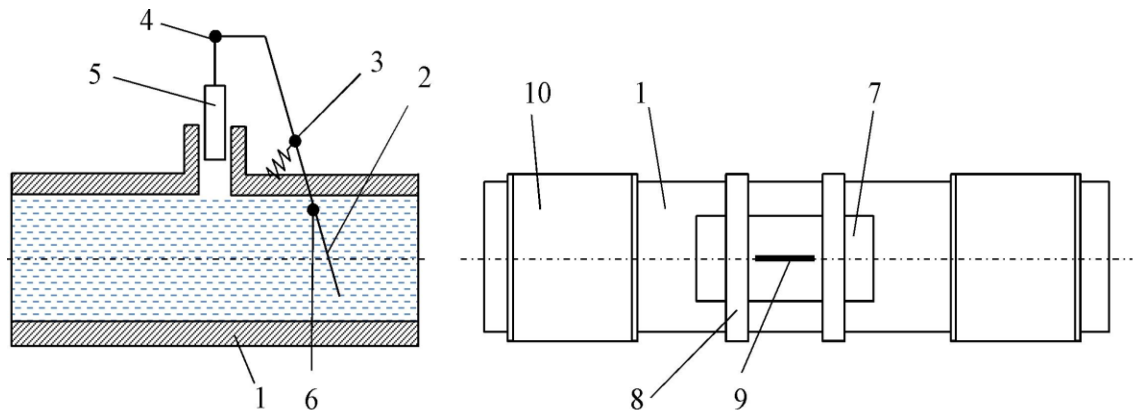


Fig. 4.8. Scheme of the installation for researching milk duct washing modes:
1 – vacuum pump; 2 – vacuum cylinder; 3 – vacuum gauge; 4 – pipeline;
5 – milking machines; 6 – collector pipe; 7 – vacuum line; 8 – bath;
9 – milk receiving tank; 10 – milk pump; 11 – drain pipeline; 12 – milk duct;
13, 15 – pulsators; 14 – pulse amplifier; 16 – grocery store; 17 – spring plugs;
18 – researched insertion of the milk duct; 19 – a transparent milk-conducting insert

The specified method of creating a cork flow in the milk pipeline system deserves attention, but its use when washing milk pipelines of different configurations is not always advisable. As practice has shown, elastic pjs are used only to remove the remains of milk or washing liquid, and cannot be used for mechanical removal of dirt in the milk duct. In addition, in the specified design of the washing system, it is not possible to automatically adjust the operational parameters of the air plugs, which can lead to poor cleaning of the surfaces of the milk pipelines or to a significant increase in pressure and the formation of a hydraulic shock of sufficient force to destroy the walls of the milk pipeline. Also, only one air plug injector is installed in the system, which makes it impossible to clean internal surfaces at a long distance from it.

Palii A.P. proposed a device for washing the milk duct systems of milking plants [109], which consists of a rod with a spring 3 that rotates on an axis 6. At the ends of the rod 3, there are a pressure valve 2 for washing with a washing liquid and an air valve 5, which connected to rod 3 using hinges 4. Rod 3 is held in a certain position by a spring element (Fig. 4.9, a). A. P. Palii also notes that in production conditions, a device for determining the sanitary and hygienic condition of milking equipment can be used to determine the cleanliness of milking equipment (Fig. 4.9, b) [109]



a b

Fig. 4.9. Schemes of technical solutions of elements
of the washing system of milking installations [109]:
a – device for washing milk-conducting systems of milking plants;
b – device for determining the sanitary and hygienic condition of milking equipment; 1 - milk duct; 2 – pressure valve; 3 – rod; 4 – hinge;
5 – air valve; 6 – axis; 7 – plug; 8 – clamps; 9 – plate; 10 – clutch.

Among the shortcomings of the specified equipment should be attributed the lack of control of the parameters of the turbulent movement of the detergent by the milk-conducting system. And the disadvantage of the device for determining the sanitary-hygienic condition is that it does not contain specific steps for determining the effectiveness of the sanitary treatment of the milk pipeline in general and cannot be used to quickly determine the state of contamination during the technological washing operation.

Automatic circulation washing systems of milking units deserve special attention. According to the ISO 5707 standard, all milking installations must be equipped with a washing machine [120].

Despite the structural differences, automatic washing systems of different companies have the following main blocks: software, block of executive bodies, dosing device and liquid tank. Additional equipment with a heater is possible. Based on the possibility of block implementation of the flushing system, a number of companies based on the basic model supply various automatic flushing systems.

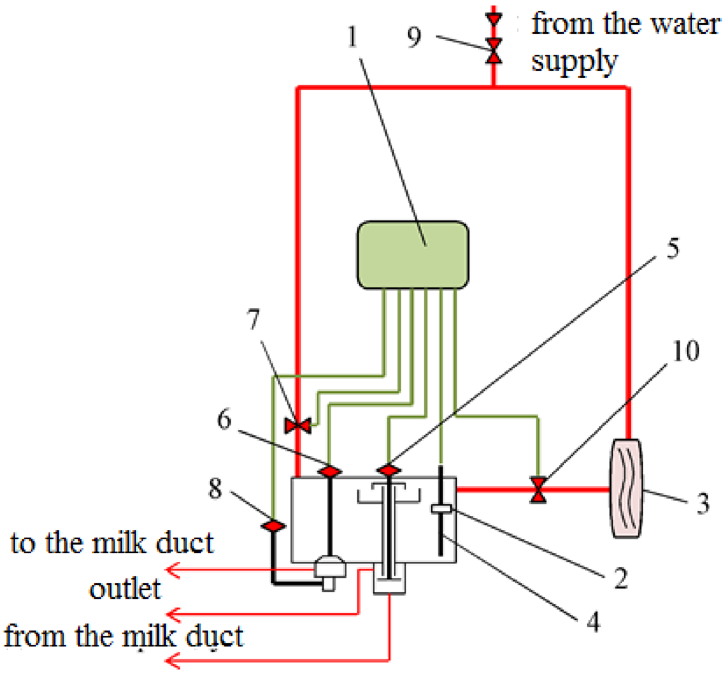
The main task, which is set in the improvement of the washing systems of milk duct systems, is resource saving, first of all, reducing the consumption of air, hot water, energy and operating costs.

The variety of operating conditions and types of milking equipment, on the one hand, and on the other, strict requirements for the quality of washing, led to the need to abandon rigid programs and to create programmable washing machines. The latter allow, depending on the conditions, to change the washing program, the duration of cycles, to monitor the execution of the process, and to signal failures in a timely manner.

The well-known washing machine AP-03 (BUAP-03), produced by TDV»Bratslav» [110] consists of a tank, a float, a sensor for the level of washing solution, a circulation-drain valve, a valve for sucking liquid into the milk pipe, a pipeline, an electromagnetic valve for starting cold water, hot water solenoid valve, water heater, detergent concentrate solenoid valve, tank, temperature control sensor and control unit (Fig. 4.10).

The disadvantages of the specified equipment include the impossibility of monitoring the state of contamination of the milk line, vacuum pressure, temperature and, accordingly, changing the mode parameters of its operation, which leads to insufficient quality of the technological washing operation.

To solve the problem of improving the quality of washing and saving resources, considerable attention is paid to the creation of a controlled cork flow. This allows, firstly, to ensure high quality washing with large lengths and diameters of milk pipes and, secondly, to significantly save the volume of circulating water and the consumption of chemical reagents [110].



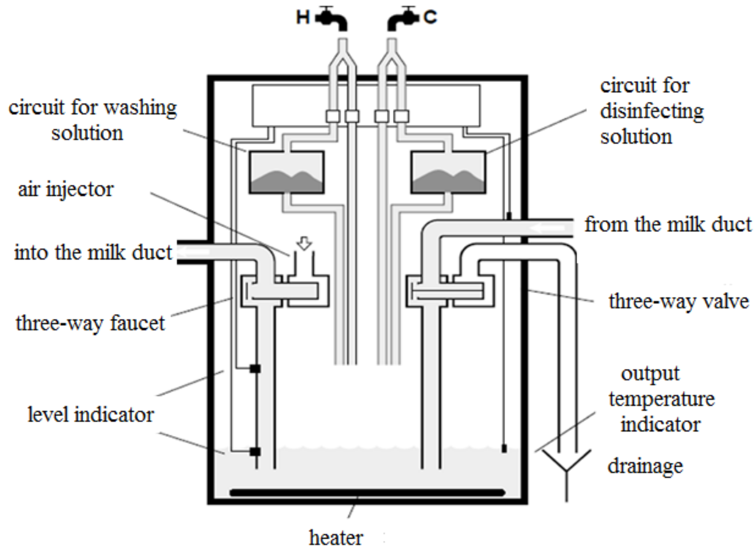
a

b

Fig. 4.10. Scheme (a) and general view (b) of the washing machine AP-03 (BUAP-03), produced by TDV «Bratslav» [110]:
1 – control unit; 2 – float; 3 – water heater; 4 – sensor of the washing solution level; 5 – circulation-drain valve; 6 – liquid suction valve on the milk line of the tank; 7 – solenoid valve for starting cold water; 8 – air start valve; 9 – tap for water connection from the water supply; 10 – hot water solenoid valve

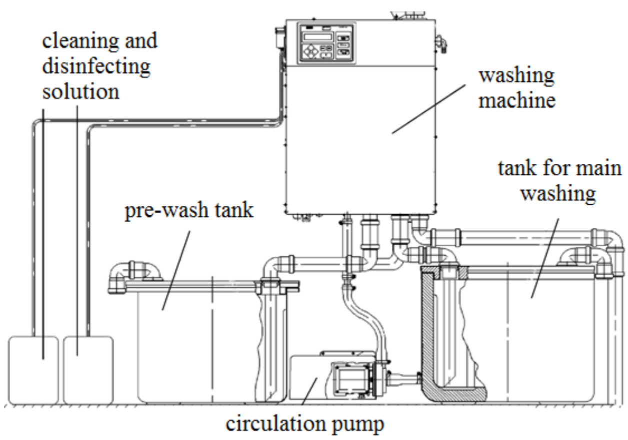
The specified resource-saving mode is used by the DeLaval company in C100E washing machines (Fig. 4.11), by GEA Farm Technologies in the SineTherm washing machine (Fig. 4.12), and others. Also, air injection for effective circulating washing of the milking unit is used in MiniWash machines of the PANAzoo company and TOP WASH of the InterPuls company (Fig. 4.13). In particular, the SAC company offers flushing systems with spontaneous formation of liquid plugs, and does not recommend the use of the Uni-Air-Pulse device, believing that it does not affect the efficiency of washing milking equipment. However, in the cork washing mode, additional air consumption increases, which leads to an increase in the load on the vacuum pumps and, as a result, an increase in energy consumption.

a



b

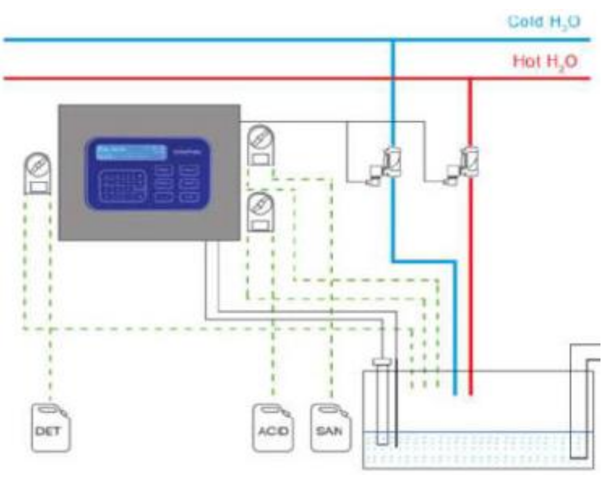
Fig. 4.11. General view (a) and diagram (b) of the C100 washing machine manufactured by DeLaval [101]



a

b

Fig. 4.12. Scheme (a) and general view (b) of the GEA Farm Technologies SineTherm washing machine [102]



a b

Fig. 4.13. Scheme (a) and general view (b) of the TOP WASH washing machine of InterPuls [103]

Table 4.1 shows that almost all companies abandoned low-capacity systems with multiple filling of tanks and switched to systems with one-time filling. Depending on the size of the milking unit, the washing machine is equipped with tanks of different capacities.

Table 4.1.

Comparative characteristics of washing machines

Producer Functions	DeLaval (C100)	GEA Farm Tech. (SineTherm)	InterPuls (TOP WASH)	PANAzoo (MiniWash)	Bratslav (АП-03)
Location of the control unit	in the case	in the case	removable	in the case	removable
Management of the flushing system	micro processor	microprocessor/ electronic fur	microprocessor/ electronic fur	microprocessor	microprocessor
Control of the drain-circulation valve	separate	combined	combined	combined	separate
Air intake valve location	in the case	removable	-	removable	in the case
Availability of a water heater	there are	there are	there are	there are	there are

Producer Functions	DeLaval (C100)	GEA Farm Tech. (SineTherm)	InterPuls (TOP WASH)	PANAzoo (MiniWash)	Bratslav (АП-03)
Availability of injectors	there is no	there are	there are	there are	there is no
Creating controlled traffic jams	there is no	uncontrolled	uncontrolled	uncontrolled	there is no
The method of dosing detergents	automaton	semi-automatic/ automaton	manual/ semi-automatic/ automatic	manual/ semi-automatic/ automatic	manual/ semi-automatic
Detergent regeneration system	there is no	there are	there is no	there is no	there is no
Tank volume, liters	40; 80; 160	20; 70	35; 70; 160	70	160

4.3 Analysis of mode parameters of technical and technological support for washing milking installations

The analysis of literary sources showed that, despite the difference in the quantitative characteristics of the process of washing milking plants (temperature, duration), in general, there is a single opinion on the composition of operations and the sequence of their execution, namely [114]:

- preliminary rinsing and removal of milk residues with warm water (30-48 ºC) for 5-10 minutes;
- preparing alkaline washing solutions in hot water at a temperature of 60-70 °C and washing for 10-30 minutes;
- rinsing and removing the remnants of the washing solution with warm or hot water for 5-10 minutes;
- washing 1-2 times a week with acidic cleaning solutions;
- flushing the system before starting work with hot water and a disinfectant solution;

- treatment of communications with acid solutions with a temperature of 60-70 °C to remove milk stone is carried out after preliminary rinsing, intermediate rinsing with hot water at a temperature of 70 °C for 5 minutes, after which they are washed with an alkaline solution.

The technological mode of the process of washing the milk duct systems determines a set of mainly such indicators as:

- speed of washing solution during washing;
- solution temperature;
- concentration of detergents;

Effective washing is possible at the speed of movement of the washing solution, which is sufficient to detach and carry away particles of dirt and sediments with the flow of liquid. At an unreasonably high speed, the energy costs for pumping the solution increase sharply. The speed of liquid when washing milking units depends on the amount of rarefaction created by vacuum units, on the hydraulic parameters of the circulation lines, and on the supply of pumping milk pumps.

At the same time, the amount of speed required to detach particles depends on the size and density of the particle, its shape, the roughness of the cleaned surface and the properties of the washing liquid. It should be noted that the researchers did not reach a consensus regarding the speed of movement of the washing solution, which corresponds to the most effective technological process of washing: the speed in different studies varies within 0.5-1.5 m/s [114].

In order to intensify the mixing of the solution, which significantly affects the rate of removal of impurities, it is proposed to install various devices for creating a turbulent flow of liquid in milk-conducting systems [114]. Research has proven that high-quality cleaning can be achieved with a developed turbulent flow regime of the washing solution (at $Re > 10^4$-10^5) [115],because in this case, the most favorable conditions are created for the mechanical effect of the flow on pollution particles.

The quality of washing milk duct systems of milking plants is directly proportional to the temperature of the washing solution [115]. As the temperature rises, the physico-chemical activity of the washing solution increases, the adhesion energy at the phase separation boundary (washing solution – contamination) decreases, the kinematic viscosity of the washing solution decreases, which increases turbulence. The effectiveness of the detergent's effect on pollution, other things being equal, directly depends on the temperature of the solution. However, its increase above 70°C does not lead to a noticeable increase in washing ability [115]. Therefore, the temperature regime was brought closer to this indicator.

A radical way of maintaining the necessary sanitary and hygienic condition of milking equipment is the use of highly effective means of sanitary processing. All sanitary products according to their properties and purpose can be divided into three groups [116]: acid, alkaline and disinfectant detergents.

When studying the duration of washing, it was established that the quality of cleaning improves with an increase in processing time [117]. So, the researchers note that the circulatory washing of milk duct systems should last 5-40 minutes. depending on the materials from which their elements are made.

5 THEORETICAL STUDIES OF THE WASHING PROCESS OF THE MILK PIPE LINE OF THE MILKING INSTALLATION

5.1 Structural and technological scheme of the milking line washing system of milking plants

In accordance with the objectives of the research, the following elements are included in the traditional milking line washing system of the milking plant: air injectors and photosensors for determining pollution (Fig. 5.1). To determine the modes of operation of the proposed air injectors, it is necessary to theoretically and experimentally establish the functional dependences of the influence of the hydrodynamic parameters of the movement of the two-phase washing solution through the milk line of milking units of various types under their influence.

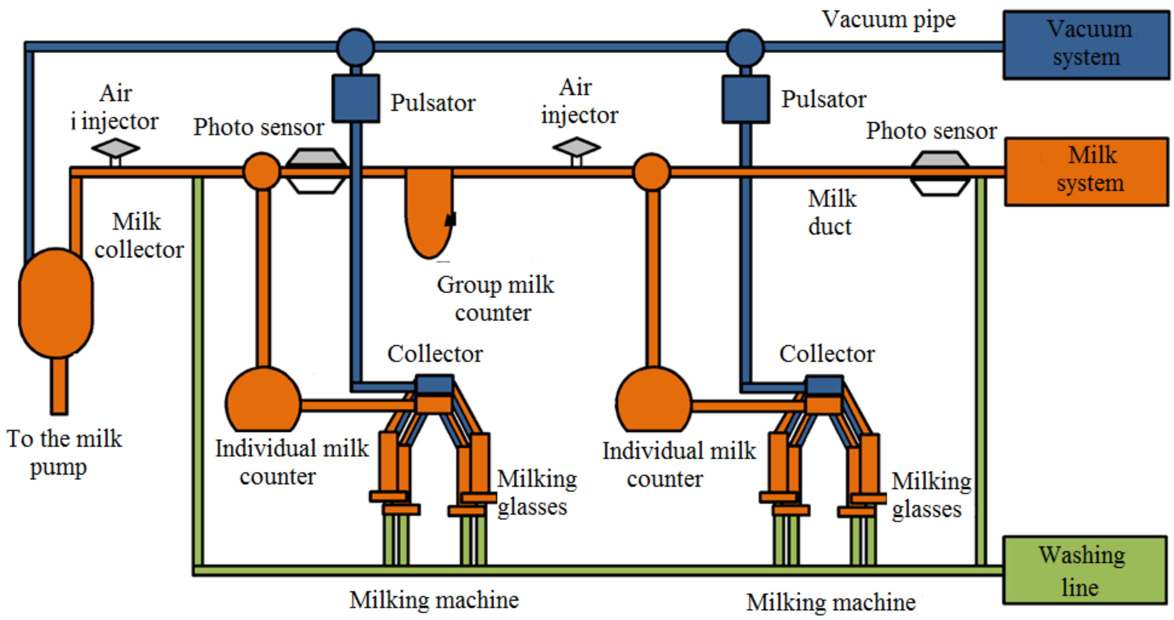


Fig. 5.1. Structural and technological scheme of the milking line washing system of milking installations

In order to create and use a system for washing the milking unit with automated control of the operating parameters of the air injector, it is necessary to develop and research a photosensor for determining the contamination of the milk line.

5.2 Justification of the conditions for overcoming adhesion forces between milk deposits and the surface of the milk line

The first stage of theoretical research is the calculation of the strength of adhesion between milk deposits and the surface of the milk line. The value of the indicated force is necessary to determine the lowest permissible speed of movement of the washing solution.

According to the conducted analysis of the factors and consequences of ineffective washing of milking installations, let us assume that milk deposits are presented in the form of liquid drops of milk origin. Let's make a calculation scheme, which is shown in fig. 5.2. According to it, the following forces act on the deposits along the axis Ox.

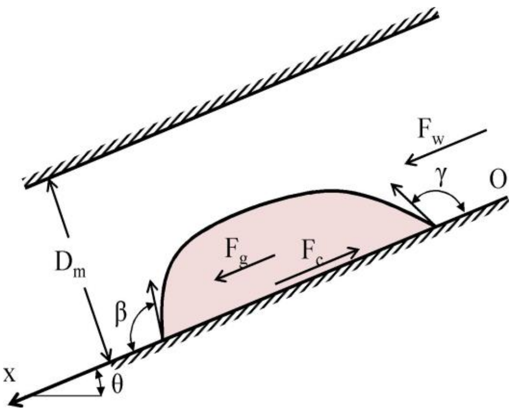


Fig. 5.2. Calculation scheme of the action of deposition forces in the form of a drop of liquid of milk origin

1. Gravity (projection on the axis of the milk line):

$$F_g = \rho_m V_m g \sin\theta, \tag{5.1}$$

ρ_m –sediment density, kg/m^3;

g – acceleration of free fall, m/s^2;

θ – the angle of inclination of the milk line, degrees;

V_m – deposit volume, m^3.

According to studies [118], the volume of a deposit in the form of a drop of liquid can be calculated using the formula

$$V_m = \frac{\pi(1-\cos\beta)^2(2+\cos\gamma)r_k^3}{24\sin^3\beta}, \tag{5.2}$$

β, γ – edge angles of drops, degrees;

r_k –radius of the contact area of the drop, m [118]:

$$r_k = \sqrt{\frac{24\sin^3\beta(\cos\gamma-\cos\beta)\sigma_{mf}}{\rho_m g(1-\cos\beta)^2(2+\cos\beta)\sin\theta}}, \tag{5.3}$$

σ_{mf} – surface tension of a drop between the washing solution and the deposit, N/m.

According to studies [118], the strength of adhesion can be represented as

$$F_c = c_f\sigma_{mf}r_k(\cos\gamma-\cos\beta), \tag{5.4}$$

c_f – empirical coefficient $c_f \approx 1{,}5$.

We will write the Stokes force (viscous friction force) in the following form:

$$F_w = c_{f2}\pi\mu_f r_k u_f\sqrt{Re_f}, \tag{5.5}$$

c_{f2} – empirical coefficient $c_{f2} \approx 1{,}8$ [119].

μ_f – dynamic viscosity of the washing solution, Pa·s;

u_f –speed of washing solution, m/s;

Re_f – Reynolds number;

$$Re_f = \frac{u_f D_m\rho_f}{\mu_f}, \tag{5.6}$$

P_f –density of washing solution, kg/m³;

D_m – diameter of the milk line, m;

The condition for the destruction of adhesion between deposits and the surface of the milk line is

$$F_w > F_c - F_g. \tag{5.7}$$

Substituting expressions (5.1)–(5.6) into (5.7), we have

$$u_f > \left[\frac{(\pi-c_f)\sigma_{mf}(\cos\gamma-\cos\beta)}{c_{f2}\pi\sqrt{D_m\rho_f\mu_f}}\right]^{2/3}. \tag{5.8}$$

With the use of the Mathematica software package, a graphical interpretation of the dependence (5.8) was obtained at $\mu_f = 8{,}9\cdot10^{-4}$ Pa·s, $\rho_f = 1000$ kg/m³ shown in fig. 5.3.

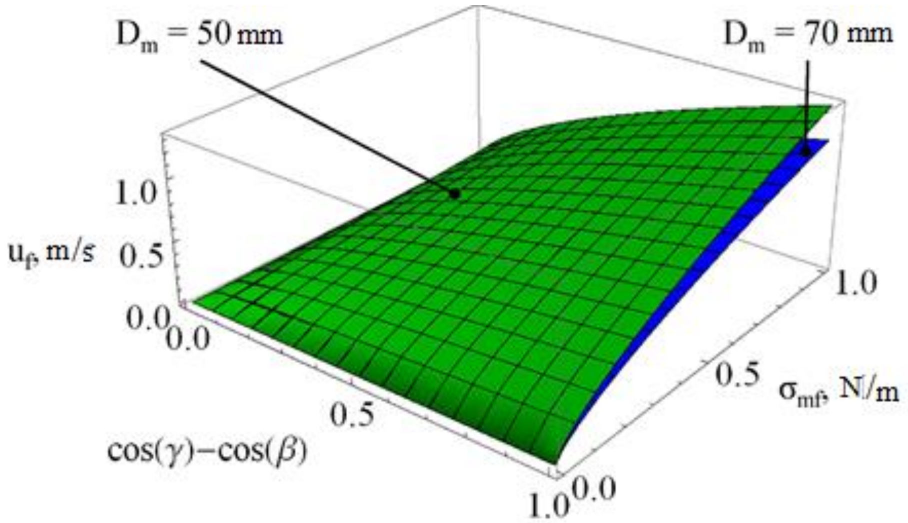


Fig. 5.3. Dependence of the smallest permissible speed of movement of the washing solution on the physical and mechanical properties of milk deposits (edge angles β, γ and surface tension between the washing solution and the deposit σ_{mf}) with different diameters of the milk line D_m

Condition (5.8) and fig. 5.3 determines the minimum flow rate of the washing solution to ensure the destruction of adhesion (cohesion) between deposits in the form of liquid drops of milk origin and the surface of the milk line.

5.3 Development of an analytical model of the movement of a two-phase washing solution through a milk line with the proposed air injector

The second stage of theoretical research is the addition of a physico-mathematical apparatus for moving the two-phase washing solution through the milk-conducting line of the milking plant, which takes into account the regulated formation of a traffic jam using an air injector.

Taking into account the equations and dependencies for the separate model of the movement of the two-phase washing solution along the milk line with the proposed air injector, the pressure drop (pressure gradient) for each phase is

$$-\left(\frac{dp_g}{dz}\right)=\frac{2f_{F0}\rho_g u_g^{\ 2}}{D}\left[1+\frac{W_g\rho_g}{\left(W_g+W_f\right)\left(\rho_g\alpha+\rho_f\left(1-\alpha\right)\right)}\right]\frac{2W_g\mu_f+W_f\mu_g}{W_g\mu_f+W_f\mu_g}+$$
$$+2\rho_g^{\ 2}u_g^{\ 2}\frac{d}{dz}\left(\frac{W_g}{W_g+W_f}\right)\left(\frac{W_g}{\rho_g\alpha\left(W_g+W_f\right)}-\frac{W_f}{\rho_f\left(1-\alpha\right)\left(W_g+W_f\right)}\right)+ \tag{5.9}$$
$$+\frac{\rho_g^{\ 2}u_g^{\ 2}}{\left(W_g+W_f\right)^2}\frac{d\alpha}{dz}\left(\frac{W_f^{\ 2}}{\rho_f\left(1-\alpha\right)^2}-\frac{W_g^{\ 2}}{\rho_g\alpha^2}\right)+g\rho_g\alpha\sin\theta;$$

$$-\left(\frac{dp_f}{dz}\right)=\frac{2f_{F0}\rho_f u_f^{\ 2}}{D}\left[1+\frac{W_f}{W_g+W_f}\left(\frac{\rho_f}{\rho_g\alpha+\rho_f\left(1-\alpha\right)}\right)\right]\frac{W_g\mu_f+2W_f\mu_g}{W_g\mu_f+W_f\mu_g}+$$
$$+2\rho_f^{\ 2}u_f^{\ 2}\frac{d}{dz}\left(\frac{W_f}{W_g+W_f}\right)\left(\frac{W_f}{\rho_f\alpha\left(W_g+W_f\right)}-\frac{W_g}{\rho_g\left(1-\alpha\right)\left(W_g+W_f\right)}\right)+ \tag{5.10}$$
$$+\frac{\rho_f^{\ 2}u_f^{\ 2}}{\left(W_g+W_f\right)^2}\frac{d\alpha}{dz}\left(\frac{W_g^{\ 2}}{\rho_g\left(1-\alpha\right)^2}-\frac{W_f^{\ 2}}{\rho_f\alpha^2}\right)+g\rho_f\left(1-\alpha\right)\sin\theta.$$

indices "g" and "f" refer to air and liquid, respectively;

p – pressure, Pa;

z – coordinate along the milk supply line, m;

f_{F0} – the coefficient of friction, which is defined as a function of the Reynolds number ($Re = \rho uD/\mu$) and relative roughness of the pipe (ε/D);

ε – absolute roughness of the milk line, m;

g – acceleration of free fall, m/s^2;

ρ – phase density, kg/m^3;

u – phase speed, m/s;

μ – dynamic viscosity of the phase, Pa·s;

D – diameter of milk line, m;

θ – the angle of inclination of the milk supply line relative to the horizon;

α – the proportion of air in the area of the milk line;

W – mass flow rates of phases, kg/s.

The air injector provides periodic air flow into the milk line, i.e. the maximum mass flow rates of the phases at z = 0 are as follows:

$$W_f\Big|_{z=0} = const;$$

$$W_g(t)\Big|_{z=0} = \begin{cases} 0, & nt_{inj} + nt_p \le t \le nt_{inj} + (n+1)t_p, n \in N; \\ W_g\Big|_{z=0} = const, & nt_{inj} + (n+1)t_p \le t \le (n+1)t_{inj} + (n+1)t_p, n \in N; \end{cases} \qquad (5.11)$$

where t_{inj} – air injector intake cycle duration, s;

t_p – air injector pause duration, s;

$W_g\big|_{z=0}$ – maximum air mass flow at z = 0, kg/s;

$W_f\big|_{z=0}$ – the maximum mass flow of the washing solution at z = 0, kg/s.

To find the total pressure drop in the milk line of the milking unit with the proposed air injector, it is necessary to add the left and right parts of equations (5.9) and (5.10):

$$-\left(\frac{dp}{dz}\right) = -\left(\frac{dp_g}{dz}\right) - \left(\frac{dp_f}{dz}\right). \qquad (5.12)$$

For a separate flow model, the share of air in the area can be calculated using the formula [119]

$$\alpha = \left[1 + \frac{\rho_g}{\rho_f}\left(\frac{W_f}{W_g}\right)\left(0{,}4 + 0{,}6\sqrt{\frac{\dfrac{\rho_f}{\rho_g} + 0{,}4\dfrac{W_f}{W_g}}{1 + 0{,}4\dfrac{W_f}{W_g}}}\right)\right]. \qquad (5.13)$$

Since the system of differential equations (5.9)–(5.13) is quite complex for analytical calculation, since it contains a periodic function of the mass flow of air, therefore the study of the movement of the two-phase washing solution through the milk line of the milking plant was carried out using the STAR-CCM+ software package, which was implemented on the basis of finite element method [120, 121]. Due to the fact that the results of the numerical simulation of the flow of two-phase washing solution significantly depend on the turbulence model used, the choice of the computational grid, the number of its nodes, boundary conditions and the computational algorithm, a verification was carried out that ensured the convergence of the results.

5.4 Modeling the movement of a two-phase washing solution along a horizontal milk line with the proposed air injector

The third stage of theoretical research is the preliminary numerical modeling of the movement of the two-phase washing solution along the horizontal milk line of the milking plant. The scheme of the numerical experiment is shown in fig. 5.4.

The initial parameters for numerical simulation are the following. The milk line of the milking unit is a straight horizontal pipe with a diameter of D_m = 50 mm and length L = 5 m. On the left side of the diagram, an injector with a diameter of D_m = 10 mm. The continuum grid of the milking line of the milking plant was formed on the basis of the generator of the surface grid and the generator of polyhedral cells. At the same time, the basic size of the grid was 0.001 m (Fig. 5.5).

Numerical modeling was carried out on the basis of the following physical models: multiphase interaction, isothermal fluid energy equation, gravity field, k-ε turbulence model, Reynolds-averaged Navier-Stokes equation, separated flow, multiphase equation of state, volume of fluid (VOF), Eulerian multiphase.

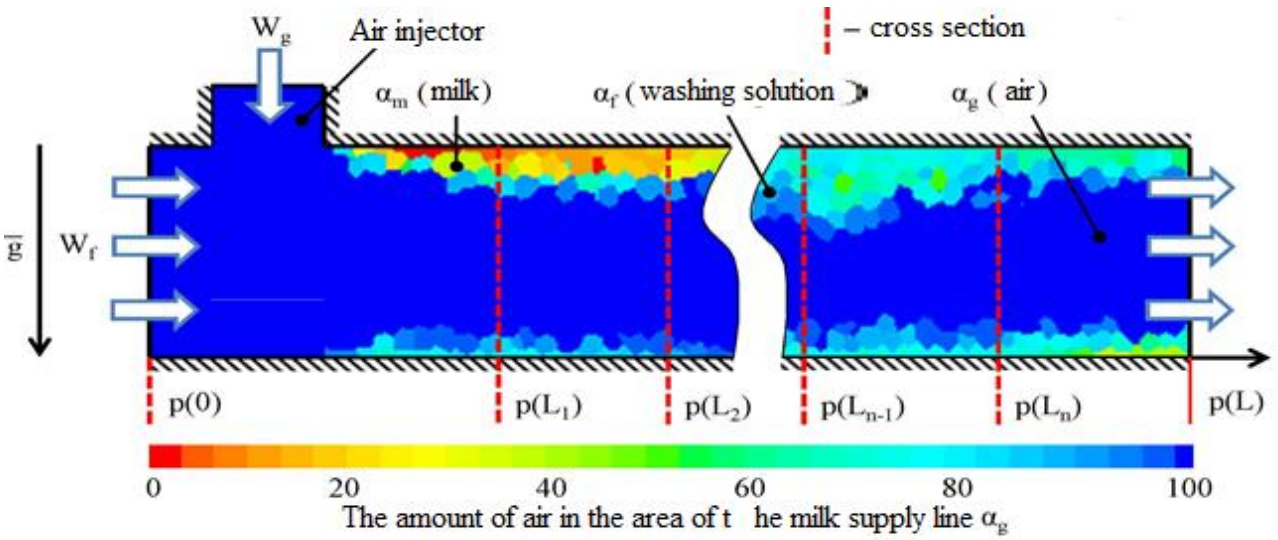


Fig. 5.4. Scheme of a numerical experiment of the process of movement of a multiphase medium along a horizontal milk-conducting line of a milking installation

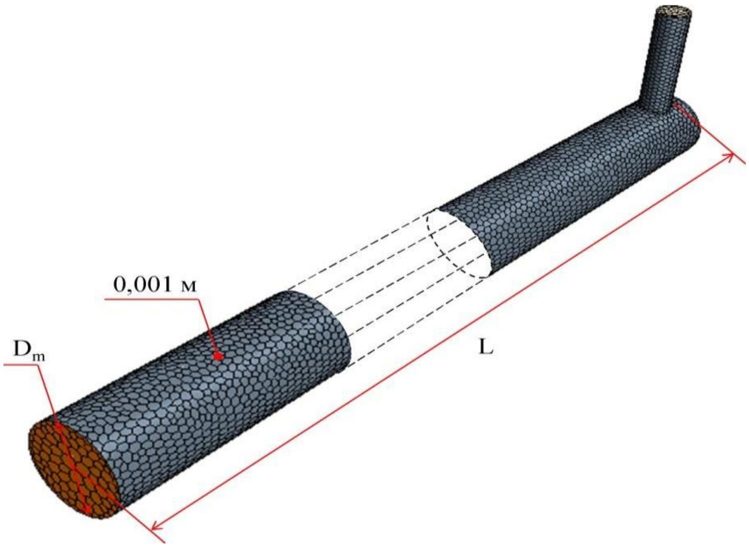


Fig. 5.5. Continuum grid of the milking line of the milking plant

As initial data, it was assumed that the washing solution had a constant density during the movement $\rho_f = 997{,}6$ kg/m^3, dynamic viscosity was $\mu_f = 8{,}88 \cdot 10^{-4}$ Pa·s. The milk also had a constant density during movement $\rho_m = 1027$ kg/m^3, its dynamic viscosity was $\mu_m = 2{,}72 \cdot 10^{-3}$ Pa·s. пthe atmosphere obeyed the ideal gas equations. The dynamic viscosity of air was $\mu_g = 1{,}85 \cdot 10^{-5}$ Pa·s, molecular weight 28.9 kg/mol [131].

At the initial moment (initial conditions) it was assumed that the entire volume of the horizontal rectilinear milk line was filled with milk, then $\alpha_m = 100$ %. At the same time, the vacuum pressure was p = 45 kPa. Further, a mass flow of air was implemented on the left border $W_g = 0{,}001$ kg/s, on the right – constant vacuum pressure p(L) = 45 kPa, and the injector nozzle was completely closed (boundary conditions).

After 16 s (the time was selected from the condition of stabilization of the content of milk and air in the volume of the milk line), the air flow stopped. Instead, a massive flow of washing solution was implemented on the left border $W_f = 0{,}2$ кг/с.

Studies were conducted for four options: the injector is permanently closed, the injector is permanently open, and the injector is periodically opened (1 s and 9 s) and closed (1 s and 9 s).

An open injector connects the internal volume of the milk line with atmospheric pressure and admits air.

In the process of numerical modeling, the dynamics of the vacuum pressure in the cross sections at a distance from the left border p(0 m), p(1 m), p(2 m), p(3 m), p(4 m), p(5 m) were determined) and the dynamics of the content of the components of the multiphase environment: washing solution α_f, air α_g, milk α_m (Fig. 5.4).

For the first option, when the injector is permanently closed, a graph of the dynamics of changes in the content of the components of the multiphase medium in the milk line of the milking plant was plotted, which is shown in Fig. 5.6.

At the first stage (from 0 to 16 s), as noted in the research methodology, milk was replaced by air in the volume of the milk line. Then, the value of milk and air content stabilized and became, respectively α_m = 34,4 % and α_g = 65,6 %. At the second stage, the supply of washing solution, which replaces air and milk, was implemented. At 31.3 s and beyond, the value of the milk content was α_m = 5,4 %, which, as a result of the calculation, corresponded to the average thickness of the layer and drops of milk of 0.68 mm, which remained on the walls of the milk line.

The dynamics of the vacuum pressure change for the first option, when the injector is permanently closed, is shown in Fig. 5.7. As can be seen from the figure, the vacuum pressure is practically unchanged for all sections of the milk line and is 45 kPa.

For the second option, when the injector is constantly open, a graph of the dynamics of changes in the content of the components of the multiphase medium in the milk line of the milking plant was plotted, which is shown in Fig. 5.8.

The first stage (from 0 to 16 s) takes place similarly to the previous option. At the second stage, the supply of washing solution, which replaces air and milk, was implemented. At the same time, the injector opens and remains in this state during the entire process, as a result of which atmospheric air enters through it. At 20.2 s and beyond, the value of the milk content stabilizes and becomes α_m = 3,9 %. Since the reason for this residual content of milk is its adhesion, the calculated average thickness of the layer on the walls of the milk line is 0.49 mm.

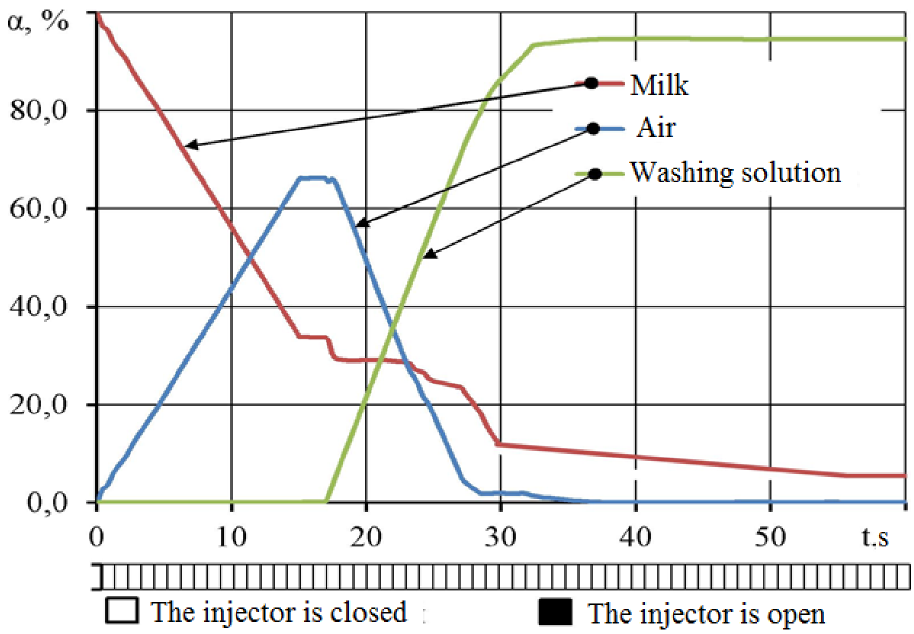


Fig. 5.6. The dynamics of changes in the content of the components of the multiphase medium in the milk line of the milking plant with the injector permanently closed

According to formulas (5.2) and (5.3), the thickness of the droplet layer held on the walls by adhesion should not exceed 1.96 mm, that is, numerical modeling adequately takes into account the phenomenon of adhesion.

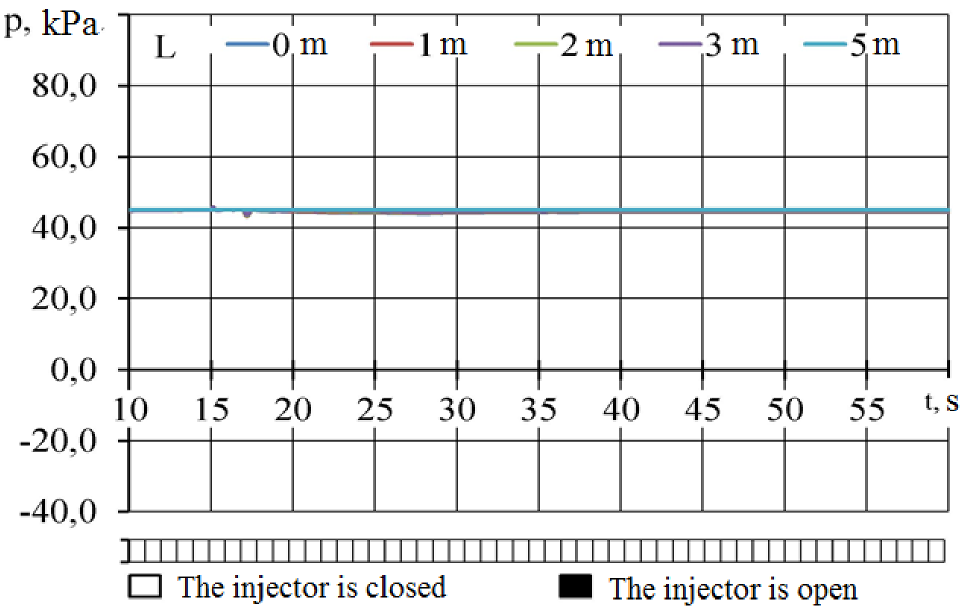


Fig. 5.7. Dynamics of changes in vacuum pressure in the milk line of the milking plant with the injector permanently closed

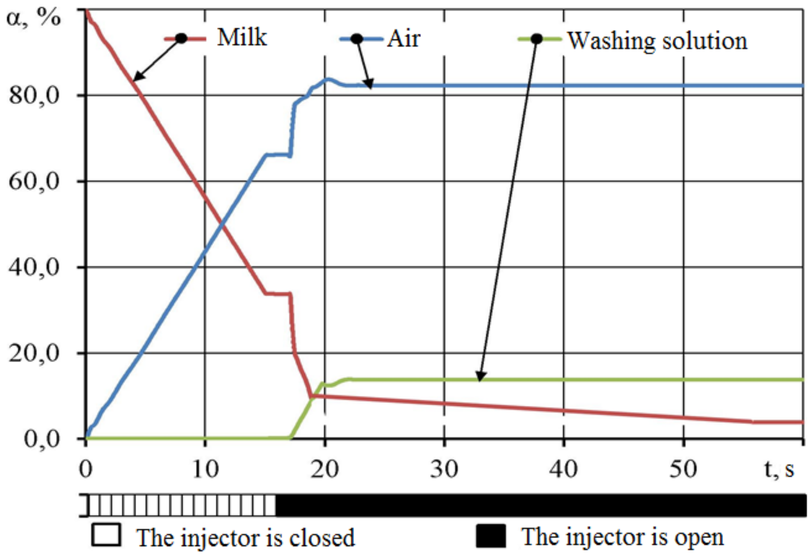


Fig. 5.8. Dynamics of changes in the content of the components of the multiphase environment in the milk line of the milking plant with the injector constantly open

The dynamics of vacuum pressure for the second option, when the injector is constantly open, is shown in fig. 5.9. As can be seen from the figure, when the injector is open, the milk line communicates with atmospheric pressure and the vacuum pressure in all areas first decreases to a value of -17.1 kPa (which is greater than atmospheric), and then damping oscillations occur and after 5.2 s stabilizes at a value of 45 kPa .

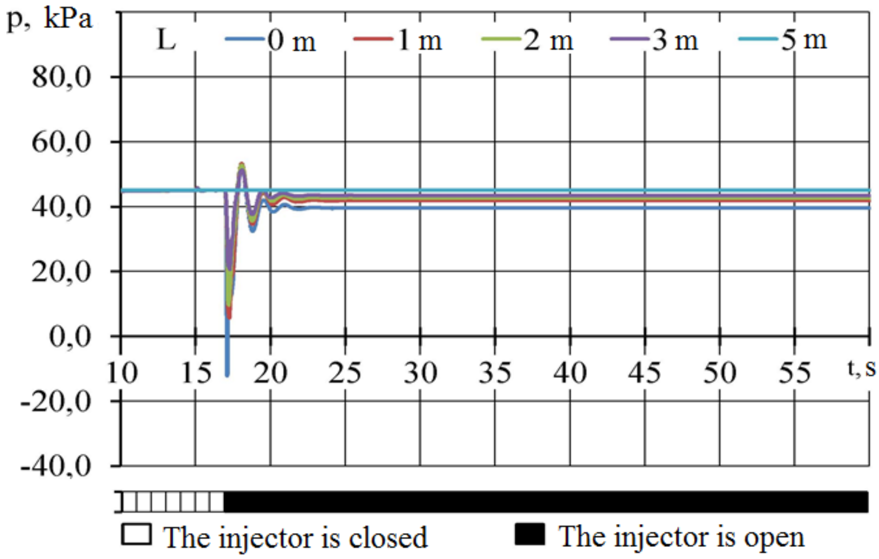


Fig. 5.9. Dynamics of changes in vacuum pressure in the milk line of the milking plant with the injector constantly open

For the third option, when the injector periodically opens (1 s) and closes (1 s), a graph of the dynamics of the content of the components of the multiphase medium in the milk line of the milking plant was drawn, which is shown in Fig. 5.10.

The first stage (from 0 to 16 s) takes place similarly to the previous options. At the second stage, the supply of washing solution, which replaces air and milk, was implemented. At the same time, during the entire process, the injector periodically opens and closes at intervals of 1 s. At 21.6 s and beyond, the value of the milk content stabilizes and becomes α_m = 2,1 %. This indicates the remaining milk on the walls of the milk line with an average layer thickness of 0.27 mm.

The dynamics of vacuum pressure change for the third option, when the injector periodically opens (1 s) and closes (1 s), is presented in Fig. 5.11. As can be seen from the figure, every time the injector is opened, the milk line communicates with atmospheric pressure, and the vacuum pressure in all areas first decreases to an average value of -39.7 kPa (which is greater than atmospheric), and then sharply increases to an average value of 91.2 kPa. Further, these vacuum pressure fluctuations are repeated. The analysis of the drawing allowed us to draw a conclusion about the decrease in vacuum pressure when moving away from the injector.

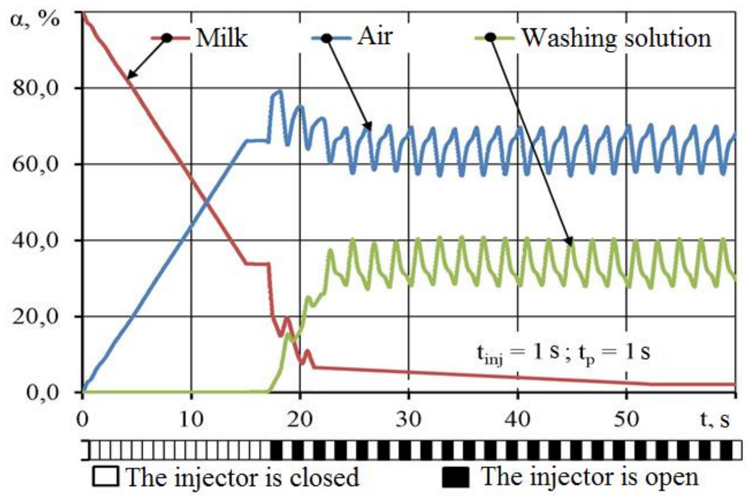


Fig. 5.10. The dynamics of changes in the content of the components of the multiphase environment in the milk line of the milking plant for variants when the injector periodically opens and closes at intervals of 1 s

In fig. 5.12 shows the graph of the maximum p_{max} and minimal p_{min} vacuum pressure depending on the distance L to the injector at p_w = 45,0 kPa, t_{inj}= 1s, t_p = 1s.

The analysis of the figure makes it possible to assert about the damping of vacuum pressure fluctuations when moving away from the injector according to the exponential law (the approximation of the obtained data was carried out in the Mathematica software package):

$$p_{max} = 46{,}1 \cdot e^{-0{,}264\,L} + 45{,}0, \tag{5.14}$$

$$p_{min} = -80{,}3 \cdot e^{-0{,}245\,L} + 45{,}0. \tag{5.15}$$

According to ISO 5707 [100] and ISO 6690 [101], permissible vacuum pressure fluctuations are 2.5 kPa, therefore, equating the difference of equations (5.14) and (5.15) to 2.5 kPa, we determine the value of the distance to the injector:

$$46{,}1 \cdot e^{-0{,}264\,L} + 80{,}3 \cdot e^{-0{,}245\,L} = 2{,}5 \rightarrow L = 12{,}8 \text{ m}. \tag{5.16}$$

That is, the smallest distance between the injectors at p_w = 45,0 kPa, t_{inj}= 1 s, t_p = 1 s should be L = 12,8 m.

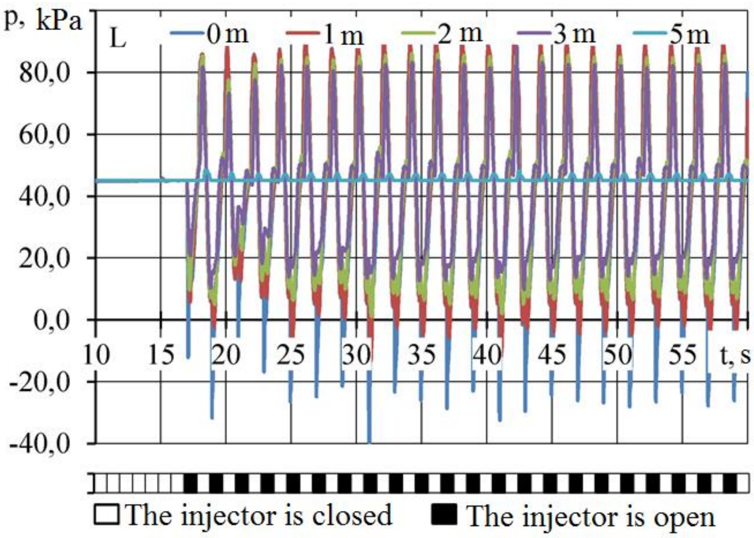


Fig. 5.11. The dynamics of changes in the vacuum pressure in the milking line of the milking unit for options when the injector periodically opens and closes at intervals of 1 s

For the fourth option, when the injector periodically opens (9 s) and closes (9 s), a graph of the dynamics of changes in the content of the components of the multiphase medium in the milk line of the milking plant was plotted, which is presented in Fig. 5.13.

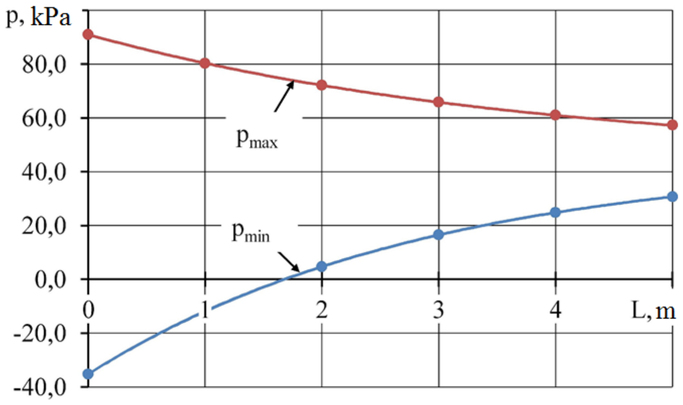


Fig. 5.12. Dependence of the maximum p_{max} and minimal p_{min} of vacuum pressure from the distance L to the injector for options when the injector periodically opens and closes at intervals of 1 s

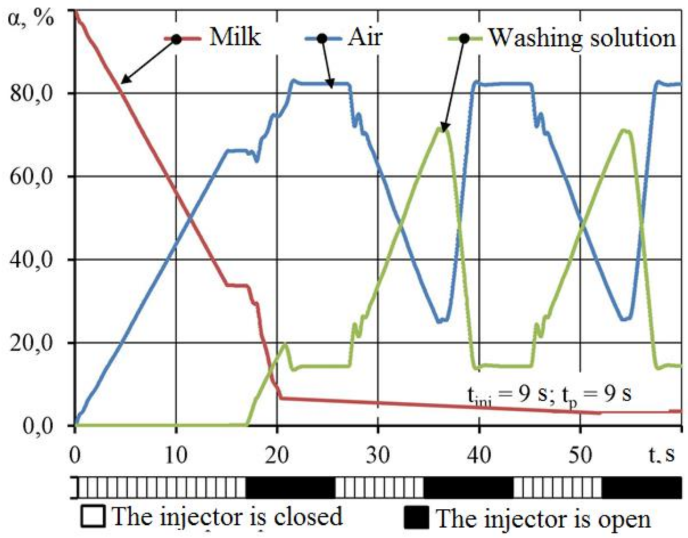


Fig. 5.13. The dynamics of changes in the content of the components of the multiphase environment in the milk line of the milking plant for options when the injector periodically opens and closes at intervals of 9 s

The first stage (from 0 to 16 s) takes place similarly to the previous options. At the second stage, the supply of washing solution, which replaces air and milk, was implemented. At the same time, during the entire simulation, the injector is periodically opened and closed at intervals of 9 s. At 20.5 s and beyond, the value of the milk content stabilizes and becomes α_m = 3,5 %. This indicates the remaining milk on the walls of the milk line with an average layer thickness of 0.43 mm.

The dynamics of vacuum pressure for the fourth option, when the injector periodically opens (9 s) and closes (9 s), is presented in Fig. 5.14. As can be seen from the figure, every time the injector is opened, the milk line is connected to the atmospheric pressure and the vacuum pressure in all areas initially decreases to the average value –137,1 kPa (which is greater than atmospheric), and then sharply increases to an average value of 72.2 kPa. Further, these vacuum pressure fluctuations are repeated. Attenuation of the vacuum pressure (to 5% of the average value) occurs in 5.2 seconds.

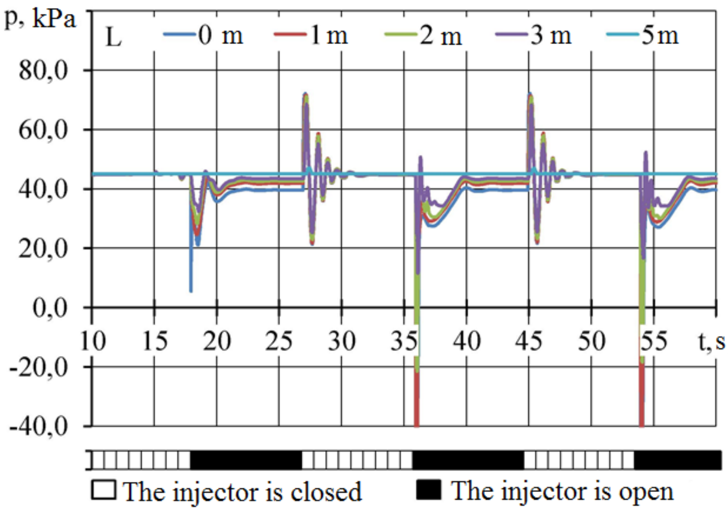


Fig. 5.14. The dynamics of changes in the vacuum pressure in the milking line of the milking unit for options when the injector periodically opens and closes at intervals of 9 s

Analysis of fig. 5.14 made it possible to draw a conclusion about the decrease in vacuum pressure when moving away from the injector.

In fig. 5.15 shows the graph of the maximum p_{max} and minimal p_{min} vacuum pressure depending on the distance L to the injector at p_w = 45,0kPa, t_{inj}= 9 s, t_p = 9 s. The approximation of the obtained data in the Mathematica software package makes it possible to assert the damping of vacuum pressure fluctuations when moving away from the injector according to the exponential law:

$$p_{max} = 27{,}2 \cdot e^{-0{,}421\,L} + 45{,}0, \tag{5.17}$$

$$p_{min} = -145{,}9 \cdot e^{-0{,}321\,L} + 45{,}0. \quad (5.18)$$

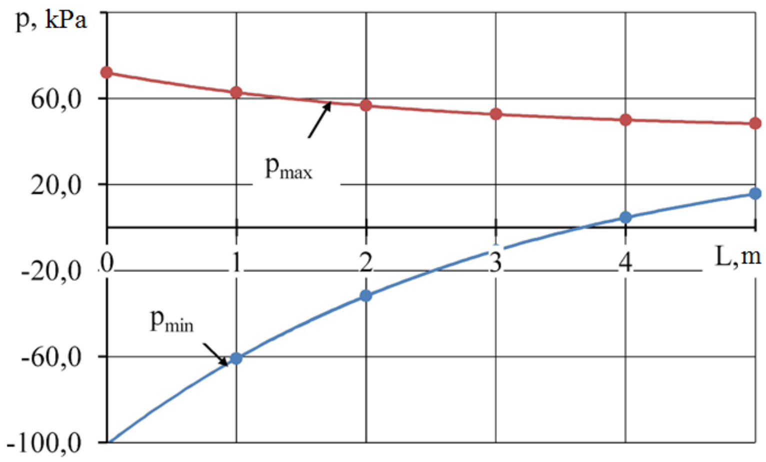


Fig. 5.15. Dependence of the maximum and minimum vacuum pressure on the distance to the injector for options when the injector periodically opens and closes at intervals of 9 s

Similarly to the previous option, having equated the difference of equations (5.17) and (5.18) to 5.5 kPa, we determine the value of the distance to the injector:

$$27{,}2 \cdot e^{-0{,}421\,L} + 145{,}9 \cdot e^{-0{,}321\,L} = 2{,}5 \rightarrow L = 12{,}7 \text{ m}. \quad (5.19)$$

That is, the smallest distance between the injectors at p_w = 45,0 kPa, t_{inj}= 9 s, t_p = 9 s should be L = 12.7 m, which practically coincides with the value for the third option (deviation - 0.2 m).

According to the above results, it can be stated that the use of a periodic injector allows to reduce the content of milk in the milk line faster and to a greater extent, which indicates a better washing process.

Preliminary numerical modeling of the movement of the two-phase washing solution along the horizontal milk line of the milking unit made it possible to determine the dynamics of vacuum pressure at a distance from the injector. The dynamics of changes in the vacuum pressure for options when the injector periodically opens and closes is presented as damped oscillations in the range from –100.9 kPa to 72.2 kPa. Sudden change in pressure (0.1-0.6 s) causes periodic water shocks that affect the reduction of milk adhesion on the surface of the milk line.

5.5 Justification of the modes of operation of the air injector of the flushing system

The fourth stage of theoretical research is the substantiation of the operating modes of the air injector of the washing system of the milking plant.

The phenomenon of hydraulic shock is caused by a sudden change in the phase distribution of the two-phase washing solution flow caused by the periodic operation of the injector. This leads to a sudden change in the momentum of the two-phase washing solution, causing a pressure wave that moves through the system. This pressure wave can lead both to the destruction of milk deposits on the walls of the milk duct and to possible damage to the equipment elements of the milk duct system. The processes of damage and destruction under the influence of a shock wave depend on a large number of structural features of the equipment itself (strength of the material, geometric dimensions, presence and quality of welds, soldering, joints, etc.) and the probability of their occurrence, therefore complex studies of the above processes are very difficult to conduct both from a theoretical point of view and from an experimental point of view. However, studies [132-134] indicated that the rate of pressure change $\Delta p/\Delta t$ to reduce the probability of hydraulic shock should be minimal.

Therefore, it is necessary to determine rational modes of operation of the flushing system with an air injector.

The studies were conducted on the basis of numerical modeling using the STAR-CCM+ software package. The scheme of the numerical experiment (Fig. 5.4), selected models, initial and boundary conditions are given in section 5.3. There is also a step-by-step description of the modeling procedure for the case when the injector periodically opens and closes.

The research factors were the diameter of the milk duct D_m, working vacuum pressure p_w, the duration of the air injector intake stroke t_{inj}, air injector pause duration t_p. Limits and intervals of research factors are given in table. 5.1.

In the process of numerical modeling, the dynamics of vacuum pressure changes at a distance from the left border p(0 m), p(1 m), p(2 m), p(3 m), p(4 m), p(5 m) were determined (appendix B) and the dynamics of changes in the content of the components of the multiphase washing solution α_f, air α_g, milk α_m.

The average thickness of the layer or drops of milk on the wall of the pipe is a qualitative evaluation criterion for research into the operating modes of the system for washing the milk pipes of milking equipment with an air injector. h_m, which was determined by the formula

$$h_m = \frac{D_m}{2}\left(1 - \sqrt{1 - \frac{\alpha_m}{100}}\right). \tag{5.20}$$

Table 5.1

Limits and intervals of numerical modeling factors

Level	Diameter milk duct D_m, mm (x_1)	Working vacuum pressure p_w, kPa (x_2)	The duration of the intake stroke air injector t_{inj}, s (x_3)	Duration pauses air injector t_p, s (x_4)
Upper (+1)	70	75	9	9
Average (0)	60	60	5	5
Lower (–1)	50	45	1	1
Interval	10	15	4	4

The smaller the value of the thickness of the milk layer on the wall of the milk duct hm, the better the washing process was carried out.

The criterion that limits the operating parameters of the system for washing the milk ducts of milking equipment with an air injector is the value of the pressure change during the intake cycle and pause of the air injector (speed of pressure change) $\Delta p/\Delta t$,which is calculated by the formula

$$\frac{\Delta p}{\Delta t} = \frac{p_{max} - p_{min}}{t_{inj} + t_p}. \tag{5.21}$$

The greater the rate of pressure change in the milk-conducting system of milking equipment, the greater the probability of an "uncontrolled" water hammer, which will destroy not only the layer of milk and milk deposits on the surface of its walls, but may also lead to damage to the elements of its equipment.

Rational operation modes of the system for washing the milk ducts of milking equipment with an air injector can be achieved by minimizing the value of the thickness of the milk layer on the wall of the milk duct and the rate of pressure change.

Modeling was carried out by successively sampling all levels of factors with a total number $3^4 = 81$ experiment. Next, using the Mathematica software package, a second-order regression model was determined for each of the proposed criteria.

As a result of numerical modeling and further processing of the obtained data in the Mathematica software package, the dependence of the change in the value of the thickness of the milk layer on the research factors was obtained in coded form

$$\begin{gathered} h_m = 0{,}337492 + 0{,}0816377\, x_1 + 0{,}0609942\, x_1^2 + 0{,}00981812\, x_2 - \\ - 0{,}0732213\, x_1 x_2 + 0{,}0305382\, x_2^2 + 0{,}0274136\, x_3 + \\ + 0{,}00457199\, x_1 x_3 + 9{,}09953 \cdot 10^{-6}\, x_2 x_3 + 0{,}0457208\, x_3^2 + 0{,}0731846\, x_4 + \\ + 0{,}0122056\, x_1 x_4 + 0{,}000024347\, x_2 x_4 - 0{,}0182404\, x_3 x_4 + 0{,}061028\, x_4^2. \end{gathered} \tag{5.22}$$

The statistical processing of equation (5.22) is given in the table. 5.2.

As a result of the analysis of the table. 5.2, the corresponding reduction of insignificant coefficients and decoding of equation (5.22) we finally have the dependence of the change in the thickness of the milk layer on the research factors

$$\begin{gathered} h_m = 0{,}87386 - 0{,}0378379\, D_m + 0{,}000609942\, D_m^2 + 0{,}013656\, p_w - \\ - 0{,}000488142\, D_m p_w + 0{,}000135725\, p_w^2 - 0{,}02288\, t_{inj} + 0{,}0001143\, D_m t_{inj} \\ + 0{,}00285755\, t_{inj}^{\,2} - 0{,}0324547\, t_p + 0{,}000305141\, D_m\, t_p - \\ - 0{,}00114002\, t_{inj}\, t_p + 0{,}00381425\, t_p^2. \end{gathered} \tag{5.23}$$

The minimum value of the thickness of the layer of milk on the wall of the milk duct h_m = 0,243 mm is reached at D_m = 50 mm, p_w = 45 kPa, t_{inj}= 3,55 s, t_p = 2,78 s. Fixing in turn the research factors at the specified level, graphic interpretations of dependence (5.23) are built in Fig. 5.16–5.17.

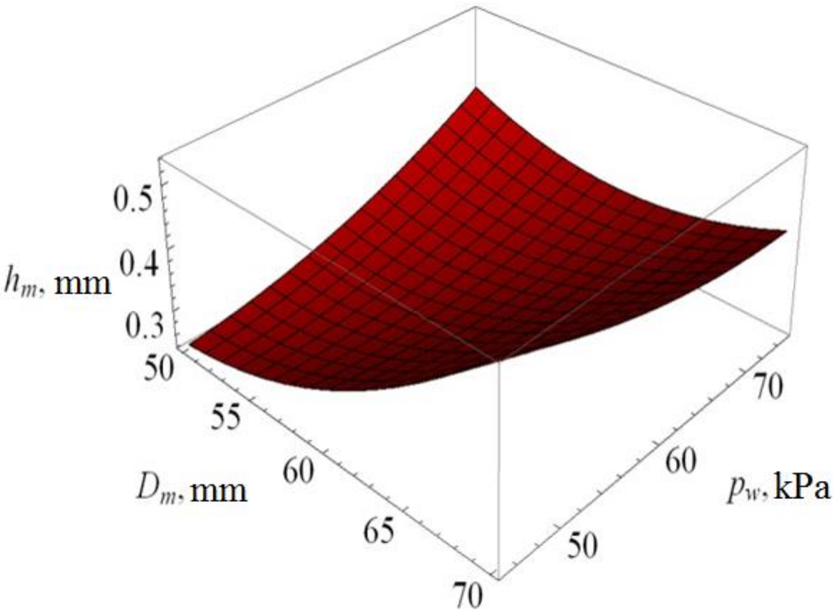


Fig. 5.16. Dependence of the value of the thickness of the layer of milk on the wall of the milk duct h_m from the diameter of the milk pipe Dm and the working vacuum pressure p_w

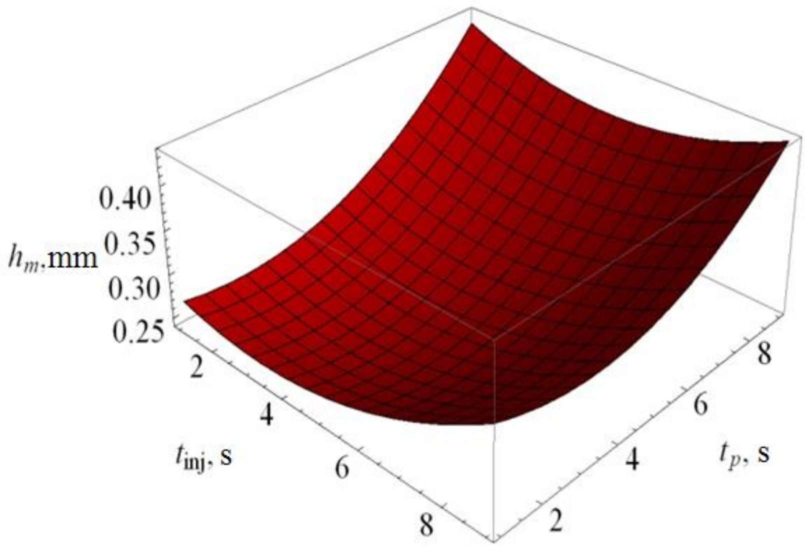


Fig. 5.17. Dependence of the value of the thickness of the layer of milk on the wall of the milk duct h_m from the duration of the air injector intake stroke t_{inj} and the duration of the pause of the air injector t_p

Fixing the value of the diameter of the milk duct D_m at the levels of 50 mm, 60 mm and 70 mm, we obtain rational values of other factors under the condition of minimizing the thickness of the milk layer:

$$h_m\,(D = 50\text{ mm},\ p_w = 45\text{ kPa},\ t_{inj} = 3{,}55\text{ c},\ t_p = 2{,}78\text{ c}) = 0{,}243\text{ mm},$$
$$h_m\,(D = 60\text{ mm},\ p_w = 57{,}5\text{ kPa},\ t_{inj} = 3{,}27\text{ c},\ t_p = 2{,}34\text{ c}) = 0{,}306\text{ mm}, \qquad (5.24)$$
$$h_m\,(D = 70\text{ mm},\ p_w = 74{,}9\text{ kPa},\ t_{inj} = 2{,}98\text{ c},\ t_p = 1{,}90\text{ c}) = 0{,}406\text{ mm}.$$

Table 5.2.

Statistical processing of the equation (5.22)

Regression coefficient	The value of the coefficient regressions	Standard error	t-statistics	P-Value
a_{00}	0,337492	0,00272132	124,018	$6{,}38376\cdot10^{-80}$
a_{10}	0,0816377	0,00111097	73,4831	$4{,}96084\cdot10^{-65}$
a_{20}	0,00981812	0,00111097	8,8374	$8{,}4597\cdot10^{-13}$
a_{30}	0,0274136	0,00111097	24,6753	$4{,}93061\cdot10^{-35}$
a_{40}	0,0731846	0,00111097	65,8743	$6{,}12415\cdot10^{-62}$
a_{12}	– 0,0732213	0,00136066	– 53,8132	$3{,}0103\cdot10^{-56}$
a_{13}	0,00457199	0,00136066	3,36013	0,00129839
a_{14}	0,0122056	0,00136066	8,97038	$4{,}90858\cdot10^{-13}$
a_{23}	$9{,}09953\cdot10^{-6}$	0,00136066	0,00668759	0,994684
a_{24}	0,000024347	0,00136066	0,0178935	0,985778
a_{34}	– 0,0182404	0,00136066	– 13,4055	$1{,}64895\cdot10^{-20}$
a_{11}	0,0609942	0,00192426	31,6974	$1{,}17348\cdot10^{-41}$
a_{22}	0,0305382	0,00192426	15,8701	$3{,}24016\cdot10^{-24}$
a_{33}	0,0457208	0,00192426	23,7602	$4{,}65679\cdot10^{-34}$
a_{44}	0,061028	0,00192426	31,715	$1{,}13378\cdot10^{-41}$
a_{00}	0,337492	0,00272132	124,018	$6{,}38376\cdot10^{-80}$

The corresponding graphic interpretation of dependence (5.23) under the conditions (5.24) is presented in Fig. 5.16 – 5.20.

Analysis of fig. 5.16–5.20 and dependences (5.24) make it possible to assert the variability of injector operating modes. The larger the diameter of the milk duct used in milking equipment, the greater the vacuum pressure must be created to ensure high-quality cleaning of its walls from milk residues. At the same time, the duration of the air injector intake cycle t_{inj} and the duration of the pause of the air injector t_p should be within 2.98-3.55 s and 1.9-2.78 s according to (5.23). The physical essence of the process can be explained as follows: at the lowest values of the injector strokes, the speed of propagation of the shock wave is high, which leads to a decrease in the time of its interaction with a layer or drops of milk; at the highest stroke values, the magnitude of the change in vacuum pressure is not large, which leads to a less destructive effect on a layer or drop of milk.

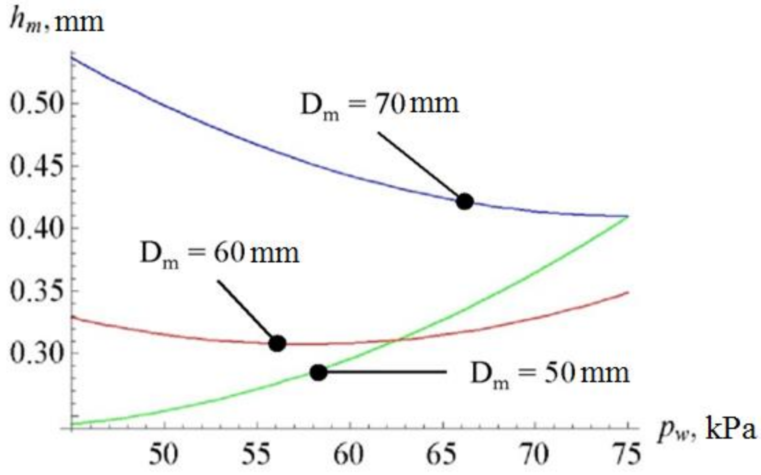


Fig. 5.18. Dependence of the value of the thickness of the layer of milk on the wall of the milk duct h_m at different values of its diameter D_m from the working vacuum pressure p_w

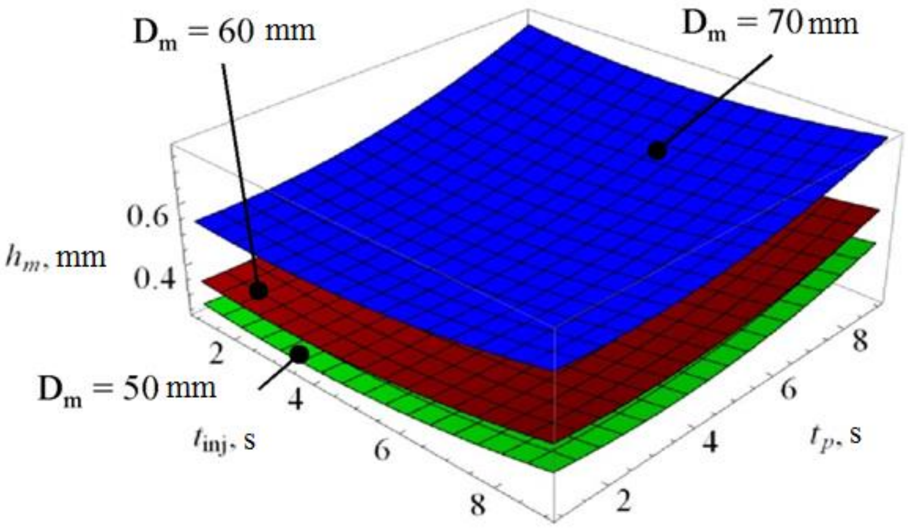


Fig. 5.19. Dependence of the value of the thickness of the layer of milk on the wall of the milk duct h_m at different values of its diameter D_m from the duration of the air injector intake stroke t_{inj} and the duration of the pause of the air injector t_p

As a result of numerical modeling and further processing of the received data in the Mathematica software package, the dependence of the pressure change during the intake cycle time and air injector pause (speed of pressure change) on the research factors in coded form was obtained

$$\Delta p/\Delta t = 19{,}0753 - 2{,}26911\ x_1 - 4{,}90715\cdot10^{-14}\ x_1^2 + 3{,}7816\ x_2 - 7{,}99361\cdot10^{-15} x_1\ x_2 - 1{,}71746\ x_2^2 - 20{,}2799\ x_3 + 2{,}36848\cdot10^{-15}\ x_1\ x_3 - 4{,}42461\ x_2\ x_3 + 16{,}1687\ x_3^2 - 9{,}10255\ x_4 + 1{,}77636\cdot10^{-15}\ x_1\ x_4 - 2{,}65746\ x_2\ x_4 + 14{,}8132\ x_3\ x_4 - 1{,}39624\ x_4^2. \qquad (5.25)$$

The statistical processing of equation (5.23) is given in the table. 5.3.

Table 5.3.

Statistical treatment of equation (5.39)

Regression coefficient	Value regression coefficient	Standard error	t-statistic	P-Value
a_{00}	19,0753	2,61737	7,28796	$4{,}97078\cdot10^{-10}$
a_{10}	– 2,26911	1,06854	-2,12356	0,0374594
a_{20}	3,7816	1,06854	3,53904	0,000741245
a_{30}	– 20,2799	1,06854	-18,9791	$1{,}96503\cdot10^{-28}$
a_{40}	– 9,10255	1,06854	-8,5187	$3{,}12889\cdot10^{-12}$
a_{12}	$-7{,}99361\cdot10^{-15}$	1,30869	$-6{,}10812\cdot10^{-15}$	1
a_{13}	$2{,}36848\cdot10^{-15}$	1,30869	$1{,}80981\cdot10^{-15}$	1
a_{14}	$1{,}77636\cdot10^{-15}$	1,30869	$1{,}35736\cdot10^{-15}$	1
a_{23}	– 4,42461	1,30869	-3,38096	0,00121742
a_{24}	– 2,65746	1,30869	-2,03064	0,0463243
a_{34}	14,8132	1,30869	11,3191	$4{,}13002\cdot10^{-17}$
a_{11}	$-4{,}90715\cdot10^{-14}$	1,85076	$-2{,}65142\cdot10^{-14}$	1
a_{22}	– 1,71746	1,85076	-0,927976	0,356801
a_{33}	16,1687	1,85076	8,73622	$1{,}28081\cdot10^{-12}$
a_{44}	– 1,39624	1,85076	-0,754414	0,453284
a_{00}	19,0753	2,61737	7,28796	$4{,}97078\cdot10^{-10}$

As a result of the analysis of the table. 5.3, the corresponding reduction of insignificant coefficients and decoding of equation (5.23), we finally have the dependence of the rate of pressure change on the research factors

$$\begin{gathered}\Delta p/\Delta t = 37{,}6294 - 0{,}226911\ D_m + 1{,}75826\ p_w - 0{,}00763316\ p_w^2 - \\ - 15{,}3799\ t_{inj} - 0{,}0737435\ p_w\ t_{inj} + 1{,}01054\ t_{inj}^2 - 3{,}37464\ t_p - \\ - 0{,}0442911\ p_w\ t_p + 0{,}925824\ t_{inj}\ t_p - 0{,}0872651\ t_p^2.\end{gathered} \tag{5.26}$$

The minimum value of the pressure change rate $\Delta p/\Delta t$ = 1,81 kPa/s is reached at D = 70 mm, p_w = 45 kPa, t_{inj}= 8,79 s, t_p = 1,0 s. Fixing in turn the research factors at the indicated level, constructed in fig. 5.22–5.23, graphic interpretations of dependence (5.26).

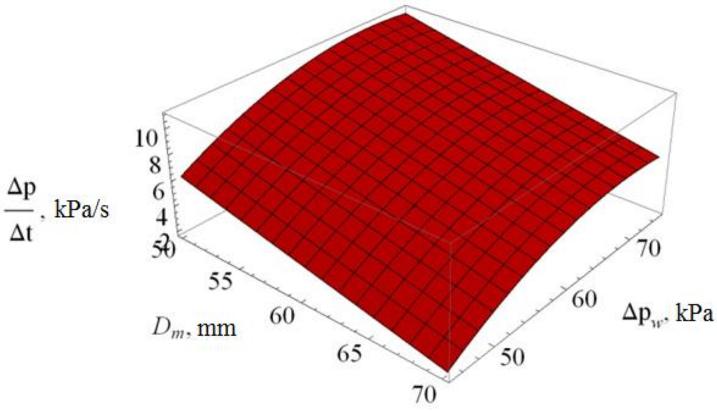


Fig. 5.20. Dependence of the value of the pressure change rate $\Delta p/\Delta t$ from the diameter of the milk duct D_m and working vacuum pressure p_w

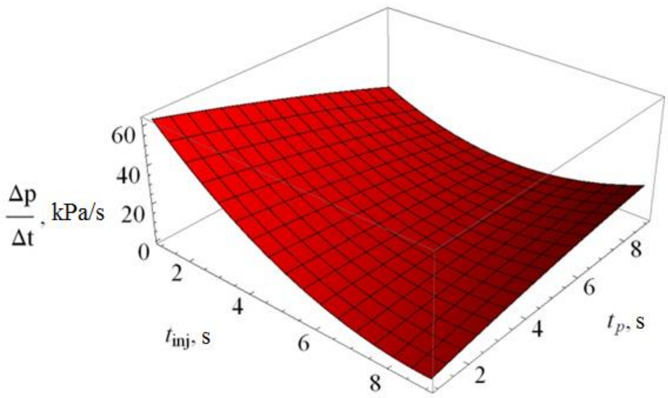


Fig. 5.21. Dependence of the value of the pressure change rate $\Delta p/\Delta t$ from the duration of the air injector intake stroke t_{inj} and the duration of the pause of the air injector t_p

Fixing the value of the diameter of the milk duct D_m at the levels of 50 mm, 60 mm, and 70 mm, we obtain rational values of other factors under the condition of minimizing the rate of pressure change:

$$\Delta p/\Delta t\ (D = 50\ \text{mm},\ p_w = 75\ \text{kPa},\ t_{inj} = 6{,}22\ \text{c},\ t_p = 9{,}0\ \text{c}) = 8{,}73\ \text{kPa/s},$$
$$\Delta p/\Delta t\ (D = 60\ \text{mm},\ p_w = 75\ \text{kPa},\ t_{inj} = 6{,}22\ \text{c},\ t_p = 9{,}0\ \text{c}) = 6{,}47\ \text{kPa/s}, \quad (5.27)$$
$$\Delta p/\Delta t\ (D = 70\ \text{mm},\ p_w = 75\ \text{kPa},\ t_{inj} = 6{,}22\ \text{c},\ t_p = 9{,}0\ \text{c}) = 4{,}2\ \text{kPa/s}.$$

The corresponding graphic interpretation of dependence (5.26) under conditions (5.27) is shown in Fig. 5.22-5.23.

Analysis of fig. 5.20–5.23 and dependencies (5.27) makes it possible to assert that the rational values of the working vacuum pressure p_w = 75 kPa, the duration of the air injector intake stroke t_{inj}= 6,22 s and the duration of the pause of the air injector t_p = 9,0 s under the condition of minimizing the rate of pressure changes do not depend on the diameter of the milk duct.

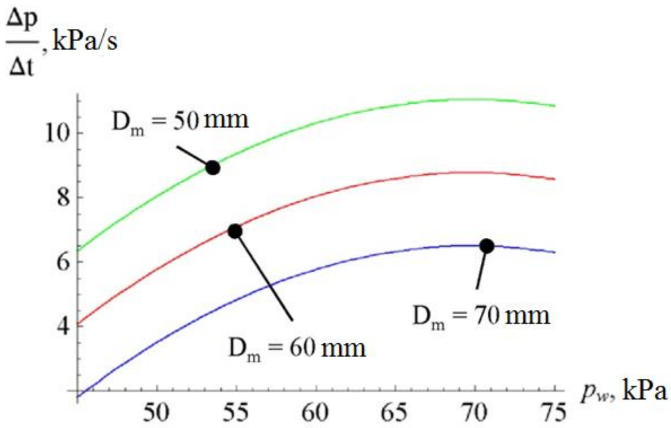


Fig. 5.22. Dependence of the value of the pressure change rate $\Delta p/\Delta t$ at different values of the diameter of the milk duct D_m from the working vacuum pressure p_w

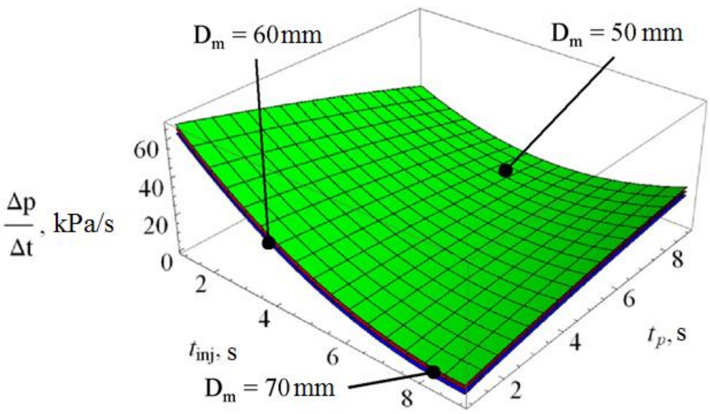


Fig. 5.23. Dependence of the value of the pressure change rate $\Delta p/\Delta t$ at different values of the diameter of the milk duct D_m from the duration of the air injector intake stroke t_{inj} and the duration of the pause of the air injector t_p

Due to the fact that the rational parameters (5.23) and (5.26) differ, it is necessary to solve a compromise problem, which boils down to minimizing the value of the thickness of the layer of milk on the wall of the milk duct and the rate of change of pressure:

$$\begin{cases} h_m(D_m, p_w, t_{inj}, t_p) \to \min, \\ \dfrac{\Delta p}{\Delta t}(D_m, p_w, t_{inj}, t_p) \to \min. \end{cases} \qquad (5.28)$$

Solving the system of equations (5.28) in the Mathematica software package for different values of the diameter of the milk duct, we obtain the corresponding rational parameters of the injector operating modes:

for D = 50 mm → p_w = 45,0 kPa, t_{inj}= 3,8 s, t_p = 2,9 s, h_m = 0,243 mm,

Δp/Δt = 27,38 кПа/с,

for D = 60 mm → p_w = 57,5 kPa, t_{inj}= 3,6 s, t_p = 2,5 s, h_m = 0,306 mm, (5.29)

Δp/Δt = 35,32 kPa/s,

for D = 70 mm → p_w = 74,5 kPa, t_{inj}= 3,4 s, t_p = 2,1 s, h_m = 0,406 mm,

Δp/Δt = 43,42 kPa/s.

CONCLUSIONS

1. According to the results of a review of research on the influence of microclimate parameters on the physiological state of animals, it was established that the air temperature in the room for their keeping has the greatest impact on their productivity.

2. The analysis of technological schemes of heat-utilizers showed that for the utilization of heat in the systems of ensuring the regulatory parameters of the air environment of livestock premises, recuperative heat-utilizers made of polymeric materials resistant to moisture and chemically active impurities are the most promising.

3. Taking into account the technological conditions of air in livestock premises (significant dustiness - up to 6 mg/m^3, high humidity - up to 80%, the presence of a high concentration of aggressive components - ammonia up to 20 mg/m^3, hydrogen sulfide - up to 10 mg/m^3, of carbon dioxide - up to 0.28%) and the results of the analysis of the designs of heat-utilizers revealed that in terms of sanitary-hygienic and operational indicators, high energy efficiency and low cost of construction, shell-and-tube heat-utilizers of the «pipe-in-pipe» type are the most suitable for the ventilation system.

4. To ensure the microclimate in livestock premises, two structural and technological schemes of a three-pipe concentric heat exchanger have been developed, which differ in the direction of flow air: direct flow and counter flow.

5. The analysis of the results of theoretical studies of the heat transfer process in concentric heat exchangers showed that the existing mathematical models do not fully describe the specified process, due to the accepted assumptions and simplifications.

6. According to the results of theoretical studies of pneumatic losses of a three-pipe concentric heat exchanger, the dependences of changes in pressure losses Δp and power N on the length of the heat exchanger L, the radius of the external duct were established r_3 and air flow supply V.

7. Based on the results of theoretical studies, a mathematical model of the heat transfer process in a three-pipe concentric heat exchanger was developed, taking into

account the phenomenon of condensation in it, which allows determining the temperature distribution of air flows along its length and its thermal power.

8. The analysis of the results of theoretical studies of the heat transfer process in the structural and technological schemes of the three-pipe concentric heat exchanger with counterflow and forward flow showed that the option with counterflow is more effective, because it provides a temperature gradient in the range from 11.6 to 15.7 °C (average value - 13.7 °C) compared to direct current - from 8.3 to 11.1 °C (average value - 9.7 °C).

9. Optimization of the results of theoretical studies allowed to determine the dependence of the design parameters of the heat exchanger (length L and radii r_1, r_2, r_3 air ducts) from the volume flow of air passing through it under the condition of the greatest useful thermal power: $L = 14{,}776 \cdot V + 3{,}7335$, $r_1 = 0{,}3619 \cdot V + 0{,}1523$, $r_1 = 0{,}343 \cdot r_3$, $r_2 = 0{,}686 \cdot r_3$ (at ambient temperature $T_c = 0$ °C).

10. Based on the results of theoretical studies, a methodology was developed and an algorithm was implemented based on it to determine the geometry of the location of the holes in the air duct of a three-pipe concentric heat exchanger for livestock premises. It was established that the distance between the holes gradually decreases to a certain value in the direction opposite to the movement of the air flow.

11. According to the results of experimental research in production conditions, it was established that for a piggery room with a ventilation system of a ground channel, more fresh air is provided in the area the animals are staying, than a ceiling ventilation system and a ventilation system through wall channels. This is evidenced by concentration CO_2 in the animal area: ventilation system of the ground channel - 0.18±0.05%, ceiling ventilation system - 0.25±0.07%, ventilation system through wall channels - 0.26±0.06%, normalized value - 0.2%. At the same time, the air flow speed for the ventilation system of the ground channel is within 0.08–0.14 m/s in winter and 0.19–0.31 m/s in summer, which is lower than other options for ventilation systems.

12. The results of experimental and numerical studies made it possible to determine temperature distributions in piggery premises with various variants of negative pressure ventilation systems. For the ventilation system of the ground channel,

the temperature field in the area the pigs are located is in the range of 20–26 °C in summer and 16–20 °C in winter. In a pig house with a ceiling ventilation system, the temperature ranges from 19 to 28 °C, while in winter it is -15 to 22 °C. The temperature field in the area the pigs stay for ventilation through wall channels is within the following limits: for the summer period – 20–30 °C, for the winter period – 12–18 °C. It is important to note that the obtained temperature values do not fully meet regulatory requirements. However, the ventilation system of the ground channel has the greatest uniformity of temperature across the machines. This happens due to the presence of turbulent air movement and bypassing solid partitions in the machines, which protect the animals from hitting them with warm air in the summer and cold air in the winter.

13. For the ventilation system of the ground duct and the ventilation system through the wall ducts on the efficiency of removing pollutants ξ was influenced by the ventilation rate, which largely depends on the age of the animal, the weight of the animal and the external temperature. For the ventilation system of the above-ground channel in production conditions, the efficiency of pollution removal ξ (in summer - 1.27±0.48, in winter - 1.37±0.41) decreased with increasing air flow rate in ventilation. For the ventilation system through wall ducts, the efficiency of removing contaminants ξ (in summer – 1.27±0.22, in winter – 1.14±0.38) increased with increasing air flow rate in ventilation, since the sampling point was located in the back of the room. In a room with a ceiling system, ventilation for the effectiveness of removing pollution ξ (in summer – 1.01±0.28, in winter – 1.02±0.35) was strongly influenced by the lying behavior of animals, and a slight decrease was observed ξ with increased ventilation.

14. The analysis of the factors of the decrease in the quality of milk as a result of bacterial contamination showed that this is the result of poor performance of the technological operation of washing milking units and the formation of milk deposits on the internal surfaces of the milk duct system. As a result of the analysis of the existing constructions of technical and technological provision of washing, it was established that the most effective are the circulation systems of washing with the regulated formation of a traffic jam. In order to increase efficiency and save resources

(by reducing the consumption of air, hot water, energy and operating costs), the washing process of milking units should be adaptive based on data obtained from monitoring tools for assessing the condition of the surfaces of the milk-conducting system and the hydrodynamic parameters of the movement of the two-phase washing solution. which is achieved by using air injectors based on automated control.

15. As a result of theoretical studies of the phenomenon of adhesion between deposits and the surface of the milk line, the dependence of the smallest permissible speed of movement of the washing solution on the physical and mechanical properties of milk deposits (edge angles β, γ and surface tension between the washing solution and the deposit σ_{ml}) with different diameters of the milk line D_m.

16. According to the results of theoretical studies, the physical and mathematical apparatus of the movement of the two-phase washing solution along the milk-conducting line was supplemented, which is based on the equations of the principle of superposition of forces and, as a result, pressures, continuity of flow, laws of conservation of mass, momentum and energy. Since the resulting system of differential equations is quite complex for analytical calculation, the study of the flow of two-phase washing solution through the milk line of the milking plant was carried out using the STAR-CCM+ software package, which is implemented on the basis of the finite element method.

17. As a result of the preliminary numerical simulation of the process of washing the milk line of the milking plant using an injector in the STAR-CCM+ software package, the dynamics of the vacuum pressure change at a distance from the injector was determined (p(0 m), p(1 m), p(2 m), p (3 m), p(4 m), p(5 m)) and the dynamics of changes in the content of the components of the multiphase medium (washing solution α_f, air α_g, milk α_m) for four options: injector permanently closed, injector permanently open and injector periodically opened (1 s and 9 s) and closed (1 s and 9 s). It was established that the use of a periodic injector allows to reduce the content of milk in the milk line faster and to a greater extent, which indicates a better washing process.

18. As a result of the numerical simulation of the process of washing the milk line of the milking plant using an injector in the STAR-CCM+ software package, the

dependences of the rate of pressure change were established $\Delta p/\Delta t$ and changes in the thickness of the milk layer on the wall of the milk duct h_m at different values of its diameter D_m from the working vacuum pressure p_w, duration of the air injector intake stroke t_{inj} and the duration of the pause of the air injector t_p.

19. Solving the compromise problem, which is reduced to minimizing the value of the thickness of the milk layer on the wall of the milk duct and the rate of change of pressure for different values of the diameter of the milk duct, the corresponding rational parameters of the injector operating modes were obtained: at D = 50 mm → p_w = 45,0 kPa, t_{inj}= 3,8 s, t_p = 2,9 s, h_m = 0,243 mm, $\Delta P/\Delta t$ = 27,38 kPa/s, at D = 60 mm → p_w = 57,5 kPa, t_{inj}= 3,6 s, t_p = 2,5 s, h_m = 0,306 mm, $\Delta P/\Delta t$ = 35,32 kPa/s, at D = 70 mm → p_w = 74,5 kPa, t_{inj}= 3,4 s, t_p = 2,1 s, h_m = 0,406 mm, $\Delta P/\Delta t$ = 43,42 kPa/s.

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